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PLANT ENGINEERING HANDBOOK

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PLANT ENGINEERING HANDBOOK

PREPARED BY A STAFF OF SPECIALISTS

WILLIAM STANIAR, M.E.
Formerly Mechanical Power Engineer,
E. I. du Pont de Nemours & Co.

EDITOR-IN-CHIEF

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PREFACE

This book is written for those people in industry who are responsible for organization, design, construction, operation, and maintenance.

Experience over the years has shown the need for a book that could be used to solve the day-to-day problems of industry. This book is a compendium of industrial know-how and presents in compact form material that appears in greater detail in specialized reference works. It is not intended to replace the usual type of engineering handbook; it is to supplement and extend.

The book deals with industry's consumption and conservation of basic resources, with management engineering, with industrial construction, fire protection, maintenance, with industrial power and its uses, with air handling, materials handling, and the handling of services. It deals with the more specialized subjects of instrumentation, bearings and lubrication, engineering materials, and mechanical power. It covers many other subjects vital to efficient industrial and institutional operations.

The book was written by men with broad training and experience, selected from forty-six of the leading industrial organizations and five of the foremost educational institutions in the United States. These men have had complete freedom of expression.

WILLIAM STANIAR

WILMINGTON, DEL.
May, 1950

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"Plant Engineering Handbook" is a product of American industry and knowledge. The idea of such a book was conceived during 1943 and now has become a reality through the generous cooperation and assistance of the leading industrial organizations and universities of America. The Editor-in-Chief desires to acknowledge with deep gratitude the willing cooperation of these institutions.

The Editor-in-Chief is most fortunate in having had the friendly encouragement, support, and assistance of a great many people and therefore acknowledges a real debt to his authors, critics, and others whose combined efforts built this handbook. He particularly desires to thank the following:

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William Staniar

Wilmington, Del.
May, 1950

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SECTION 1

BASIC INDUSTRIAL COSTS AND COST REDUCTION

BY SYDNEY STEELE

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I Basic industrial costs
  A Use of the tables
  B Plots of basic costs vs. calendar years for selected industries
  C Tables of basic costs for selected industries
II Basic-cost reduction

BASIC INDUSTRIAL COSTS AND COST REDUCTION*

BY SYDNEY STEELE, B.A., M.A., PH.D. (CANTAB.)

Sales Promotion Manager, Fischer & Porter Co.

BASIC INDUSTRIAL COSTS

When you figure your costs, do you think only in terms of dollars, pounds sterling, pesos, cruzeiros, francs, or other national currencies? Is money the only criterion used in making the profit-and-loss statement for your industry, or your plant, or your business? Have you considered *basic costs*, and do you make a man-power, materials, and utilities balance sheet for your enterprise? The real success of a business depends primarily on the efficient use of those basic-cost elements, which are actually any country's fundamental resources.

In this section basic costs, not dollar costs, have been determined for many United States industries and have been put in tabular form. By basic costs are meant the man-hours of labor, kilowatt-hours of electric energy, weights of raw material, etc., per unit of production of goods or services. For the most part, these costs are shown for the calendar years 1929, 1933, and 1939. Complete data were not available for all the years, however, and figures for other years are used for some industries. The costs were calculated from Census of Manufactures and Bureau of Labor Statistics data, supplemented by information from several trade associations, the Minerals Yearbook, and other sources. The Bureau of the Census 1939 classification of industries was used as a guide in assembling the tables of basic costs for manufacturing industries, and data for some nonmanufacturing industries are also included. It is regrettable that no census of manufactures for the United States has been made since 1939 and that data on man-hours of labor are sketchy prior to 1933. Reasonably complete figures should eventually be available for the calendar year 1947 and for subsequent years.

Since almost every enterprise depends to some extent upon the efforts of the engineer, he has a continuing responsibility for the economical production, distribution, and use of goods and services. No less important and closely related are his responsibilities in the personnel field— to promote and maintain harmonious relations with those to whom he reports and with those who work with and for him. In the broader view, he should also bend his efforts toward improving his company's standing in the community. Good personnel and public relations form a favorable climate in which the engineer can make his greatest contributions to industrial advancement, and the engineer himself can do much to create the climate in which his efforts can flourish.

Unless the engineer knows the basic costs of the operations under his charge, he is severely handicapped. Without comparable national-average data he may have difficulty in setting realistic goals of improvement toward which to work. And without data covering a wide span of years, there is no reliable criterion of industrial progress. In the long run, no enterprise can succeed if its basic costs are excessive.

In order to measure industrial performance and to make possible comparisons

---

* The author's thanks are due Richard S. Witmeyer, of the du Pont Company, for considerable assistance in preparing and checking the tables of basic costs. The author is also indebted to Edmond E. Lincoln, du Pont Company economist, for helpful comments and suggestions and to G. M. Read, du Pont Company chief engineer, for permission to reproduce Fig. 1-3.

Information on work being done on interindustry relations by Professor Leontief at Harvard and in the Bureau of Labor Statistics under Marvin Hoffenberg was obtained through the office of Senator C. Douglass Buck.

The author also acknowledges with thanks the cooperation of officers of many trade associations in furnishing pertinent information or helpful suggestions in connection with data shown in the tables of basic costs.

between industries, standards of reference must be set up which are adequate for the purpose and for which a reasonable amount of data is available.

The national currency is frequently used in this kind of work, but the practice has some defects, notably the following:

1. Changes in the intrinsic worth of a national currency make comparisons between different calendar years misleading, if not meaningless. As anyone who has experienced 20 years of economic life can testify, even the comparatively stable dollar has fluctuated considerably in intrinsic worth. Roughly twice as many dollars, for example, must be given in exchange for automobiles and radios in 1947 as in 1940 and almost three times as many for some basic food products.
2. If it is difficult to make comparisons between different calendar years in terms of one national currency, the problem of translating from one national currency to another over a period of years is well-nigh insurmountable.
3. Even within national boundaries, the unit cost of similar items entering into a given industry, measured in terms of the national currency, may vary with the geographical location. An example of this is the lower unit cost of man-hours which, some years ago, was a factor in the migration of textile and other industries to the southern United States which still continues.

Also used in measuring various aspects of industrial activity are a great many indexes. Usually some year, or period of years, is selected as a base, or 100, and data for other years are shown as percentages of the base period. Not all indexes have the same base period, and the base periods are changed from time to time Furthermore, the items taken into consideration in setting up the indexes sometimes vary. This necessitates a recalculation of data and recorrelation of data between indexes

In measuring industrial performance, the engineer thinks first in terms of man-hours of labor, kilowatt-hours of electric energy, tons of coal, cubic feet of gas, gallons of water, weights of raw materials used, etc. His measures of production are equally concrete: number of articles produced, weight of products made, ton-miles of freight handled, etc. These are the true criteria of industrial performance, and they are independent of currency variations. They enable direct and meaningful comparisons to be made between any number of years for which adequate data are available Moreover, these basic costs translate directly into the units used by other countries, and the conversion factors do not vary from year to year.

Most industries may be considered as separate entities, drawing on and consuming the resources of the country and producing in return goods and services. Among the resources consumed are man power; electric energy, fuels, and water; minerals, agricultural products, output from other industries; etc. The figures presented later in this section were prepared with this point of view in mind.

Determining consumption of resources, or basic costs of those industries which turn out a single product, such as flour, cement, or cigarettes, presented little difficulty. For others with a variety of products the following general procedure was adopted. The primary purpose of the industry was considered. For example, the wholesale meat-packing industry produces edible animal products for sale in the grocery stores, the supermarkets, the delicatessens, and for export. The total weight of these items alone is taken as the industry's production. From the over-all point of view no distinction is made between steaks, lard, lamb chops, sausage, and canned meats and this industry's other products for human consumption. Along with these items are manufactured by-products such as dog and cat food, livestock feeds and fertilizer, oils, hides, etc., which testify to the efficiency of the industry in its use of raw materials. These are not included in the production figures for the industry.

A different type of analysis is used in the transportation field. United States class I railroads and air-transport data are based on thousands of "traffic units," a traffic unit being defined as ton-miles of revenue freight plus three times revenue passenger-miles.

Use of the Tables

The tables of basic costs (pages 6 to 21) may be used in several ways, among which are the following·

1 They present United States national-average basic-cost data for selected industries to serve as a yardstick in measuring performance of individual producing units and for setting goals of improvement

2 By multiplying the basic costs given in the tables by the price of man-hours, utilities, and raw materials at any geographical location, an estimate of these costs in terms of any currency can be obtained

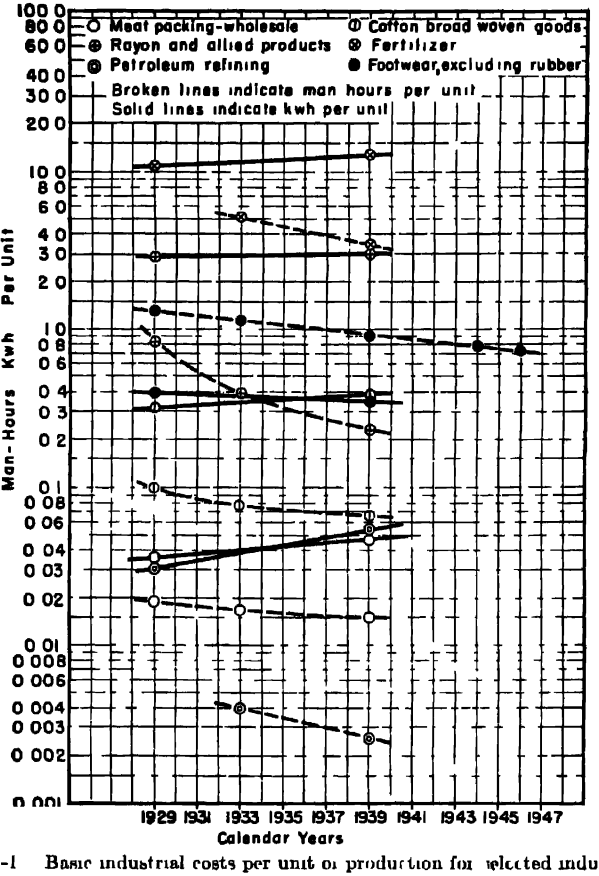

Fig. 1-1 Basic industrial costs per unit of production for selected industries

3 For those planning to broaden their business interests the tables provide information that can be used to estimate the major resources which will be required in setting up a new enterprise

4 Technological changes over the years are shown by altered consumption or substitution of basic resources and by changes in utility requirements

5 Changes in man-hours per unit of production reflect changes in productivity of labor brought about by worker efficiency changes in management, equipment, and procedure per cent utilization of equipment, and other factors

It is clear from the above that the tables reflect technological changes and variations in labor requirements * These are the only two factors that affect basic costs

---

* When more complete data are available for all major industries tables of this sort can be used in studies of interindustry relations See Leontief Dr Wassily A New Approach to the Problem of Market Analysis, American Management Association Marketing Series No 5) 1945 and Cornfield Jerome W Duane Evans and Marvin Hoffenberg Full Employment Patterns 1950, Bureau of Labor Statistics Serial No R 1868 1947

in any industry, and they are difficult, if not impossible, to follow in terms of unconstant, though indispensable, currencies

In Figs 1-1 and 1-2, man-hours, kilowatt-hours, and other basic costs per unit of production in several industries are plotted on a calendar-year base to demonstrate trends (The complete data are given in the tables that follow Fig 1-2) In spite of the fact that data for so few years were available, distinct trends can be observed

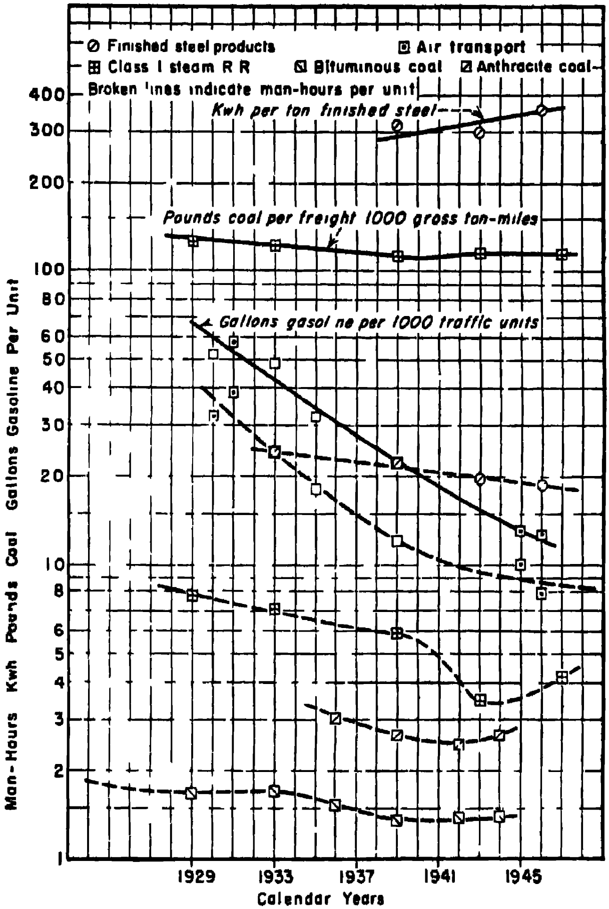

Fig 1-2 Basic industrial costs per unit of production for selected industries

In general, man-hours per unit are decreasing while kilowatt-hours per unit tend to remain constant or increase It will be noticed that the "youngest" industries for which data are plotted, the air-transport and the rayon and allied products industries, show the most marked rates of change *

Following the tables is a brief discussion of basic-cost reduction and of the methods that can be used to achieve this end

* This may have interesting philosophical implications The reader is referred to E R Dewey and E F Dakin "Cycles" Henry Holt and Company Incorporated New York 1947 and in particular to p 49 Fig 8 where various American industries are located with approximate accuracy on a basic industrial growth curve

BASIC INDUSTRIAL COSTS

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1-2	2 Tobacco Manufactures	Cigarettes Cigars	9
1-3	3 Textile-mill Products and Other Fiber Manufactures	Cotton Broad-woven Goods Rayon Broad-woven Goods	10
1-4	4 Apparel and Other Finished Products[a]	None	
1-5	5 Lumber and Timber Basic Products[a]	None	
1-6	6 Furniture and Finished Lumber Products[a]	None	
1-7	7 Paper and Allied Products	Paper and Paperboard Mills Pulp Mills	10-11
1-8	8 Printing, Publishing and Allied Trades[a]	None	
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[a] Tables omitted. Satisfactory data not currently available (1948).

GENERAL NOTES. The many gaps in the tables which follow suggest that an extension of industrial and governmental thinking along basic cost lines may be timely. It is regrettable that in the welter of statistical information gathered and disseminated (at high man-hour cost) there is comparatively little that can be used to determine the rate at which we are depleting our basic resources. The following symbols are used throughout the tables:

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BASIC INDUSTRIAL COSTS

Table 1-1. Food and Kindred Products

Industry Basic unit	BAKERY PRODUCTS Pound		
Year	1929	1933	1939
Units produced $\times 10^6$	13 354[a]	10 275[a]	14 242[a]
Labor man-hr/unit	*	*	0 035
Electric energy kwh/unit	0 029	*	0 038
Principal raw materials lb/lb			
Flour (all kinds)	0 631	*	0 647
Sugar	0 073	*	0 088
Eggs	*	*	*
Shortenings	0 042	*	0 054
Milk	0 027	*	0 033
Yeast	0 008	*	0 010
Salt	*	*	0 013
Total accounted for	0 781	*	0 845

[a] Pounds of bread (and other yeast raised products) crackers pretzels soft cakes pies and pastries.

Industry Basic unit	CANDY AND OTHER CONFECTIONERY PRODUCTS Pound		
Year	1929	1933	1939
Units produced $\times 10^6$	1 902 2[a]	1 639 3[a]	2 050[a]
Labor man hr/unit	*	0 060	0 048
Electric energy kwh/unit	0 056†	*	0 053
Principal raw materials lb/lb			
Sugar	0 37	*	0 37
Dairy products	0 042	*	0 0023
Cornstarch and gelatin	*	*	0 0121
Chocolate	0 094	*	0 11
Coconut	0 011	*	0 015
Cocoa products	0 0062	*	0 0108
Corn sirup	0 23	*	0 25
Nuts chicle and fruits	0 126	*	0 13
Total accounted for	0 9822	*	0 9002

[a] Data from National Confectioners Association

* Data unavailable from sources checked

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BASIC INDUSTRIAL COSTS

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BASIC INDUSTRIAL COSTS

Table 1-1 Food and Kindred Products

Industry	BAKERY PRODUCTS		
Basic unit	Pound		
Year	1929	1933	1939
Units produced ×10⁶	13 354[a]	10 275[a]	14 242[a]
Labor man hr/unit	*	*	0 035
Electric energy kwh/unit	0 029	*	0 038
Principal raw materials lb/lb			
Flour (all kinds)	0 631	*	0 647
Sugar	0 073	*	0 088
Eggs	*	*	*
Shortenings	0 042	*	0 054
Milk	0 027	*	0 038
Yeast	0 008	*	0 010
Salt	*	*	0 013
Total accounted for	0 781	*	0 845

[a] Pounds of bread (and other yeast raised products) crackers pretzels soft cakes pies and pastries.

Industry	CANDY AND OTHER CONFECTIONERY PRODUCTS		
Basic unit	Pound		
Year	1929	1933	1939
Units produced ×10⁶	1 902 2[a]	1 639 3[a]	2 050[a]
Labor man hr/unit	*	0 060	0 048
Electric energy kwh/unit	0 056†	*	0 053
Principal raw materials lb/lb			
Sugar	0 37	*	0 37
Dairy products	0 042	*	0 0023
Cornstarch and gelatin	*	*	0 0121
Chocolate	0 094	*	0 11
Coconut	0 014	*	0 015
Cocoa products	0 0062	*	0 0108
Corn sirup	0 23	*	0 25
Nuts chicle and fruits	0 126	*	0 18
Total accounted for	0 8922	*	0 9002

[a] Data from National Confectioners Association

* Data unavailable from sources checked

† Author's estimate

BASIC INDUSTRIAL COSTS

Table 1-1. Food and Kindred Products (*Cont.*)

Industry	CANE-SUGAR REFINING			
Basic unit	Short Ton			
Year	1929	1933	1937	1939
Units produced	5,115,307	3,938,000	4,515,582	4,126,508
Labor, man-hr/unit	*	*	6.1	6.7
Electric energy, kwh/unit	17.2	*	*	48.9
Principal raw materials, short tons/short ton				
Raw sugar treated	1.04	*	*	*

Industry	CREAMERY BUTTER		
Basic unit	Pound		
Year	1929	1933	1939
Units produced, $\times 10^6$	1,650†	1,763[a]	1,782[a]
Labor, man-hr/unit	*	0.023†	0.024
Electric energy, kwh/unit	0.089†		0.103
Principal raw materials, lb/lb			
Milk[a]	20		20
Refined cane sugar			0.008

[a] Data from U.S. Department of Agriculture, "Agricultural Statistics."

Industry	FLOUR AND OTHER GRAIN MILLS		
Basic unit	Hundredweights of Flour, Meal, Feed, Etc.		
Year	1929	1933	1939
Units produced	404,363,690	307,933,770	324,819,032
Labor, man-hr/unit	*	0.17	0.17
Electric energy, kwh/unit	1.45†	*	2.09
Principal raw materials, lb/cwt			
Wheat	81.05	86.05	93.85
Corn	12.98	8.20	11.50
Oats	4.31	1.17	1.63
Rye	1.43	1.46	1.42
Barley	1.41	0.86	1.04
Buckwheat, rice, etc	0.30[a]	0.09[a]	0.58
Sugars, shortening, enrichment, etc	*	*	0.65
Other and mixed grains	0.29	0.21	*
Total accounted for	101.77	98.04	110.67

[a] Buckwheat only.

Industry	ICE CREAM AND ICES			
Basic unit	Gallon			
Year	1929	1933	1937	1939
Units produced, $\times 10^6$	279.8	166.4	300.0	278.0
Labor, man-hr/unit	*	0.22	0.15	0.14
Electric energy, kwh/unit	0.94†	*	0.88†	1.07
Principal raw materials, lb/gal				
Milk	13.6	13.4	14.0	16.3
Refined cane sugar	*	*	*	0.47

* Data unavailable from sources checked. † Author's estimate.

BASIC INDUSTRIAL COSTS

Table 1-1. Food and Kindred Products (*Cont*)

Industry Basic unit	MEAT PACKING WHOLESALE Pounds for Human Consumption[a]		
Year	1929	1933	1939
Units produced $\times 10^6$	16 852	15 221	17 046
Labor man hr/unit	0 019	0 017	0 015
Electric energy kwh/unit	0 036†	*	0 047
Principal raw materials lb on the hoof/lb			
Cattle	0 53	0 56	0 64
Calves	0 05	0 06	0 07
Sheep and lambs	0 08	0 10	0 10
Hogs	0 73	0 75	0 63
Total accounted for	1 39[b]	1 47[b]	1 44[b]

[a] Fresh cured and canned meats canned and other sausage and sausage casings lard

[b] Part of the excess represents dog and cat food, livestock feed and fertilizer animal skins, wool, hair etc

Table 1-2 Tobacco Manufactures

Industry Basic unit	CIGARETTES Thousands of Cigarettes Weighing Less Than 3 Lb/1 000				
Year	1929	1933	1937	1939	1944
Units produced[a]	122 392 381	114 874 217	169 969 320	180 666 824	323,583 888
Labor man hr/unit	*	*	0 29	0 28	*
Electric energy kwh/unit	0 31†	*	*	0 30	*
Principal raw materials lb/1 000 cigarettes					
Leaf tobacco[a] unstemmed equivalent[b]	2 83	2 84	2 82	2 81	2 84
Sugar	*	*	*	0 15	*
Licorice	*	*	*	0 06	*
Total accounted for	2 83	2 84	2 82	3 02	2 84

[a] Data from U S Treasury Department annual reports of the Commissioner of Internal Revenue

[b] Stemmed leaf and scraps etc converted to unstemmed equivalent on the basis of 3 lb stemmed to 4 lb unstemmed

Industry Basic unit	CIGARS Thousands of Cigars Weighing More Than 3 Lb/1 000				
Year	1929	1933	1937	1939	1944
Units produced[a]	6 518 533	4 300 015	5 303 369	5 197 627	5 198 670
Labor man hr/unit	*	*	19 5	18 1	*
Electric energy kwh/unit	2 6†	*	*	5 6	*
Principal raw materials lb/1 000 cigars					
Leaf tobacco[a] unstemmed equivalent[b]	23 15	24 18	24 11	23 49	24 89
Sugar	*	*	*	0 025	*
Licorice	*	*	*	0 01	*
Total accounted for	23 15	24 18	24 11	23 525	24 89

[a] Data from U S Treasury Department annual reports of the Commissioner of Internal Revenue

[b] Stemmed leaf and scraps etc , converted to unstemmed equivalent on the basis of 3 lb stemmed to 4 lb unstemmed

* Data unavailable from sources checked † Author's estimate

BASIC INDUSTRIAL COSTS

Table 1-6. Textile-mill Products and Other Fiber Manufactures

Industry Basic unit	COTTON BROAD WOVEN GOODS Yards of Material 36 In Wide		
Year	1929	1933	1939
Units produced $\times 10^6$	8 512	8 089	9 045
Labor man-hr/unit	0 10†	0 077†	0 066
Electric energy kwh/unit	0 32†	*	0 38
Principal raw materials lb/yd			
Raw cotton short staple	*	*	0 27
Long staple	*	*	0 032
Rayon staple	*	*	0 0036
Shorn and pulled wool scoured wt	*	*	0 0004
Misc noils and mill waste	*	*	0 0062
Total accounted for	*	*	0 3122
Average wt/yard 36 in wide lb	0 277	0 263	0 274

Industry Basic unit	RAYON BROAD WOVEN GOODS Linear Yards of Material over 12 In Wide
Year	1939[a]
Units produced $\times 10^6$	1 341
Labor man hr/unit	0 099
Electric energy kwh/unit	0 32
Principal raw materials (lb/yd)	
Raw cotton short staple	0 028
Long staple	0 012
Rayon staple	0 030
Shorn and pulled wool etc scoured wt	0 00026
Cotton noils and mill waste	0 0017
Raw silk other fiber etc	0 00037
Total above items	0 07233
Total yarn consumed	0 2345
Average wt/lin yd lb	0 2104

[a] Data comparable with 1939 not available from sources checked for years 1929 to 1933

Table 1-7 Paper and Allied Products

Industry Basic unit	PAPER AND PAPERBOARD MILLS Short Tons of Paper and Paperboard[a]		
Year	1929	1933	1939
Units produced	11 140 235	9 190 017	13 509 612
Labor man hr/unit	*	20 0	17 2
Electric energy kwh/unit	630†[b]	*	670[b]
Principal raw materials short tons/short ton			
Wood pulp	0 565	*	0 64
Manila stock	0 0116	*	0 0047
Rags	0 066	*	0 035
Waste paper	0 345	*	0 32
Straw	0 052	*	0 038
Other fiber (cotton etc)	0 0045	*	0 0086
Casein	0 0017	*	0 0013
Clay	0 042	*	0 032
Rosin	0 0034	*	0 0027
Rosin sizing	0 009	*	0 007
Other nonfibrous materials alum starch etc	0 022	*	0 042
Total accounted for	1 1222	*	1 1312

[a] Newsprint groundwood book cover text writing wrapping tissue absorbent building and other papers and boards

[b] A considerable amount of power is used direct from the prime mover without putting it into the form of energy *e g* grinders connected directly to water wheels

* Data unavailable from sources checked † Author s estimate

BASIC INDUSTRIAL COSTS

Table 1-7. Paper and Allied Products (*Cont*)

Industry Basic unit	PULP MILLS Short Tons of Wood Pulp		
Year	1929	1933	1939
Units produced	4,862,885	4,276,204	6,993,334
Labor, man hr/unit	*	9.7	7.9
Electric energy, kwh/unit	[a]	[a]	[a]
Principal raw materials cd of wood/short ton			
Yellow pine southern	0.21	0.37	0.55
Spruce domestic and imported	0.64	0.48	0.35
Hemlock domestic and imported	0.27	0.26	0.32
Other woods slabs and mill waste	0.45	0.43	0.33
Total accounted for	1.57	1.54	1.55

[a] No electric-energy data available for pulp mills as distinguished from paper and paperboard mills.

Table 1-9 Chemicals and Allied Products

Industry Basic unit	FERTILIZER Short Tons of Fertilizer (All Types)		
Year	1929	1933	1939
Units produced	9,320,000	5,196,000	10,161,606
Labor man hr/unit	*	5.23	3.43
Electric energy kwh/unit	11.0†	*	12.9
Principal raw materials short tons/short ton			
Superphosphates	0.32	*	0.28[a]
Phosphate rock	0.25	*	0.20
Sulfuric acid	0.25	*	0.185[b]
Sulfate of ammonia	0.049	*	0.033
Muriate of potash	0.024	*	0.049[c]
Organic nitrogen-containing materials	0.053	*	0.05
Inorganic nitrogen-containing materials	0.039	*	0.036
Inorganic potash containing materials	0.069	*	0.017
Bones	0.005	*	0.002
Other	0.15	*	0.026
Phosphoric acid	*	*	0.012[d]
Total accounted for	1.209	*	0.890

[a] Basis 16 per cent APA [b] Basis 50 Bé Basis 50 per cent K_2O [d] Basis 50 per cent H_3PO_4

Chemical Basic unit	AMMONIA ANHYDROUS[a] Short Ton	
Year	1939	1944[b]
Units produced	113,610	
Labor, man-hr/unit	*	About 21.8
Electric energy kwh/unit		
Other basic costs per unit of production		
Water, gal		81,000
Coke, tons		1.78
Coal, steam and power		2.8

[a] Production figures from 1939 government census Basic-cost data from "Chemical and Metallurgical Flowsheets" 1944 4th ed

[b] Basic costs per ton liquefied NH_3 from Nitrogen Engineering Corporation System.

* Data unavailable from sources checked † Author s estimate

BASIC INDUSTRIAL COSTS

Table 1-9. Chemicals and Allied Products (*Cont*)

Chemical Basic unit	CAUSTIC SODA[a] Short Ton			
Year	1939[b]	1939[c]	1944[d]	1944[e]
Units produced	426 250	523 907	*	*
Labor, man-hr/unit	*	*	18 0	0 9
Electric energy kwh/unit	*	*	2 500	18
Other basic costs per unit of production	*	*		
Steam, lb			20 000	2 700
Water, lb			*	18 000
Refrigeration tons			0 9	
Reburning fuel, Btu				13 000 000
Soda ash (58 per cent) lb			50	2 900
Salt, tons			1 6	
Lime make-up as 90 per cent CaO lb				160
Sulfuric acid (66° Bé) lb			200	

[a] Production figures from 1939 government census Basic cost data from Chemical and Metallurgical Flowsheets" 1944 4th ed

[b] Electrolytic process

[c] Lime-soda process

[d] Basic costs per ton 76 per cent NaOH plus 1 750 lb chlorine and 8 750 cu ft hydrogen (50 lb hydrogen) (electrolytic process)

[e] Basic costs per ton NaOH in 11 per cent solution produced in plant of 100 tons per day capacity (lime-soda process)

Chemical Basic unit	HYDROCHLORIC ACID[a] Short Tons of 100 Per Cent Acid (HCl)		
Year	1939	1944[b]	1944[c]
Units produced	140 338	*	*
Labor man-hr/unit	*	16 5	9 9
Electric energy kwh/unit	*	280	5 6
Other basic costs per unit of production	*		
Water, gal		9 012	1 550 3 100
Salt, lb		3 263	
Sulfuric acid, 100 per cent lb or		2 937[d]	
Niter cake, lb		8 173[e]	
Coal, lb		2 300	
Chlorine, lb			1 920
Hydrogen			60

[a] Production figures from 1939 government census Basic cost data from Chemical and Metallurgical Flowsheets' 1944 4th ed

[b] Salt process [c] Synthetic process [d] Process also produces 3,915 lb salt cake

[e] Process also produces 8 825 lb salt cake

Chemical Basic unit	NITRIC ACID[a] Short Tons of 100 Per Cent Acid (HNO_3)	
Year	1939	1944
Units produced	167 7[illegible]0	
Labor man-hr/unit		1 0
Electric energy, kwh/unit		270
Other basic costs per unit of production		
Anhydrous ammonia lb		575
Platinum-catalyst loss (some plants) †		0 012

[a] Production figures from 1939 government census. Basic-cost data from Chemical and Metallurgical Flowsheets," 1944, 4th ed

* Data unavailable from sources checked † Author's estimate.

BASIC INDUSTRIAL COSTS

Table 1-9 Chemicals and Allied Products (*Cont*)

Chemical Basic unit	SODA ASH[a] Short Tons		
Year	1939[b]	1939[c]	1944[d]
Units produced	2 012 804	132 897	*
Labor direct	*	*	2 0
Repair and maintenance	*	*	2 6
Electric energy kwh/unit	*	*	*
Other basic costs per unit of production	*	*	
Cooling water gal			15 000–18,000
Salt tons			1 50–1 75
Limestone tons			1 20–1 35
Coke tons			0 095–0 11
Coal boilers tons			0 25–0 5
Dryers tons			0 16–0 25
Ammonia loss as $(NH_4)_2SO_4$ kg			2–4
Carbon dioxide cu ft			1 000–1 200
Sodium sulfide lb			2 4

[a] Production figures from 1939 government census Basic cost data from Chemical and Metallurgical Flowsheets 1944 4th ed

[b] Ammonia soda process [c] Natural and electrolytic

[d] Figures per ton 58 per cent soda ash by ammonia soda process

Chemical Basic unit	SULFURIC ACID[a] Short Tons of 50°Bé Acid (62 6 Per Cent H_2SO_4)			
Year	1939[b]	1939[c]	1944[bb]	1944[cc]
Units produced	4 300 480	3 411 007	*	*
Labor man hr/unit	*	*	0 4	0 69
Electric energy kwh/unit	*	*	3 1	9 4
Other basic costs per unit of production	*	*		
Water gal			2 500	1 565
Steam lb			125[d]	
Sulfur lb			431	424
Niter (as anhydrous ammonia) lb				3 13

[a] Production figures from 1939 government census Basic cost data from Chemical and Metallurgical Flowsheets 1944 4th ed

[b] Contact process [bb] Contact process 50 tons per day capacity plant

[c] Chamber process [cc] Chamber process 50 tons per day capacity plant

[d] Contact process also produces up to 1 250 lb steam per short ton 50° Bé acid

Industry Basic unit	RAYON AND ALLIED PRODUCTS Pounds of Rayon and Related Products[a]		
Year	1929	1933	1939
Units produced	123 612 000†	245 522 700†	416 874 129†
Labor man hr/unit	0 83	0 39	0 23
Electric energy kwh/unit	2 97†	*	3 00
Principal raw materials lb/lb			
Cotton linters	0 49	*	0 26
Wood pulp	0 78	*	0 94
Total accounted for	1 27	*	1 20

[a] Yarn staple fiber transparent sheets etc

* Data unavailable from sources checked † Author s estimate

BASIC INDUSTRIAL COSTS

Table 1-10 Products of Petroleum and Coal

Industry	BEEHIVE COKE	
Basic unit	Net Ton	
Year	1937	1939
Units produced	3 164 721	1 444 328
Labor man hr/unit	0.84	0.67
Electric energy kwh/unit	*	3.29

Industry	PETROLEUM REFINING		
Basic unit	Gallon		
Year	1929	1933	1939
Units produced $\times 10^6$	40 546[a]	34 278[a]	51 686[a]
Labor man-hr/unit	*	0.0040	0.0026
Electric energy kwh/unit	0.032†	*	0.054
Principal raw materials gal/gal			
Crude petroleum	1.11	*	1.02
Partially refined oils etc	0.05	*	0.05
Natural gasoline	0.04	*	0.04
Totals[b]	1.20	*	1.11
(Lb/gal)			
Soda ash	0.0009	*	0.00044
Sulfuric acid	0.073	*	0.036
Caustic soda	0.0066	*	0.0038
Fuller's earth	*	*	0.0061

[a] Gallons of gasoline and other liquid fuels, oils, tar, greases, wax, and liquefied petroleum gases

[b] Part of the excess over 1.00 represents asphalt, petroleum coke, and other refinery products

Table 1-11 Rubber Products

Industry	AUTOMOTIVE TIRES AND INNER TUBES		
Basic unit	Automotive Tire and Tube		
Year	1929	1933	1939
Units produced[a]	68 726 000	45 301 230	57 013 000
Labor man hr/unit	*	1.92	1.71
Electric energy kwh/unit	12.4†	*	18.6
Principal raw materials			
Rubber lb/average tire casing[a]	9.47	12.11	13.73
Average lb/inner tube[a]	1.97	2.04	2.34
Lb/average tire and tube			
Carbon black	2.20	*	*
Zinc oxide	1.62	*	*
Sulfur	0.63	*	*
Tire fabrics	4.07	*	*
Hose and belting duck	9.20	*	*
Other cotton fabrics	0.28	*	*
Total accounted for	29.44	14.15	16.07

[a] Data from The Rubber Manufacturers Association, Inc

* Data unavailable from sources checked † Author's estimate

BASIC INDUSTRIAL COSTS

Table 1-12 Leather and Leather Products

Industry Basic unit	FOOTWEAR EXCLUDING RUBBER[a] Pairs of Footwear				
Year	1929	1933	1939	1944	1946
Units produced $\times 10^6$	361.4	350.4	435.3	467.9	529.0
Labor man hr/unit	1.31	1.13	0.93	0.78	0.73
Electric energy kwh/unit	0.39†	*	0.35	*	*
Principal raw materials					
Leather for uppers sq ft/pair[b]	2.23	2.16	1.90	1.72	1.50
Leather for soles[c] cattle hides/pair of leather soles	0.036	0.027	0.031	0.038	0.034

[a] All data except electric energy from National Shoe Manufacturers Association

[b] Calculated from following equivalents 1 avg cattle hide yields 40 sq ft upper leather 1 avg calfskin yields 10 sq ft upper leather 1 avg kidskin yields 5 sq ft upper leather

[c] Based on 75 per cent of shoes manufactured having leather soles Approximate distribution of footwear 50 per cent women's 25 per cent men's 25 per cent infants and children's Normally 95 per cent of all shoes manufactured have leather uppers but in 1944 and 1946 this figure dropped to 85 per cent 1947 will have normal figures

Table 1-13 Stone, Clay, and Glass Products

Industry Basic unit	FLAT GLASS Square Foot		
Year	1929	1933	1939
Units produced $\times 10^6$	479.7	269.3	598.0
Labor man hr/unit	*	0.83	0.51
Electric energy kwh/unit	0.[illegible]†	*	0.43
Principal raw materials lb/sq ft			
Silica sand	1.605	*	1.442
Soda ash	0.458	*	0.402
Salt cake	0.148	*	0.093
Limestone	0.361	*	0.219
Feldspar	*	*	0.033
Grinding sand	4.808	*	1.419
Lime	0.089	*	0.104
Total accounted for	7.469	*	3.712

Industry Basic unit	GLASS CONTAINER Gross (All Kinds)				
Year	1929	1933	1939	1940[a]	1941[a]
Units produced $\times 10^6$	35.7	34.8	51.5	54.5	70.8
Labor man hr/unit	*	0.98	0.91	1.09	0.96
Electric energy kwh/unit	10.66†	*	7.05	7.08	6.71
Principal raw materials lb/gross					
Silica sand	74.55†	*	57.78	54.05	55.98
Soda ash	27.38†	*	22.74	*	*
Salt cake	0.69†	*	0.38	0.13	0.12
Lime	5.81†	*	5.85	10.15	10.02
Limestone	8.22†	*	4.36	0.96	1.89
Feldspar	*	*	4.54	4.68	5.14
Total accounted for	116.65†	*	95.65	69.97	73.15

[a] Data from Glass Container Manufacturers Institute Inc

* Data unavailable from sources checked † Author's estimate

BASIC INDUSTRIAL COSTS

Table 1-13. Stone, Clay, and Glass Products (*Cont.*)

Industry Basic unit	PORTLAND CEMENT Barrels (376 Lb) Finished Cement		
Year	1929	1933	1939
Units produced	170,646,036	63,473,189	122,259,154
Labor, man-hr/unit	0.58	0.50	0.39
Electric energy, kwh/unit	16.2†	*	22.7
Principal raw materials, short tons/short ton			
Limestone and cement rock	1.35	1.35	1.35†
Clay and shale, including kaolin	0.16	0.18	0.16†
Blast-furnace slag	0.041	0.022	0.021†
Marl	0.045	0.035	0.034†
Gypsum	0.038	0.035	0.033†
Other, including iron ore	0.049	0.047	0.07†
Total accounted for	1.683	1.609	1.668†
Fuels[a] Year	1929	1933	1940
Coal, lb/bbl	132.3	130.1	119.5
Oil, gal/bbl	11.0	9.4	9.1
Natural gas, cu ft/bbl	2,008	1,655	1,561

[a] Figures shown are alternates, not additive.

Table 1-14. Iron and Steel and Their Products, Except Machinery

Industry Basic unit	IRON AND STEEL[a] Net Tons Finished Steel Products[b]			
Year	1933	1939	1943	1946
Units produced for sale	21,553,046	34,955,175	62,210,261	48,775,532
Labor, man-hr/unit	24.0	22.1	19.5	18.4
Electric energy, kwh/unit	*	315†	295	360
Principal raw materials, net tons/net ton				
Scrap	1.02†	0.82	0.76	0.72
Iron ore	1.02	1.70	1.43	1.58
Limestone	0.22	0.35	0.32	0.37
Fluxes	*	*	0.11	0.10
Coking coal	*	*	0.93†	0.99†
Coal (not coking), bituminous and anthracite	*	*	0.23	0.22
(Lb/net ton)				
Copper	*	*	*	2.3
Lead	*	*	*	0.9
Tin	*	*	*	1.5
Zinc	*	*	*	9.9
Chromium	*	*	*	3.34
Cobalt	*	*	*	0.02
Columbium	*	*	*	0.01
Molybdenum	*	*	*	0.22
Nickel	*	*	*	1.40
Titanium	*	*	*	0.08
Tungsten	*	*	*	0.09
Vanadium	*	*	*	0.03
Zirconium	*	*	*	0.04
(Gal/net ton)				
Tar and pitch	*	*	3.6	3.6
Fuel oil	*	*	34.0	24.0
(Cu ft/net ton)				
Natural gas	*	*	5,850	4,000

[a] Data from "Annual Statistical Reports," The American Iron & Steel Institute, except as noted (see also Table 1-22).

[b] Including all items listed in Table on p. 58 of above report for 1946, but not including iron-foundry products (see also diagram reproduced on p. 53 of *Steel*, Oct. 27, 1947).

* Data unavailable from sources checked. † Author's estimate.

BASIC INDUSTRIAL COSTS

Table 1-17. Machinery, except Electrical

Industry Basic unit	TRACTOR Tractors (All Types)	
Year	1937	1939
Units produced[a]	283,155	219,288
Labor, man-hr/unit	304†	280
Electric energy, kwh/unit	*	966
Principal raw materials, lb/tractor		
Pig iron		1,051
Scrap iron and steel		721
Total accounted for		1,772

[a] No data available prior to 1937.

Table 1-18. Automobiles and Automobile Equipment

Industry Basic unit	AUTOMOBILE Passenger Car				
Year	1929	1933	1939	1941	1942
Units produced	4,587,400	1,573,512	2,866,796	3,744,300	220,814
Labor, man-hr/unit	*	*	*	550†	
Principal raw materials, lb/typical passenger car, 1942 model with accessories[a]	Gross	Net		Gross	Net
Iron	696.66	549.63	Asbestos	3.45	2.60
Steel	3,385.88	2,397.81	Asphalt	30.51	25.74
Antimony	1.89	1.32	Glass in safety glass	81.21	46.57
Aluminum	3.78	3.21	Glass, other	24.75	15.24
Cadmium	0.047	0.042	Mica	0.116	0.102
Chromium	5.56	4.32	Cork	0.929	0.626
Copper	54.49	45.57	Cotton	66.80	63.13
Lead	35.64	33.88	Hair	0.627	0.583
Magnesium	0.070	0.009	Jute	18.18	16.70
Molybdenum	0.117	0.009	Leather	2.664	2.447
Nickel	1.85	1.56	Paper products	61.27	50.13
Silver	0.005	0.005	Rubber compound	170.58	161.38
Tin	2.88	2.40	Wood	5.19	3.94
Tungsten	0.008	0.008	Wool	8.93	8.26
Vanadium	0.0029	0.0016	Paint and thinner	81.40	26.38
Zinc	25.51	23.28	Plastics (incl. fillers)	6.75	4.91
			Sulfuric acid	5.00	5.00
			Total accounted for	4,781.69	3,496.85

[a] Original models before substitution and conservation of materials took place. Figures from Automobile Manufacturers Association.

* Data unavailable from sources checked. † Author's estimate.

BASIC INDUSTRIAL COSTS

Table 1-19 Transportation Equipment, except Automobiles

Industry	**CARS AND CAR EQUIPMENT**		
Basic unit	**Steam Railroad Electric and Self propelled and Trackless Trolley Cars**		
Year	**1929**	**1933**	**1939**
Units produced	88 305	9 680	26 777
Labor man hr/unit	*	*	1 715
Electric energy kwh/unit	1 501†	*	3 275
Principal raw materials short tons/car			
Pig iron	0 54	*	0 12
Scrap	3 01	*	4 57
Iron and steel bars	*	*	1 11
Alloy steel	*	*	0 18
Stainless steel	*	*	0 05
Structural shapes	*	*	1 99
Sheets and strip	*	*	1 11
Plates	*	*	4 86
Castings purchased as such	0 31	*	1 20
Iron and steel pipe and tubing	*	*	0 05
Steel all forms total	14 50	*	*
Copper brass bronze	0 02	*	*
(1,000 board feet/car)			
Hardwoods	0 20	*	*
Softwoods	2 16	*	*
Total accounted for (short tons/car)	18 38[a]	*	15 21

[a] Plus hardwoods and softwoods

Industry	**LOCOMOTIVE**	
Basic unit	**Railroad, Mining, and Industrial Locomotives**	
Year	1929	1939
Units produced	1 695	824
Labor man hr unit	*	14 985
Electric energy kwh/unit	47 700†	56 695
Principal raw materials short tons/locomotive		
Pig iron	14 14	10 64
Iron and steel scrap	3 98	7 41
Iron and steel bars	*	10 99
Alloy steel	*	1 57
Stainless steel	*	0 03
Structural shapes	*	2 97
Sheets and strip	*	5 83
Plates	*	20 31
Castings	*	19 80
Castings and forgings	11 52	*
Iron and steel pipe and tubing	*	2 69
Iron and steel all other forms	179 90	*
Copper brass bronze	1 93	*
Total accounted for	241 47	82 24

* Data unavailable from sources checked † Author's estimate

BASIC INDUSTRIAL COSTS

Table 1-21. Transportation

Industry	AIR TRANSPORT[a]						
Basic unit	Thousands of "Traffic Units" (Ton-miles[b] Plus Three Times Revenue Passenger-miles)						
Year	1930	1931	1933	1935	1939	1945	1946
Units produced	219,379	281,181	455,805	843,334	2,044,323	10,447,141	18,272,865
Labor, man-hr/unit	32.0†	38.6	24.4	18.0†	12.0†	10.0†	8.0†
Principal raw materials, gal/1,000 traffic units							
Gasoline	52.2	57.0	47.8	32.1	22.8	13.0	12.7
Oil	1.69	1.98	1.76	0.84	0.36	0.16	0.15

[a] Data from Air Transport Association of America.
[b] Ton = 2,000 lb (ton-miles of revenue freight).

Industry	CLASS 1 STEAM RAILROADS[a]				
Basic unit	Thousands of "Traffic Units" (Ton-miles[b] Plus Three Times Revenue Passenger-miles)				
Year	1929	1933	1939	1943	1947
Units produced	540,543,966	298,244,710	401,392,414	990,534,004	792,490,032
Labor, man-hr/unit	7.83	7.10	5.84	3.61	4.19
Other performance data					
Freight loco.-miles/loco.-day[c]	91.2	89.8	104.0	124.5	120.0
Pass. loco.-miles/loco.-day[c]	164.5	160.9	184.2	220.9	218.7
Freight car-miles/car-day[d]	34.4	24.7	36.4	51.0	48.8
Net ton-miles/serviceable frt. car-day	582	386	610	1,092	1,057
Freight cars/train	47.6	44.8	48.5	52.1	52.9
Gross tons/frt. train	1,865	1,742	1,984	2,362	2,432
Net tons/frt. train	804	699	813	1,116	1,146
Net tons/loaded frt. car	26.9	25.6	26.9	33.3	32.6
Freight train speed, mph	13.2	15.7	16.7	15.4	16.0
Gross ton-miles/frt. train hour	24,539	27,344	32,808	35,970	38,462
Fuel Consumption, lb					
Freight (per 1,000 gross ton-miles)	125	121	112	114	114
Pass. (per pass.-train car-mile)	14.9	15.2	14.8	15.0	15.9

[a] Data from Association of American Railroads (see also Table 1-22).
[b] Revenue ton-miles of freight. [c] Active locomotives. [d] Serviceable cars.

Table 1-22. Coal Mining

Industry	ANTHRACITE COAL MINING				
Basic unit	Net Ton				
Year	1933[a]	1936	1939	1942	1944
Units produced, $\times 10^3$	49.541	53.814	50,747	58,904	62,181
Labor, man-hr/unit	3.45	3.02	2.63	2.42	2.62

[a] Data for 1933 from "Minerals Yearbook"; other years from Anthracite Institute.

Industry	BITUMINOUS COAL—MINING AND MARKETS[a]						
Basic unit	Net Ton						
Year	1919	1929	1933	1936	1930	1942	1944
Units produced, $\times 10^3$	465,860	534,989	333,631	439,088	394,855	582,693	619,576
Labor, man-hr/unit	2.10	1.67	1.69	1.52	1.34	1.38	1.39
Markets							
Power, lb coal/kwh electricity	3.2	1.7	1.5	1.4	1.4	1.3	1.3
Railroads,[b] lb coal/1,000 frt. ton-miles	164	125	121	119	112	111	114
Lb coal/passenger-train car-mile	18.1	14.9	15.2	15.3	14.8	15.1	14.8
Pig iron,[c] lb coal/gross ton pig iron	3,428	2,948	2,876	2,901	2,853	2,604	2,618

[a] Data from National Coal Association. [b] See also Table 1-21. [c] See also Table 1-14.
* Data unavailable from sources checked. † Author's estimate.

BASIC INDUSTRIAL COSTS

Table 1-23 Utilities

Industry	ELECTRIC POWER AND LIGHT[a]								
Basic unit	Thousand Kilowatthours								
Year	1921	1925	1929	1933	1936	1939	1942	1946	1947[b]
Units produced total	37 180 123	61 451 091	92 180 273	81 740 069	109 316 033	127 641 804	185 979 476	223 177 783	255 725 173
Hydro	14 703 013	21 797 874	32 647 659	33 457 189	39 057 647	43 563 627	63 870 575	78 405 973	78 419 811
Steam	22 311 423	39 367 118	58 965 559	47 708 527	69 359 153	82 783 741	120 478 951	142 412,240	174 501 312
Internal comb	165 687	286 099	567 055	574 353	899 233	1 294 436	1 629 950	2 359 570	2 804 050
Principal raw materials									
Coal, short tons	26 603 972	35 615 365	41 827 358	28 542 854	40 085 035	44 538 529	66 256 667	72 196 730	89 543 867
Fuel oil 42-gal bbls	11 504 999	9 793 922	9 782 920	9 606 208	14 078 920	17 138 911	15 235 851	36 315 871	45 300 002
Gas thousand cu ft	21 700 882	45 471 839	112 353 453	101 984 743	154 084 268	188 876 692	235 208 023	306 941,419	373 168 761
Total coal and equiv short tons	30 435 972	40 014 365	49 039 358	35 273 854	50 144 035	57 598 037	79 075 132	93 471 419	115 560 550
Net generation by fuels[c] 1 000 kwh	22 343 460	39 442 625	59 154 239	48 170 167	69 823 024	83 627 710	121 584 578	144 555 407	177 000 000
Lb kwhr[d]	2.7	2.0	1.66	1.46	1.44	1.38	1.30	1.29	1.31

---

[a] Data from *Statistical Bulletin* 1947 published by Edison Electric Institute

[b] Preliminary figures

[c] Excludes generation by wood and waste fuels

[d] Average for all stations. It should be noted that the current power shortage (1946–1948) has made it necessary to operate all stations old and modern. As new stations come into operation and older stations are shut down this ratio will decrease appreciably below 1.29. The best plants today (1948) are producing a kilowatthour on 0.75 lb of coal and the best designs will make it for 0.7 lb. One system is now using 1 lb of coal per kwh. It has been estimated that the national average in three years will be 1.2 lb coal/kwh and in 12 years will be 1 lb coal/kwh.

The average Btu per pound of coal burned by the utilities in 1945 was 13 100. According to company reports the figure now varies from a low of 9 000 Btu/lb to something over 14 000 Btu/lb and averages perhaps closer to 12 000 than to 13 000 Btu/lb. This may be a temporary situation, which could improve in a few years.

* Data unavailable from sources checked † Author's estimate.

BASIC INDUSTRIAL COSTS

Table 1-23. Utilities (*Cont.*)

Industry	MANUFACTURED GAS[a]							
Basic unit	Thousand Cubic Feet (550-Btu Gas)							
Year	1932	1933	1936	1939	1942	1945	1946	1947
Units produced	265,138,000	244,743,000	238,553,000	241,405,000	291,718,000	358,876,000	377,132,000	398,805,000
Labor, man-hr/unit	0.63	0.67	0.65	0.60	0.44	0.32	0.32	0.31
Principal raw materials, lb/1,000 cu ft.								
Solid fuels, anthracite, bituminous, and coke[b]	84	86	86	77	74	66	62	57
Oil, gal/1,000 cu ft[b]	2.3	2.2	2.2	2.6	2.6	2.6	2.8	3.1

---

[a] Data from "Gas Facts, 1945–1946," and other publications of the American Gas Association, Bureau of Statistics. By-products are coke, tar, light oil, briquettes, etc. Between 0.65 and 0.75 lb of coke is produced per pound of coal carbonized.

[b] Average Btu content of fuels used:

Anthracite	12,500–14,000 Btu/lb
Bituminous	11,700–15,300 Btu/lb
Coke	12,300–13,000 Btu/lb
Light industrial oil	145,000 Btu/gal
Heavy industrial oil	155,000 Btu/gal

* Data unavailable from sources checked. † Author's estimate.

BASIC-COST REDUCTION

The Second World War forced belligerents and nonbelligerents alike to use up the world's basic resources at an unprecedented rate. We now face the problems created by depleted soil and drastically reduced reserves of oil, iron, copper, and other commodities on which our present civilization is based. There is also a marked, although possibly temporary, shortage of technically trained personnel, without whom it will be difficult to maintain, still less improve, standards of living in all countries. Basic-cost reduction is therefore an urgent necessity.

The first basic-cost reduction should be the elimination of waste all along the line from source to ultimate consumption or use. The toll of basic resources taken as result of carelessness and indifference is particularly inexcusable—soil erosion accelerated by poor soil management, pollution of streams, lakes, and rivers, forest fires started through carelessness, and the losses of food through spoilage and waste in the average home are typical examples. The same pattern of waste appears in many industries, in some to a lesser extent than in others. In general, unless there is (1) a continuing improvement in quality or (2) a reduction in the amount of raw materials consumed per constant unit of production, (3) a substitution of more plentiful for scarce raw materials or (4) an increase in the weight or number of useful by-products from the same weight of raw materials, the industry probably stands in need of technological improvement. The engineer can contribute to quality improvement and elimination of waste in industry and should cooperate in curtailing losses in other fields.

Not only are materials wasted, but countless man-hours are lost for a variety of reasons. Among these are poor personnel relations and failure to make the most of the individual employee's talents, training, and inclination; inefficient management, featherbedding, and disputes between labor and management; accidents and illness; failure to plan and execute the job properly the first time; and bureaucracy and pressure groups. In these matters the engineer is frequently out of his depth, although his responsibilities are no less marked here than in the field of raw-material conservation.

If only part of these losses in raw materials and man-hours were eliminated, much higher material standards of living would be achieved.

In addition to raw-material and man-hour losses, excessive use of utilities (electric energy, steam, water, etc.) frequently occurs. In fact, it is exceptional to find an industrial plant where a reduction of utilities consumed per unit of production cannot be made. Of course, it should not be forgotten that increases in utility consumption are sometimes justified, *e.g.*, an increase in electric energy to reduce physical effort and to make possible a higher output per man-hour.

The many ways in which industrial-engineering know-how and procedures can be applied to basic-cost reduction were presented by G. M. Read[*] in a paper read before the American Management Association in 1940. Read's tabulation is reproduced in Fig. 1-3 and needs no elaboration.

No discussion of cost reduction would be complete if limited to engineering methods. Among others which have proved successful are the following:[†]

1. *Quantity output,* which permits overhead expenses and the various fixed charges to be distributed over a very large number of units so that a low selling price can still allow for a reasonable profit margin.

2. *Specialization* in the manufacture of a commodity or commodities for which a plant is best fitted. Closely related is *standardization* of both operations and types of goods produced, the first leading to methods simplification and the second to elimination of unprofitable lines.

3. *Utilization of by-products* is one of the most effective means by which costs can be reduced. It sometimes leads to a new industry or industries and frequently permits the main product to be sold at a lower price.

---

[*] Now chief engineer, E. I. du Pont de Nemours & Co., Inc., Wilmington, Delaware (see American Management Association, Production Series No. 122, 1940).

[†] For a more detailed discussion of the subject, see Edmond E. Lincoln, "Applied Business Finance," 5th ed., McGraw-Hill Book Company, Inc., New York, 1941.

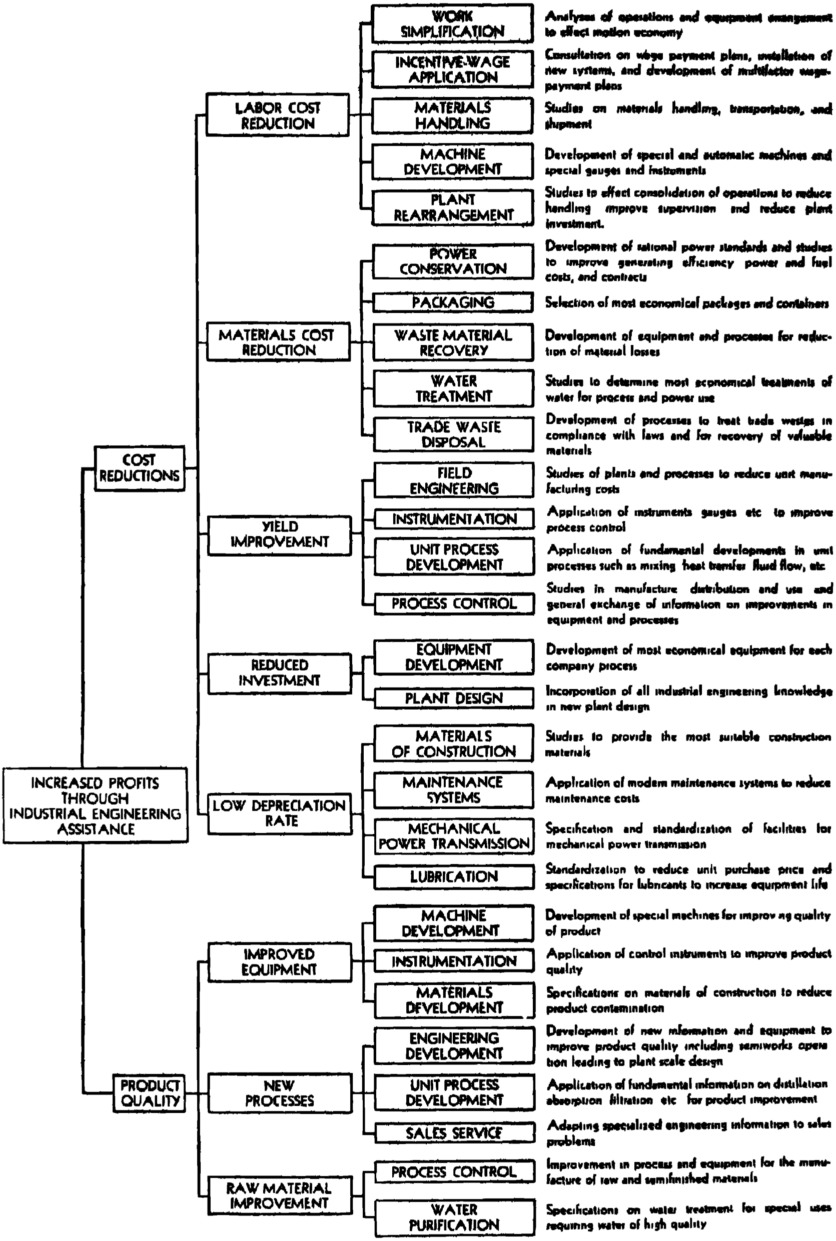
INCREASED PROFITS THROUGH INDUSTRIAL ENGINEERING ASSISTANCE
COST REDUCTIONS
LABOR COST REDUCTION
WORK SIMPLIFICATION
Analyses of operations and equipment arrangement to effect motion economy
INCENTIVE-WAGE APPLICATION
Consultation on wage payment plans, installation of new systems, and development of multifactor wage-payment plans
MATERIALS HANDLING
Studies on materials handling, transportation, and shipment
MACHINE DEVELOPMENT
Development of special and automatic machines and special gauges and instruments
PLANT REARRANGEMENT
Studies to effect consolidation of operations to reduce handling improve supervision and reduce plant investment.
MATERIALS COST REDUCTION
POWER CONSERVATION
Development of rational power standards and studies to improve generating efficiency power and fuel costs, and contracts
PACKAGING
Selection of most economical packages and containers
WASTE MATERIAL RECOVERY
Development of equipment and processes for reduction of material losses
WATER TREATMENT
Studies to determine most economical treatments of water for process and power use
TRADE WASTE DISPOSAL
Development of processes to treat trade wastes in compliance with laws and for recovery of valuable materials
YIELD IMPROVEMENT
FIELD ENGINEERING
Studies of plants and processes to reduce unit manufacturing costs
INSTRUMENTATION
Application of instruments gauges etc to improve process control
UNIT PROCESS DEVELOPMENT
Application of fundamental developments in unit processes such as mixing heat transfer fluid flow, etc
PROCESS CONTROL
Studies in manufacture distribution and use and general exchange of information on improvements in equipment and processes
REDUCED INVESTMENT
EQUIPMENT DEVELOPMENT
Development of most economical equipment for each company process
PLANT DESIGN
Incorporation of all industrial engineering knowledge in new plant design
LOW DEPRECIATION RATE
MATERIALS OF CONSTRUCTION
Studies to provide the most suitable construction materials
MAINTENANCE SYSTEMS
Application of modern maintenance systems to reduce maintenance costs
MECHANICAL POWER TRANSMISSION
Specification and standardization of facilities for mechanical power transmission
LUBRICATION
Standardization to reduce unit purchase price and specifications for lubricants to increase equipment life
PRODUCT QUALITY
IMPROVED EQUIPMENT
MACHINE DEVELOPMENT
Development of special machines for improving quality of product
INSTRUMENTATION
Application of control instruments to improve product quality
MATERIALS DEVELOPMENT
Specifications on materials of construction to reduce product contamination
NEW PROCESSES
ENGINEERING DEVELOPMENT
Development of new information and equipment to improve product quality including semiworks operation leading to plant scale design
UNIT PROCESS DEVELOPMENT
Application of fundamental information on distillation absorption filtration etc for product improvement
SALES SERVICE
Adapting specialized engineering information to sales problems
RAW MATERIAL IMPROVEMENT
PROCESS CONTROL
Improvement in process and equipment for the manufacture of raw and semifinished materials
WATER PURIFICATION
Specifications on water treatment for special uses requiring water of high quality

FIG. 1-3. Industrial engineering applied to basic-cost reduction.

4. *Integration* of industrial processes under one management can reduce production costs through control of raw materials, and selling costs through elimination of middlemen. (The benefits are not always passed along to the customer.) Integration is not an unmixed blessing, and, to name but one possibility, the increased capital* required may constitute a hazard.

5. If *stock turnover* can be increased, less working capital will be required, and unit costs can be reduced.

6. *Output per man-hour* is perhaps the most important single item in cost reduction and one in which labor's and management's interests should be parallel. Higher output per man-hour is a sound basis for increasing wages and should lead to lower costs and broader markets (see Figs. 1-1 and 1-2).

7. Closely related to (6) are *output per unit of equipment* and *elimination of bottlenecks*. The output of many a plant has been increased without additional facilities or man power, sometimes even with a reduction in these items, by bringing all units into production balance and by increasing the output per unit.

8. *Equipment utilization* has a direct bearing on unit costs. Taxes, depreciation, interest on investment, and many general and administrative expenses continue whether the equipment is fully utilized or not. The ideal is 100 per cent equipment utilization, although this cannot be attained in practice except for limited periods of time.

9. *Control of inventories* of raw materials, semifinished, and finished products is essential in a well-ordered business. Purchasing should be governed by business conditions and anticipated manufacturing requirements and inventories held at levels determined by current market demands. Speculation in inventory appreciation is hazardous and has no place in manufacturing.

Basic-cost reduction is the responsibility of all who profit from the use of the world's basic resources—consumer, labor, management, and government. Working in harmony, these four groups can raise material standards of living to new and higher levels. Working at cross purposes, it will be difficult to maintain present standards

---

* Capital in the economic sense is ordinarily thought of as the man-made instruments of production, such as mills, factories, plants, and commercial buildings, and their machines, tools, equipment, and furnishings.

SECTION 2

MANAGEMENT ENGINEERING

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  - *A.* Actual hour
  - *B.* Standard hour
  - *C.* Operating performance
  - *D.* Occupational or evaluated rate
  - *E.* Guaranteed rate
  - *F.* Pay performance
- III. Earning curves
  - *A.* Purpose
  - *B.* Classification by earning curves
  - *C.* Expected earnings
  - *D.* Guaranteed period and rate
  - *E.* Analysis by earning curves
- IV. Individual or group plan
- V. Direct or indirect workers
- VI. Characteristics of operation affecting plan
  - *A.* Unavoidable delays
  - *B.* Man versus equipment performance
  - *C.* Quality and defective work
  - *D.* Nonstandard time
  - *E.* Equipment or method change
- VII. Standards
  - *A.* Performance measurement
  - *B.* Standards development
  - *C.* Standards presentation
  - *D.* Temporary standards
- VIII. Labor participation in incentive problems
  - *A.* General labor attitude
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  - *C.* Development and installation
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- IX. References

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  - *A.* Objectives
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  - *A.* Principles
  - *B.* Problems
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  - *A.* Policies or practices
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  - *A.* Basis
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ELEMENTS OF SUPERVISION AND TOOLS OF SUPERVISION

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ELEMENTS OF SUPERVISION

Supervision. A great deal of confusion surrounds the meaning of the term "supervision." This misunderstanding can be removed quickly when it is remembered that supervision is an act. It is the act of overseeing, inspecting, or superintending. It is an act performed by a person toward other persons.

Supervisor. From the above it is seen that a supervisor is one who oversees, inspects, or superintends the actions or results of others. The supervisor is a person who is responsible for the behavior, conduct, and activity of others. He, therefore, is a leader and gets results through others.

Many men who are responsible for materials, methods, or money consider themselves supervisors. In the true sense of the word, these men are not supervisors because they do not have responsibility for men. Since a supervisor is responsible for men, it is essential that the supervisor's duties, as well as his responsibilities, be clearly defined. It is necessary for the supervisor to know how to manage and guide the man power under his direction and control to achieve the most effective and least expensive results.

Authority. Authority is the legal or rightful power to command or to act. It is the power exercised by the supervisor by virtue of his office to supervise. This power is received by him from higher authority or management. In the case of a manager, such power is received from the owner of the business. Authority, being the power to act or to achieve a certain goal, is energy in the hands of a supervisor. It is therefore the supervisor's duty to direct this energy or authority into the proper channels to achieve the goal which he is responsible for reaching.

Responsibility. Another misunderstood term in industry is "responsibility." Since it comes into being at the time when the power to act is given a supervisor, it is often confused with the power to act.

Responsibility is not a power. It is the *state of being accountable or answerable for* the trust accorded one and which one is obligated to discharge.

Duty. A duty is that act which one is obliged or bound to do. It is plain to see, therefore, that one's responsibilities are very similar to one's duties.

Upon receipt of authority or power, a supervisor becomes accountable for this power vested in him. The act of transfer of power also becomes the act of creation of accountability for the power. A supervisor must understand this principle clearly, and it will be developed at length.

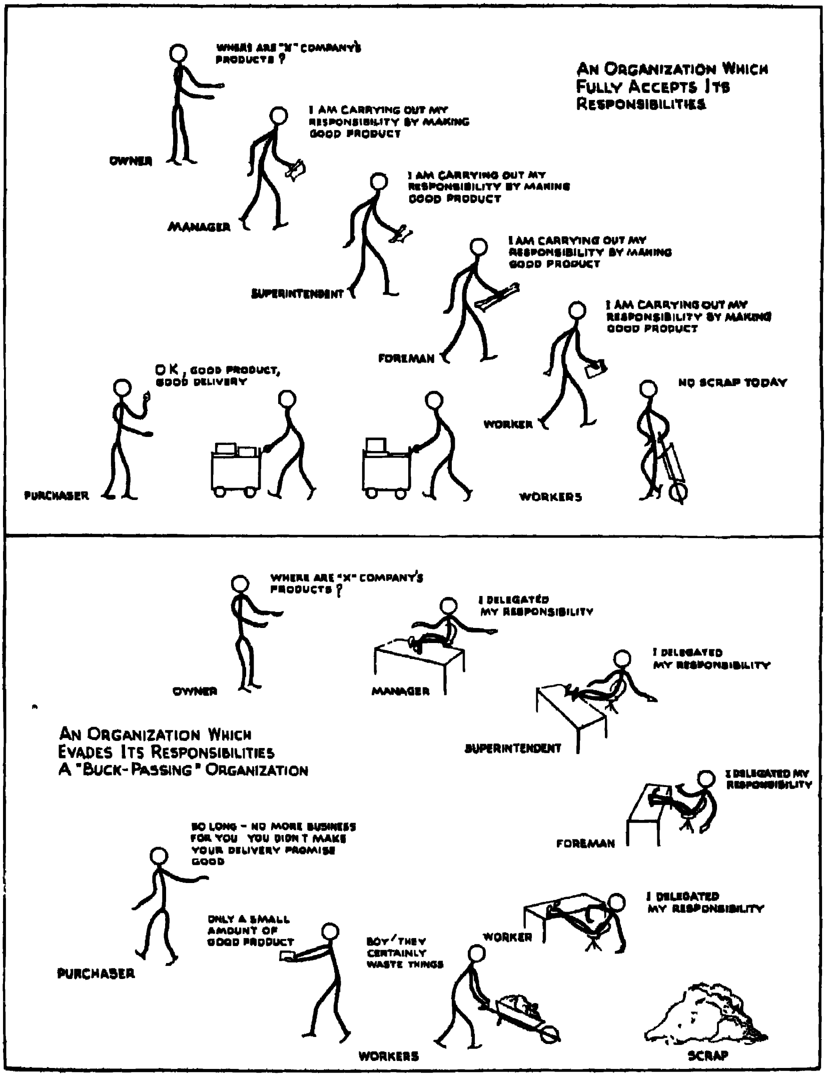
However, the important difference between duty and responsibility is that duty is an act which one is accountable for the performance of, or accountable to perform, whereas responsibility is the state of accountability.

Supervisor's Fundamental Duties. The supervisor's fundamental duties are organizing, deputizing, and supervising.

Organizing. Organizing is the act of planning work in order to achieve the desired goal, using the authority or power vested in the supervisor to achieve this end.

Deputizing. Deputizing is the act of using the best efforts of persons under a supervisor's control by giving to them a particular share of the power or authority vested in the supervisor and seeing to it that these persons use this power to their best knowledge and ability in assisting the supervisor to reach the goal he is responsible for attaining.

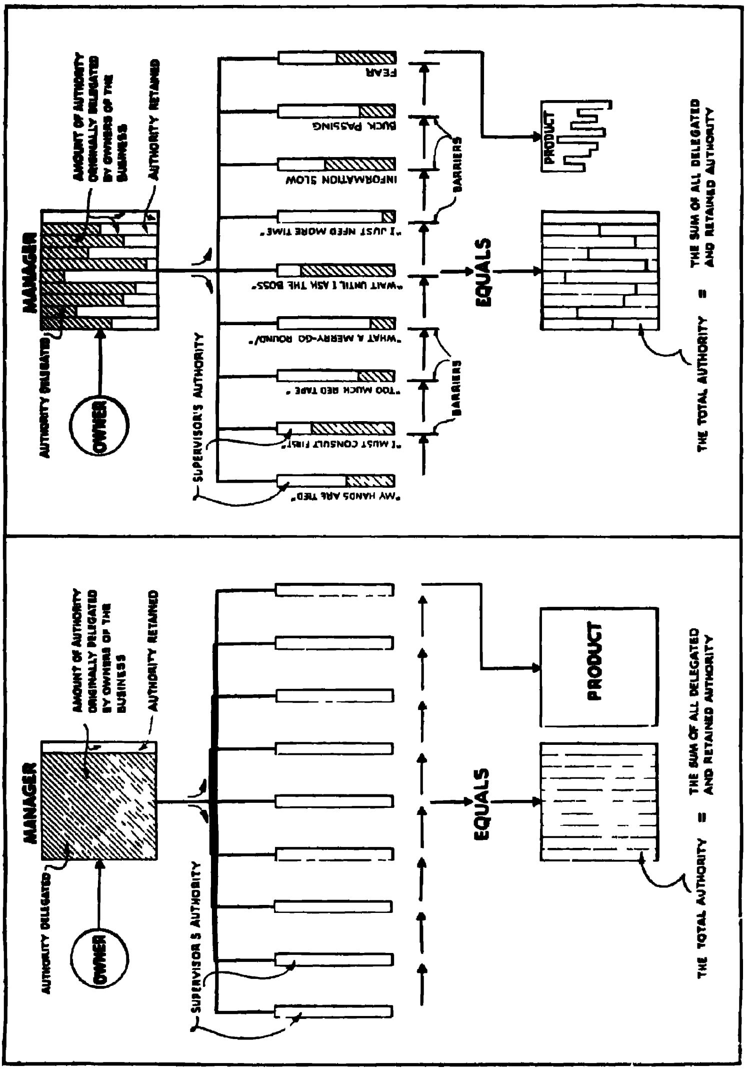
Supervising. Supervising has been defined under *supervision*


WHERE ARE "X" COMPANY'S PRODUCTS ?
OWNER
AN ORGANIZATION WHICH FULLY ACCEPTS ITS RESPONSIBILITIES
I AM CARRYING OUT MY RESPONSIBILITY BY MAKING GOOD PRODUCT
MANAGER
I AM CARRYING OUT MY RESPONSIBILITY BY MAKING GOOD PRODUCT
SUPERINTENDENT
I AM CARRYING OUT MY RESPONSIBILITY BY MAKING GOOD PRODUCT
FOREMAN
I AM CARRYING OUT MY RESPONSIBILITY BY MAKING GOOD PRODUCT
WORKER
O K, GOOD PRODUCT, GOOD DELIVERY
PURCHASER
NO SCRAP TODAY
WORKERS
WHERE ARE "X" COMPANY'S PRODUCTS ?
OWNER
I DELEGATED MY RESPONSIBILITY
MANAGER
I DELEGATED MY RESPONSIBILITY
SUPERINTENDENT
AN ORGANIZATION WHICH EVADES ITS RESPONSIBILITIES A "BUCK-PASSING" ORGANIZATION
I DELEGATED MY RESPONSIBILITY
FOREMAN
SO LONG - NO MORE BUSINESS FOR YOU YOU DIDN'T MAKE YOUR DELIVERY PROMISE GOOD
PURCHASER
ONLY A SMALL AMOUNT OF GOOD PRODUCT
BOY! THEY CERTAINLY WASTE THINGS
I DELEGATED MY RESPONSIBILITY
WORKER
WORKERS
SCRAP

Figs 2-1 and 2-2. (*Above*) An organization which fully accepts its responsibilities (*Below*) An organization which evades its responsibilities—a buck-passing organization.

Delegation of Authority. The act by a supervisor of deputizing other people to act for him, and the vesting of these people with a portion of the authority the supervisor has, is called the delegation of authority (see Figs. 2-3 and 2-4)

Amount of Authority a Supervisor Should Delegate. How much authority should a supervisor retain, and how much should he delegate? A supervisor should delegate all the authority it is possible to delegate in order to get work done, reserving for himself the necessary remainder of authority to oversee, check, and direct the work of others. This is a fundamental principle of good supervision (see Fig. 2-4).



Figs. 2-3 and 2-4. (*Left*) The correct delegation of authority. (*Right*) The incorrect delegation of authority.

Creation of the State of Responsibility. The delegation of authority has been confused in a good many instances by the supervisor with the act of delegating responsibility. *Responsibility cannot be deputized or delegated.*

Responsibility must be created. When authority is accepted by a supervisor and, in turn, the supervisor delegates a portion of this authority, he does not delegate responsibility or accountability. He is fully accountable at all times to the source of his authority for his use of this delegated power. When the supervisor accepted the authority, he became fully accountable for its use and created in himself the state of responsibility for the authority received.

If it is necessary to delegate a portion of the original authority to get results, it becomes necessary, during the act of delegation, for the person accepting the authority to become responsible. By this act of becoming accountable, additional responsibility is created. Since the superior, who delegated the authority to the supervisor, is still fully accountable, it is plain to see that there is now more responsibility, when both men's responsibilities are added together, than the sum total of the authorities of the two men (see Fig. 2-5).

It is rather interesting to study the effect of this process of creation of responsibility. If we start off with a definite amount of authority, it can usually be measured in terms of dollars. There are a definite number of dollars' worth of authority to perform a certain task, a portion of which is parceled out from top management to each succeeding echelon in the supervisory line. Since the person who originally held and delegated the authority is fully accountable and since each person under him is fully accountable for the portion of the authority accepted by him, we find that the sum total of authority adds up to the full authority originally vested in the first person, but that the responsibility adds up to a far greater amount than the original responsibility (see Figs. 2-5 and 2-6).

Failure to understand this process of creation of responsibility has been at the bottom of a great deal of trouble for supervision, which has been called, for want of a better name, "buck-passing" supervision (see Fig. 2-2).

Organization. In order that a supervisor (1) use his authority, (2) discharge his duties, and (3) accept his responsibilities, he must have a common-sense series of methods, accepted by all concerned, when he performs these acts. The sum total of these methods and the efforts of people using them to reach the desired goal is called an "organization." It can readily be seen that an organization is the product or end result of a series of acts of organization. An organization is, as its name implies, an organism or structure that has been put together to accomplish a single or a series of continuing results. This structure has certain definite characteristics which are centered around the end or goal for which the structure or organization was constructed. This structure must be so clearly defined that each unit component can work in the structure with definite regard for

1. The flow of authority or power
2. The definite acceptance of responsibility
3. The effective performance of duties
4. Close regard for relations of these unit components

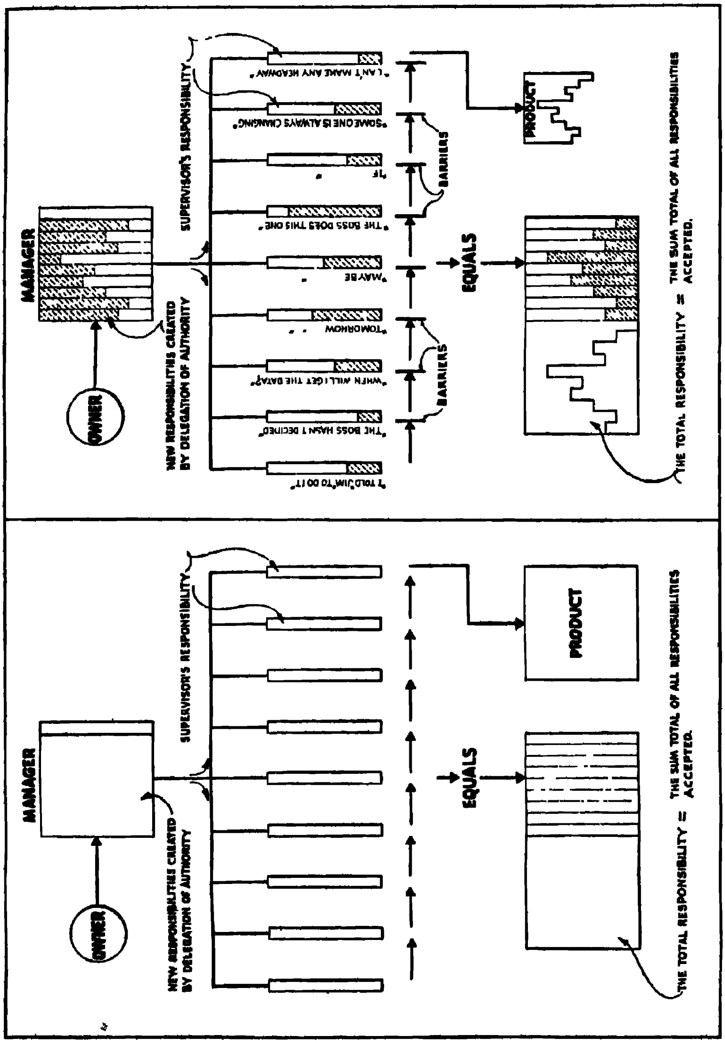
These unit components may be individuals or groups of people. The supervisor is an important unit component in this structure or organization.

Act of Organizing. In the process of structuralizing a business we have the act of organizing. It is necessary for supervision to segregate all duties of a similar nature and so combine these duties, or, as they are sometimes called, functions, to get the most effective effort from the people who are performing the duties.

In order to perform these duties effectively, the functions of various people must be clearly defined. In addition, where two or more persons are involved, relationships between these persons must also be clearly defined. In the process of definition, the supervisor will also have to determine the path and flow of authority to achieve the desired result. This path will be the path along which responsibilities will be created, accepted, and discharged. It will be, in addition, the path of duties which must be performed.

Development of an Organization. What constitutes an organization, and how can one be developed? An organization, being an operating entity with functionalized parts and with definite flow, can be compared in a manner of speaking with a mechanism. Each function or duty must be related to other functions or duties in order to have a smooth-running production machine. There cannot be rivalry or conflict between the functions of the mechanism, or the energy employed will be dissipated by other things than those for which it was originally to be used. Therefore, it is necessary to set down on paper, in an adequate manner, the fundamental channels of authority and methods by which these channels will be used for the flow and use of authority.

In so doing, it will be found that certain functions can be well defined and their use definitely prescribed. It will be found, also, that methods for checking the use


OWNER
MANAGER
NEW RESPONSIBILITIES CREATED BY DELEGATION OF AUTHORITY
SUPERVISOR'S RESPONSIBILITY
EQUALS
PRODUCT
THE TOTAL RESPONSIBILITY = THE SUM TOTAL OF ALL RESPONSIBILITIES ACCEPTED.
OWNER
MANAGER
NEW RESPONSIBILITIES CREATED BY DELEGATION OF AUTHORITY
SUPERVISOR'S RESPONSIBILITY
"I CAN'T MAKE ANY HEADWAY"
"SOMEONE IS ALWAYS CHANGING"
"IF"
"THE BOSS DOES THIS ONE"
"MAYBE"
"TOMORROW"
"WHEN WILL I GET THE DATA?"
"THE BOSS HASN'T DECIDED"
"I TOLD 'HIM' TO DO IT"
BARRIERS
BARRIERS
EQUALS
PRODUCT
THE TOTAL RESPONSIBILITY = THE SUM TOTAL OF ALL RESPONSIBILITIES ACCEPTED.

Figs. 2-5 and 2-6. (*Left*) Effective creation of responsibility. (*Right*) Failure adequately to create responsibility, resulting in only partial creation of same; insufficient to do a good job.

of these functions can also be laid down in detail. We call the first operation the setting up of a system by which the work can function and call the second operation the method of controlling the system through records.

System. System is nothing more nor less than an orderly habitual or routine method or methods by which the regular activities of a business or part of a business are carried on.

Control. Control is the process by which comparisons are made between the plan and the performance either during or after execution. Control relies on effective, complete planning and accurate, quantitative observation. It is the process of comparing these two operations.

In addition, it will be found necessary to have certain "do's" and "don'ts" clearly defined for the human beings who are going to use the structure we call an organization to obtain the desired ends of the business and the desired personal ends of the people employed in the business. These "do's" and "don'ts" are called policies, rules, and regulations.

Policies. Policies are broad directions of an authoritative nature laid down for the purpose of enabling all administrative and managerial decisions of the business to be properly determined and adequately carried out in the successful attainment of the goal or goals for which the business was organized.

Rules. Rules are standards or guides for performing specific operations or limiting the activities of people.

Regulations. Regulations are rules that are concerned with methods of activity or performance. Both rules and regulations are derived from the broader policies of a business.

The effective handling of an organization structure with its human participants calls for the exercise of the most important quality of supervision—that of leadership.

Leader and Leadership Defined. Leadership is a peculiar thing. A leader is a person who goes forth to guide or show the way: He precedes and directs. A supervisor, therefore, must show the way to the people he directs in order to be a leader. It might be interesting to know that leadership—the word "leader" or the word "lead"—has a very ancient meaning and comes from an Anglo-Saxon word that signifies "carrying the load." The leader carries the load. By this means he is able to attract, to direct, or to motivate, by his influence, the people under his control.

It is plain to see from the above that a supervisor must have a definite method or system of doing business because if he would carry the full load, instead of placing a load upon the shoulders of each individual under his jurisdiction, he would not only fail to supervise but would damage himself in the process. For this reason, supervision, when successful in delegating the necessary authority, operates on a rather interesting principle—sometimes known as the "exception principle."

Exception Principle. The exception principle is very simple. If a supervisor handles only the problems that are outside the scope of the authority he has delegated to others, he then handles problems within the scope of the residue of authority he has kept for himself. These problems are the exception rather than the rule for those who have a portion of the original supervisory authority. When this "exception principle" is adhered to, persons operating with delegated authority feel that they have more freedom of action and usually will be very much more willing fully to accept their responsibilities and discharge their duties. This fact explains why the "exception rule" is such a successful supervisory rule. Organizations where the "exception rule" is not used intelligently are usually in trouble.

Records as Supervisory and Control Tools. In the process of assisting the flow of authority, it is sometimes necessary to rely on written records instead of attempting to keep a multitude of details in one's head. For this purpose various forms and records have been established. However, these are merely supervisory and control tools and should be used as such and not as ends in themselves.

Results of Failure to Define Responsibility. Since responsibility is the state of being accountable to higher authority for certain acts or performance, it is necessary that such responsibility be clearly defined. Responsibility, poorly defined and hazy in the minds of the people concerned, is apt to create situations where people have been held accountable for acts of others over whom they have no control. When this is true, we have supervision, if it is held accountable for such acts, frustrated and out of sorts. It is necessary, therefore, that responsibility be very clearly defined. Once accepted by a person, that person should be held fully accountable until he is formally relieved of responsibility. Many industrial difficulties today are due primarily to (see Figs. 2-5 and 2-6)

1. The failure to recognize the fact that responsibilities are not clearly defined
2. The failure of people to accept their responsibilities fully in connection with the authority vested in them
3. People being held responsible without the authority or power to get results

Coordination. Coordination is the act of synchronizing various functions and duties to obtain a desired result. It is easy to see that, if methods and efforts are not synchronized, people act at cross-purposes with each other and desired goals cannot be reached. To prevent such failures of methods and to ensure that all efforts shall be in the same direction, we speak of coordinating these efforts or methods; *e.g.*, the process of delegation of authority, or delineation of duty in an organization, is a coordinating act.

Examples and Diagrams of Delegation of Authority and Creation of Responsibility. Authority was defined as power over people. It is finite in nature. Its delegation can be well illustrated by an example and by Figs. 2-3 and 2-4.

EXAMPLE: If the owners of concern A, which has 100 employees and 10 supervisors, wish to make $1,000,000 worth of product, the manager of the firm is vested with the power to spend this money in the performance of the necessary operations which convert this $1,000,000 into $1,000,000 worth of product. The manager delegates the necessary authority or power to each of his 10 subordinates to do the job effectively, retaining enough authority to supervise and control the operation. However, the manager is still fully responsible to the owners for the $1,000,000. Let us assume that he delegates to each of his subordinates $90,000. They accept the authority and become responsible for its use. They then proceed with the necessary tools, equipment, labor, and material, each to do his function in the manufacture of the finished product. It is interesting to note that, in this very simple example of manufacture of $1,000,000 worth of product, $1,900,000 worth of responsibility was created if each supervisor fully accepted his responsibility. This $1,900,000 worth of responsibility will be discharged fully only when each person has fully and successfully discharged his component responsibility, and not before (see Figs. 2-5 and 2-6).

Nature of Industrial Authority. Since each supervisor has power over men, methods, materials, and money limited in amount and kind, he must not mistake the strength of his power or its type. He is empowered to use this authority as effectively, justly, and conscientiously as he knows how in the prosecution of the business. By no stretch of the imagination is he empowered to arouse emotionally or frustrate people in the exercise of this power.

Abuse of Authority. When industrial supervision uses industrial authority in directions other than those in which it should be rightfully used, *viz.*, to get results through people without disturbing them or causing them to give more than can justly be expected from human beings for the compensation received by them, the effects of such abuse of authority are invariably bad for the individual, the supervisor, and the business. Conflicts, frustrations, demeaning attitudes, humiliation, and retaliation are potentially or actually present in employees where supervisors have incorrectly and unjustly abused their authority. Supervisors should never use their authority in a way that causes employees under their control to have such feelings aroused because employees with these feelings and attitudes are invariably less effective in the performance of their work.

The ineffective and unjust use of power denies the fundamental rights of the individual and defeats the aims of a business.

Functionalization. It is necessary to divide large groups into smaller functional groups to use most effectively the single or, at the most, the relatively few skills possessed by each person. The efforts of these groups of people must be directed by others—called supervisors—because few people are capable of performing every act in any business today. The operation that divides the structure of a business into component parts, with the units of each part having specific relationships to each other, is called the act of "functionalizing." The organization is the sum total of all these small functional segments properly related to each other.

Line Organization. In order most effectively to examine these segregated functions, assume a one-man business. John Doe, the owner, must procure raw material, machinery, and a space in which to work; he must manufacture his product, store it, and sell it, granting credit or depositing the cash in the bank. He continuously repeats this cycle of operations. Assume that John Doe's business grows to the point where he feels it necessary to add one or more other persons. He has the option of looking for another person like himself, who will perform every one of the duties he has been performing, or he has the option of looking for a person who can perform one or more of these duties. John Doe knows that it will be much more difficult to obtain a person with capabilities similar to his than it would be to take a person who can be trained to do one of these duties. John Doe finally decides to employ a person who does the manufacturing operation. As the business grows it becomes necessary to employ a number of such people to manufacture the product. It also becomes necessary for John Doe to employ another person to help sell the product. John Doe is so busy now that he finds it necessary to add a bookkeeper and a stenographer. In all this process of growth, the owner has exercised authority. The authority emanates from him and proceeds down the line to those people making his product and to the salesmen selling it. There is a clearly defined line of authority from John Doe, the owner. This is a typical example of a business operating on a "line" or "line-of-authority" basis. As this business grows, the owner would find it exceedingly burdensome, and beyond even the scope of his varied abilities, to take care of all details. He eventually would have to employ a supervisor who would look after the manufacturing operations. This supervisor would be a "line" supervisor, being in the direct line of authority to manufacture the product. John Doe would also employ a sales supervisor. This supervisor would be a "line" supervisor because his efforts are needed to distribute the goods.

Line-and-Staff Organization. Now assume that competition overtakes John Doe and that he must modify or improve his product. It becomes necessary to employ the services of experts who would work not in the line of authority, but auxiliary to it. These experts would advise the "line" supervision of the changes necessary to meet the competition. These specialists might be engineers, advertising men, packaging experts, etc. They are called "staff men." It is easy to see that, should they increase in numbers, a "staff" organization would develop. John Doe's organization has changed from a "straight-line" organization to a "line-and-staff" organization.

Most businesses today are line and staff in their organizational concept, with the amount of authority delegated to the staff functions varying within each organization. The line-and-staff organization is much more flexible than the straight-line organization defined above. There is a greater possibility of less effective direction in a line-and-staff organization than there is in a straight-line organization.

Taylor Functional Management. The line-and-staff organization is an outgrowth of the activities and research of Frederick Taylor, who was an exponent of fully functionalized organization. Taylor's concept of an organization was one where an employee had a supervisor for each function he was required to perform instead of a single "boss." Taylor's functional management finally reached the point where a man had eight people telling him

What to do
How to do it
Where to do it
And when!

Obviously, while this work of Taylor's contained the germ of a brilliant idea now used in industry as line-and-staff organization, it was unworkable because of the difficulty experienced in trying to get work done by the use of such an organization structure. People had too many bosses to work well and effectively.

Committee Management. A method of management sometimes used, but which has never shown any great promise, is that of "committee" management. Committee management is usually used where people are unwilling to delegate to any one person the necessary and requisite power to perform. By splitting this authority up and placing it in the hands of members of a committee, management is much slower and usually more dilatory than the line or line-and-staff management.

Recapitulation of Supervisor's Duties

1. The Supervisor Is an Interpreter of Basic Company Policies. It was brought out that policies add up to a system of regulative measures which are necessary for the proper conduct, direction, and progress of a business. Policies are usually broad in their scope and subject to a great deal of interpretation of meaning. The supervisor, being a person directly in charge of others, of necessity must be the one who interprets, clarifies, or makes detailed decisions (when within the scope of his authority) based on the general policy rules of his company.

It frequently happens that people otherwise skilled in the art or craft for which they are employed by the company do not have a ready command or comprehension of written policy rules. These must be explained and interpreted in a patient and friendly way to such employees.

2. The Supervisor Enforces Company Rules and Regulations. Specific rules and regulations are the outcome in some cases of policy interpretation in detail. These rules and regulations are usually a series of "do's" and "don'ts" which it is necessary for all employees to observe for the welfare, safety, well-being, and effectiveness of all employees. Part of the supervisor's duties is to see that these "do's" and "don'ts" are complied with.

Some of these rules and regulations in business involve the life or safety of employees and must be rigidly enforced. This enforcing process is called the supervisor's use of discipline.

3. The Supervisor Is a Trainer and Teacher. The supervisor finds it necessary to train the people under his control in order that they may effectively absorb the company's "know-how."

Estimates were made by people who had command of the facts that show that in the lower supervisory groups of assistant foreman, foreman, and chief foreman, training and teaching constituted 50 per cent or more of the supervisor's job, especially during the Second World War.

Many supervisors do not understand the principles of training or of learning. It is worth while to state them briefly.

What Training and Teaching are NOT:

1. Telling a man what to do is *not* teaching.
2. Showing a man what to do is *not* teaching.
3. Making a man do a job without either telling him or showing him is *not* teaching.

What Training and Teaching ARE: Teaching is a process that involves the following steps:

1. Tell a man what to do.
2. Tell him how to do it.
3. Tell him why he is doing it.
4. Show the man what to do.
5. Show the man how to do it.
6. Show the man why he is doing it.
7. Make the man tell what he is to do.
8. Make the man tell how he is to do it.
9. Make the man tell why he is doing it.
10. Have the man do the job.
11. Correct the man in a friendly way where he has performed erroneously until
12. The man is able to do the job perfectly, even though he does it very slowly and awkwardly.
13. Have the man repeat the performance of the job perfectly until he has by a sufficient number of repetitions acquired the necessary skill.
14. Check and review to make sure that all the above steps have been fully and successfully carried out.

The Learning Process. This procedure shows that there are two people involved in the teaching process—the teacher and the student. Obviously the student does not go through the teaching process but goes through another process called the "learning process."

A fundamental rule of the learning process is this: "*No one learns without doing.*"

Examination of this rule shows why practically nothing is learned when you tell a man; very little is learned when you show a man; and it is not until you allow the man to learn by a number of repeated efforts that you get any learning. For this reason, it can be seen that, while all the *above* steps in the teaching process are important, steps 10 to 13 are of paramount importance. It should be remembered that a supervisor who is a good teacher usually has well-qualified, highly skilled, and enthusiastic people under his jurisdiction, and both the supervisor and the company profit by his teaching.

4. The Supervisor Is an Inspector. The supervisor inspects the performance of the people under his jurisdiction.

This duty should not be confused with the formal process of inspection. Rather this duty is one of general oversight in order that the supervisor *may* keep himself fully acquainted with the activities of the employees under his control.

5. The Supervisor Is a Disciplinarian. This was taken up under 2. A more specific statement of the supervisor's role as a disciplinarian is necessary. The supervisor uses disciplinary force fairly and effectively—and only when necessary. *Discipline* is the process of subjecting people to control, training, or education. It is easy to see from the above that discipline does not have to be either harsh or unjust but should be fair, impartial, and moderate in determination and application.

6. The Supervisor Is an Arbitrator. The supervisor handles grievances that occur among the employees under his control and arbitrates the disputes between personnel under his jurisdiction. In some cases he also referees the settlement of differences between men under his and other supervisors' jurisdictions.

The Supervisor Handles Grievances. The handling of grievances is a very important duty of a supervisor. In the process the supervisor must be an open-minded listener until he has obtained all the facts.

There are three kinds of grievances:

a. Frequently employees who are improperly trained or instructed will have grievances due to misunderstanding. Patient and friendly explanation will usually be sufficient to eliminate grievances of this nature.

b. Impatient "brush-off" methods of handling this type of grievance will cause it to change into another, more serious, type of grievance—one where the employee's feelings are deeply involved. Grievances of this nature create a class of employees who are called "problem children." Such employees are usually disappointed, disgruntled, poorly trained, improperly indoctrinated people, with either resentful or fearful attitudes toward their supervision and toward the company for which they work. It is the supervisor's undivided responsibility to keep the number of such employees at a minimum. This can be done if a supervisor understands the fact that the behavior and speech of all supervisors are of paramount importance in the handling of every personnel situation. The supervisor must realize that employees are other human beings like himself, with cares, worries, and responsibilities similar to those which he possesses.

c. A third class of grievances stems from troubles, worries, or cares of a personal or domestic nature, which every employee has confronting him at some time or other. Worries and cares of this nature are not left at the time clock, at the gate, or taken off with the employee's coat when he goes to work. They are carried around inside him all the time he is present on the job. Frequently the supervisor can do little about this type of grievance, except to be a sympathetic and friendly listener, allowing the employee to "get it off his chest." Sometimes this affords complete relief for the employee. Other times specialized industrial agencies, other than the supervisor, may have to be employed in straightening matters out.

The supervisor should understand that people under his jurisdiction are entitled to their feelings but are circumscribed in their behavior. For example, it is better to say, "I know you feel angry, Joe, but I have full confidence in the way you will handle this matter," rather than to say, "Now don't get angry, Joe, you know that isn't the right thing to do." Employees will have more regard for supervision that shows it understands that people are entitled to their feelings. Such supervision confines its control to the employee's behavior.

A fundamental rule in the handling of all grievances by supervision is as follows: Employees should not be subjected to, or remain in, continuously frustrating situations. If the situation is caused by the business policy or procedure, such policies or procedures should be changed, if possible. If it is necessary for an employee to remain in a frustrating situation, he should fully comprehend why and be given free choice as to whether or not he wishes to continue in the situation. The basic attitude will prevent grievances due to frustration from becoming a major problem.

The Supervisor as an Arbitrator. In the handling of disputes between personnel under his jurisdiction and in the settling of differences between men under his and other supervisors' jurisdictions, a supervisor must again be a good listener and get all the facts. He must realize that "word battles" between two employees are an indication of a feeling of anger between these two employees. It is not sufficient for the supervisor to give them both a tongue lashing and make them shake hands and make up—rather the supervisor should tell both parties that he realizes they are both angry. Furthermore, he should state that they are entitled to feel angry and should then let them talk the situation out until both parties have ventilated a sufficient amount of anger to settle their differences on a logical and amicable basis. If this is done, there will be no stale emotion overhanging employees under a supervisor's jurisdiction.

In the settlement of disputes between personnel in two sections, the same strategy and tactics should be employed by both supervisors. These supervisors should agree on the methods used. Remember: Most fights have to be fought out to a finish. If this is not allowed, feelings will smoulder inside an employee and become the gathering point for other resentments, which are usually turned against a supervisor who fails to allow proper ventilation of these feelings.

7. The Supervisor Is a Planner. The supervisor plans how work, for which he is responsible, shall be done.

If the supervisor is a member of a large organization, he has the aid of a good many "staff" men. If the supervisor is in a small organization, he may have the problem of doing a great deal of planning in order to accomplish effectively what he is responsible for doing.

8. The Supervisor Is an Executive. The supervisor carries out the company's plans and basic instructions from higher management. He executes, in detail, both the company's and his plans for effective work.

9. The Supervisor Is the Friend of All Employees under His Control. The supervisor is usually the counselor and father confessor for these same employees. He sees to it that their side is well, clearly, honestly, and adequately represented to management and that management is well, clearly, honestly, competently, and fully represented to the employees under his jurisdiction.

If the supervisor feels that he knows his employees a whole lot better than the employees know themselves and, therefore, attempts to tell his employees how to live and conduct themselves both on and off the job—if the supervisor does nothing but tell the employees of their shortcomings and never recognizes their good performances; if the supervisor is arrogant and superior—rest assured that he will neither be a friend to his employees nor have a friend among them.

The supervisor who is honest, fair, just, temperate in his dealings with his people, and loyal both to his company and to the people working for him is a better supervisor than one who is arrogant, unfair, irresponsible, and always ready to shift the blame.

10. The Supervisor Is a Salesman of His Company to the Employees under His Jurisdiction. It behooves every supervisor to speak well of his company to the employees under his control. His example will undoubtedly be imitated, and his actions will be copied. In the process of selling the company to the employees, it is never necessary or wise to be either untruthful or insincere. The company should be sold to its employees on truth alone.

It is necessary for a supervisor always to realize that in a good many cases he represents the company to the people under his jurisdiction. His attitude, conduct, and demeanor are, therefore, the attitude, conduct, and demeanor of the company as far as the employee is concerned. Frequently, the management of a company is astounded to learn that the unfriendly and antagonistic attitudes of employees were created by supervision, which neglected or failed to behave as the management of the company wished it to behave.

11. The Supervisor Is a Counselor. The best counsel that a supervisor can give an employee is given by the supervisor being a good listener. If, in a 20-min interview with a supervisor, where an employee is seeking counsel, the supervisor talks for more than 4 min, he is *talking too much.* No supervisor should counsel any employee on how to govern his life—that is the prerogative of the individual. No individual should be told

What to do
Where or how he should do it

when such advice involves the employee's spare time. He should never be told

What to buy
Or whom to marry
Or how to spend his money

These are individual rights and should be respected as such. In these areas, involving the personal rights of an individual, a supervisor has no authority or right to interfere.

Improper counseling sometimes elevates the supervisor, *in his own mind,* to a position where he is superior to the average run of people. When this occurs, the supervisor has lost his effectiveness. He becomes a hazard to the business or industry with which he is connected.

12. The Supervisor Is a Student. Supervision is too prone to think that it has no more learning to do. Some people are so constituted that they will instinctively find better ways to do work. Some mental and a great many manual operations are improved by employees possessing this kind of instinctive coordinating skill. The supervisor should be open minded—looking, listening, and learning from the employees under him. It is usually true that a supervisor who is a good teacher is also a good student and willing to be shown—willing to learn.

A great deal of assistance can be rendered employees and supervisors by the use of some excellent short training courses such as

Job-instruction Training
Job-relations Training
Job-management Training

These courses were codified and published for wartime use. The material has been adapted for peacetime training of supervisory personnel. Its study by supervision will pay excellent dividends.

13. The Supervisor Is a Human Being. The only difference between supervisors and people under their control is the power or authority that is vested in the supervisor. The supervisor is the same kind of individual as those working for him.

If the supervisor fails to realize that human beings are mental, physical, and emotional creatures and that he is such a being, poor human relations will exist in the business or in the component of the business under the supervisor's direction.

If the supervisor has a proper realization of his make-up and the make-up of the people under his direction—if the supervisor conveys this understanding to those under him and insists upon the understanding of these fundamentals by those above him—he will be a successful supervisor.

TOOLS OF SUPERVISION

It is not the intent in this subsection to do more than describe and indicate the fundamental ideas behind the following supervisory tools:

1. Office methods
2. Cost-control methods
3. Quality-control methods

The literature is voluminous in these fields. It should be consulted when solving specific problems involving these supervisory tools. However, the strategic principles involved in handling such problems should be clearly understood.

Office Methods

Office methods, for a supervisor, are not ends in themselves but means to an end. They enable a supervisor to discharge his duties effectively in a systematic way. Office methods have as their foundation

1. Systems
2. Forms
3. Records
4. Procedures

System. Basically, a system is a combination of items in an orderly arrangement, according to some common or accepted rule, to make a coherent whole or unit.

Form. A form is an orderly, standardized arrangement of the area of a paper sheet. It is used to record facts systematically and precisely. When facts are recorded on a form, they can be sent from one place, or person, to another. When used this way, a form is a fact-transmitting vehicle. Forms are usually printed but do not necessarily have to be.

Record. The writing of authentic factual data accounting for, or otherwise explaining, the transactions involved in doing business is the act of recording. The resulting compilation of facts on paper is called a "record." It should be noted that recording in business is usually performed on forms, but this is not always true.

What a Supervisor Should Expect from a Record. A supervisor wishes to know what the results of the controlled actions of the people for whom he is responsible are. Therefore, records, as supervisory tools, must show a supervisor *how, when, where, why, and how well* he has performed his duties and discharged his responsibilities.

Procedure. A procedure is a detailed orderly method of passing from one step or operation to the next succeeding step or operation in any transaction involving the conduct of a business. For convenience, procedures are sometimes written, but it is not necessary that they be so recorded.

When procedures are written, they are sometimes recorded on forms in order that standardized systematic methods of recording them be used. This again is a matter of convenience which increases the effectiveness in the use of the procedure. However, it is not necessary that procedures be recorded in this manner.

Symbol. In the process of recording, and in the use of the record thereafter, it is usually found convenient and timesaving to use symbols instead of writing voluminous and wordy descriptions of the items being recorded. A symbol, therefore, is a mark, sign, character, or groups of characters representing something else. It is used as an abbreviation for that which it represents. These abbreviations may and do represent ideas, items, people, activities, etc. They may be composed of alphabetical, numerical, geometrical, pictorial, or meaningless components, or a combination of these components when representing that for which they stand.

Classification. The act of classifying is that of arranging and segregating persons, items, or ideas having similar characteristics of rank, order, time, dimension, or principle into orderly related groups.

Reason for Classification of Data. Data should be classified to save time and simplify their use.

It is essential to determine how the recording of items (called "data") can best be performed. A little thought will usually show the basis of similarity which will cause items to be grouped together naturally or normally. For example, some items will group on a geographical or location basis. Other items will group on a time basis, while still others will group on a basis of similar characteristics, such as materials of the same composition, parts of the same shape, assemblies used for the same purpose, etc. Such sequences or groups are given the more familiar names of *pay rolls, buildings, departments, job numbers, projects, lot numbers, parts, etc.* Classifying data into such groups enables a supervisor to make his activity more effective.

Method of Classifying Data. All the above looks rather formidable and complicated. The operation of classifying can be simplified by a clear realization on the part of the supervisor as to the use, as a supervisory tool, to which he wishes to put the record. All records must conform to two very simple principles. The consistent use of these principles will greatly simplify the act of classification, the art of recording, and the use of the record.

The record must tell the supervisor a story. The story can be that of a series of happenings one after the other in time, or it can be a story of a series of related happenings, items, or characteristics which have occurred or are present at a specific time.

Journal Type of Record. If items are recorded one after the other in a time sequence, it is a journal type of record.

Ledger Type of Record. If items possessing similar characteristics are all grouped together to make clear a situation as it exists *at a particular time*, it is a ledger type of record.

METHOD OF DERIVING A LEDGER-TYPE RECORD. It is clear that, if a journal type of record is made, it is easy to extract from it items possessing similar characteristics. If this process of extraction is performed at stated intervals and a new record is made of all the items possessing similar characteristics, a new record will be derived or made from the original journal type of record. This new record is the ledger type and gives a picture of the situation concerning the items involved at a particular time. If the two simple principles summarized below are understood,

1. Record the data in some journal sequence.
2. Derive ledger-type records by extraction.

If reasonable common sense is used in classifying items, records can and will tell a time-flow story if they are of the journal type.

If records are of the ledger type, they tell a story of the situation at a particular time.

EXAMPLE: Take the problem of recording a series of jobs which can be called making up "job orders." As time goes on and business is transacted, it becomes necessary to do one job after the other. Each job is written up on a job order as the necessity for the performance of the job comes to pass. Job orders, therefore, differ in age (or time).

It would be easy to take a sheet of paper, calling it the job-order journal, and, by serially numbering each job, set down in job-order-number sequence a record of all the job orders issued. This job-order journal becomes a running record in time of all jobs involved.

Now, assume that these jobs are done by different departments. By going through the job-order-journal record and extracting all jobs pertaining to departments A, B, C, etc., setting each one of these extractions up on a sheet devoted to the specific department, a ledgerized record is obtained from the original journal. However, in this case this ledgerizing or classifying segregates *only* by departments.

A further examination of these new records will show that each one is a *new departmental job journal*. If, within department A, it is desired to segregate types of jobs by machine, the ledgerizing or classifying process can again be used, and by the process of extraction, a new derived record will be obtained. It is interesting to note that the ledgerizing process can and may create new journalized types of records.

SUMMARY OF STEPS IN DERIVING LEDGER-TYPE RECORDS.

1. Record in journal sequence.
2. Classify.
3. Extract.
4. Rerecord the new data, using the classifying characteristic on which the extraction was made.
5. Repeat steps 2, 3, and 4 if necessary.

Failure on the part of the person constructing records clearly to understand and perform these fundamental operations is at the bottom of most record trouble. Records that are not constructed in accordance with these principles are usually ineffective when interested parties wish to use them.

The Dynamics of Recording. In addition to the strategy of classifying described above, it is necessary to understand clearly that records, and the symbols contained thereon, must stand in a definite relationship to the items they represent. Such records must indicate all (or at least the essential) activities concerning the recorded items. In any operation or series of operations, materials that eventually comprise components of the product must have a certain *flow* velocity. The principle of industrial flow, of which recorded data are the symbol, will now be described.

1. *The Flow of Authority.* Authority or power to produce results must flow through the necessary recognized channels in order that the results may be obtained. This is amplified under Elements of Supervision.

2. *The Flow of Decision.* In the exercise of industrial authority, it becomes necessary to decide

1. What to do
2. Where to do it
3. How it is to be done
4. When it is to be done
5. Who is to do it
6. In what quantities product is to be made

The decision involving the above items will pass from one person to another when these persons are involved in the various operations. These decisions are the visible and audible exercise of industrial authority. The process of passing these decisions is called the "flow of authority."

3. *The Flow of Material.* Material, as it is operated on, will flow in channels corresponding to those channels defined by the flow of decision.

4. *The Flow of Paper.* Assume that, as material flows, a paper symbol flows along a path corresponding to, and in some cases identical with, the path of material flow. This symbol, in the shape of a paper record, can be used to acquaint those who are responsible with what is happening to the material without direct examination of the material. The movement of this paper symbol or record is called the "flow of paper."

5. *The Flow of Recorded Data.* If the symbol or record described above carries the recorded data in sufficient detail and with adequate clarity, it will perform two most useful functions—that of assisting in the control of the flow of material and that of controlling the operations performed on this material during the process of flow.

Coordination of Flow of Records—How Many Records to Keep, How Detailed Should They Be. From the above, a logical conclusion can be drawn. Records and data must flow in synchronism with the flow of

1. Authority
2. Decision
3. Material
4. Personnel

A further conclusion is that there should be no more records than are necessary to control adequately and effectively the flow of these items. Furthermore, records should have no more and no less data than are necessary for the regulation and control of this same flow of authority, decision, material, and personnel.

When records are set up on the basis described above, with due regard for the fundamentals of flow, they become very effective supervisory tools. With them, supervisors can achieve the desired ends of supervision, *viz.*, making the necessary product or rendering the necessary service, by effectively controlling the people and material under their jurisdiction and completely and adequately discharging their responsibilities.

Records that do not flow correctly, as described above, rapidly lose their effectiveness.

Records which are extreme in detail, redundant in data, and which stagnate are the basis for the creation of excessive "red tape."

Cost-control Methods

Cost. A cost is a price paid. In a business, costs are

1. The prices paid during the process of converting the component materials to a finished product

2. The expenses incurred in getting this finished product into hands of consumers

Types of Costs. There are five types of costs incurred by business in the process of manufacturing goods or rendering service:

1. Capital Cost. Capital cost is the price paid for the use of the capital and goods, such as buildings, machinery, and equipment.
2. Labor Cost. Labor cost is the price paid to all people directly or indirectly responsible for the manufacture of goods or for the rendering of service to those directly or indirectly responsible for the manufacture of the goods.
3. Material Cost. Material cost is the price paid for all materials that enter into the manufacturing process, whether or not they appear in the final product.
4. Overhead Cost. Overhead cost is the expenses which, although necessary for producing the manufactured goods, cannot be charged directly to the product and which must, therefore, be allocated or distributed.
5. Sales Cost. Sales cost is the price paid for the distribution of the product and the placing of it in the hands of the consumer.

Cost Control. Cost control is the regulation of the component operations of a business in order that the prices paid by the business for the various operations or services involved in the manufacture of the product and its distribution be kept to a minimum. It can be seen that cost control involves

1. Decision
2. Guidance and direction
3. Measurement of performance
4. Effective action

Supervision's Authority and Responsibility with Respect to Cost Control. Line supervision's authority, in lower echelons, and responsibility with regard to cost control are usually limited to authority over and responsibility for

1. Labor
2. Material
3. Operations performed by labor on material

Prime Cost. The costs of labor and material, when taken together, are sometimes called the "prime cost" of manufacture. Lower line supervision is currently said to be responsible for prime cost.

Cost Records. Ample records of the various elements entering into the cost of labor, material, and the operations performed by labor on material must be kept in order that supervision exercise adequate control over the manufacturing process. Such records, called "cost records," also have an added use by becoming storehouses of performance information when new jobs or operations are in the planning stage. Such records show the difference in worth between various employees and also show such things as the amounts of materials used and the amounts of time used in individual or groups of operations.

Cost control endeavors to keep waste material and ineffective use of labor at a minimum. One specific procedure by which cost- and production-control records minimize waste is called "quality control."

By using quality-control records, supervision making production and cost-control decisions and acting on these decisions has at its command powerful new instruments for cost reduction.

Quality-control Methods

For many years, industrial operations were governed by the process of inspection. If an operation was performed and, by some measuring means, the material on which the operation was performed was measured, the measurement could and would become an index of the acceptability of the product and the operation. With the increase in complexity of manufacturing operations, the process of making sure that the product conformed to the necessary detailed specifications during each operation and met the requirements demanded of satisfactory finished product became a very expensive one when methods that involved the inspection of every part, piece, subassembly, or batch were used.

In some industries it was thought necessary for such inspection operations to be performed on every part or batch. Because of the staggering cost of such inspection, studies were made to determine the fundamental principles underlying the control of industrial operations.

The important result of such studies was the creation of a new concept of control, called "quality control," and new records used for control, called "quality-control charts."

Inspection. Inspection is the act of subjecting to careful, quantitative scrutiny parts or components of parts, batches of product, or even the finished product itself. In this scrutiny the process of measurement must be used and comparisons with predetermined standards made. If the measurement, after being compared with the standard, lies within prescribed limits, the part, component of the part, operation, or other item under inspection is said to be within the tolerance allowed and passes inspection as a satisfactory or acceptable unit.

By implication, it is seen that inspection might be construed to be an operation that would have to be performed on every part or batch going through the operation under inspection.

A further implication lies in the assumption that there is nothing further to be done should the inspection be satisfactory and all inspected parts lie within the prescribed tolerance.

If, under this type of procedure, the inspector finds parts or material that do not conform to the standards, these items become "rejects" or "defectives." However, until this situation comes to pass, nothing of a preventive nature is done, usually, to avert such industrial waste. It can be seen that even 100 per cent inspection will not ensure the production of good material or prevent the production of defective material.

Quality Control. Quality control is defined as a philosophy that aims at using mathematical, statistical, and inspection techniques to control the characteristics, attributes, dimensions, or physical or mechanical properties of materials within definite, known limits. This operation of control is performed on such a basis that preventive techniques can be used in order to

1. Lessen the cost of inspection to the lowest safe and economical amount
2. Make the maximum amount of acceptable product of the desired standard
3. Make the minimum amount of rejected product by preventing defects which cause the product characteristics to fall outside the tolerance limits

Foundation on Which Quality-control Practice Is Based. Quality control is based on a mathematical and physical foundation. This foundation can be illustrated by the following example:

> If a large number of similar objects be examined and any characteristic of these objects be measured, it will be found that these measurements of the same characteristic will differ among themselves by small amounts.
>
> In other words, if the diameters of 100 small cylinders, which were turned and cut off in an automatic screw machine, were measured with micrometer calipers and the measurements tabulated, differences between individual measurements would be found. Some measurements would occur more frequently than others. If a "tick" sheet of the dispersion of these measurements were made, it would appear similar to that shown in Fig 2-7.

Variation. The extent to which these measurement readings differ, scatter, or disperse is called the "variation" in the measurement.

Sources of Variation. 1. Some of this variation may be due to the inherent characteristics of the machine itself or to variation in the characteristics of the machine.

2. Some of the variation may be due to the activities of operator of the machine.

3. Some variation may be due to defective measurements by the inspector.

4. Other variation may be due to causes outside those named above.

The causes of variation under 2, 3, and 4 are called "assignable causes" and can usually be tracked down. When these causes are isolated, correction in operation or process can be made. By making such corrections, variation due to assignable cause is eliminated from the operation or process.

Types of Variations. There are two types of variation:

1. That which is *normal and inherent in the process, material, or machine* and to which *no assignable cause can be given*
2. That which is *abnormal and not inherent in the process* and for which *an assignable cause* may be *found*

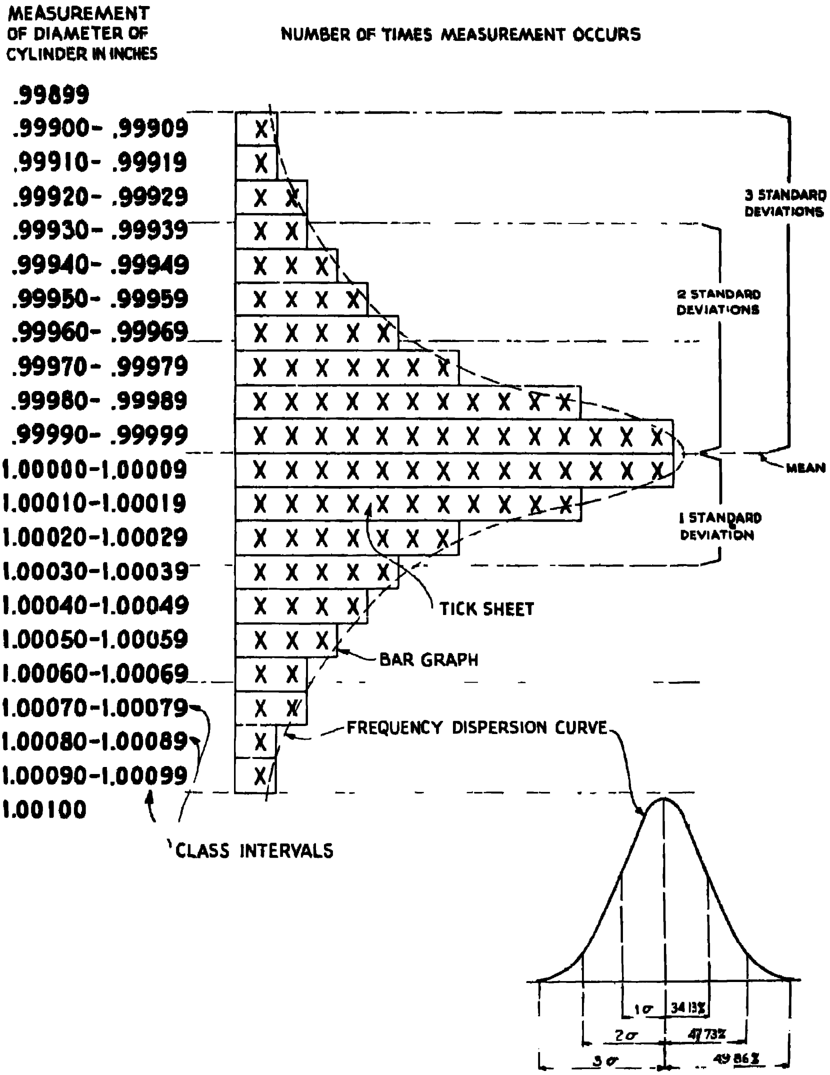


Fig. 2-7. Typical construction of "tick" sheet, bar graph, and frequency-dispersion curve.

Illustration of Sources of Variation and Their Detection. From an examination of the data that were made into a frequency-dispersion "tick" sheet in Fig. 2-7, it will be seen that all the measurements were within the prescribed tolerance. However, if these measurements are plotted in the time sequence in which

COLUMN 1 MEASURED DIAMETER IN THOUSANDTHS OF AN INCH		COLUMN 2 NUMBER OF READINGS IN EACH INTERVAL f	COLUMN 3 DEVIATIONS FROM AVERAGE VALUE $x - \bar{x}$	COLUMN 4 FREQUENCY X DEVIATION COL.2 X COL.3	COLUMN 5 FREQUENCY X DEVIATION SQUARED COL.3 X COL.3 X COL.2 σ
	.99899"				
.99900"	.99909	1	- .00095	- .00095	0000009025
.99910	.99919	1	- .00085	- .00085	0000007225
.99920	.99929	2	- .00075	- .00150	0000011250
.99930	.99939	2	- .00065	- .00130	0000008450
.99940	.99949	3	- .00055	- .00165	0000009075
.99950	.99959	4	- .00045	- .00180	0000008100
.99960	.99969	5	- .00035	- .00175	0000006125
.99970	.99979	7	- .00025	- .00155	0000004375
.99980	.99989	11	- .00015	- .00165	0000002475
.99990	.99999	14	- .00005	- .00070	0000000350
1.00000	1.00009	14	+ .00005	+ .00070	0000000350
1.00010	1.00019	11	+ .00015	+ .00165	0000002475
1.00020	1.00029	7	+ .00025	+ .00155	0000004375
1.00030	1.00039	5	+ .00035	+ .00175	0000006125
1.00040	1.00049	4	+ .00045	+ .00180	0000008100
1.00050	1.00059	3	+ .00055	+ .00165	0000009075
100060	1.00069	2	+ .00065	+ .00130	0000008450
1.00070	1.00079	2	+ .00075	+ .00150	0000011250
1.00080	1.00089	1	+ .00085	+ .00085	0000007225
1.00090	1.00099"	1	+ .00095	+ .00095	0000009025
1.00100"					
SUM		100		0	0000132900

① IN COMPUTING THE DEVIATIONS FROM THE MEAN, ASSUME ALL READINGS IN THE INTERVAL TO BE AT THE MID POINT OF THE INTERVAL.

② THE SUM OF COLUMN 4 EQUALS ZERO

③ THE SUM OF COLUMN 5 DIVIDED BY 100 EQUALS THE SQUARE OF THE STANDARD DEVIATION

④ THE STANDARD DEVIATION $= \sqrt{\frac{.0000132900}{100}} = .000349$

⑤ THIS MEANS THAT 68 4% OF THE MEASUREMENTS WILL LIE WITHIN A TOLERANCE OF ONE STANDARD DEVIATION i.e

DIAM. = 1.000"± .00035"

⑥ THIS ALSO MEANS THAT 95 46% OF THE MEASUREMENTS WILL LIE WITHIN A TOLERANCE OF TWO STANDARD DEVIATIONS i.e.

DIAM. = 1000"± .00070"

⑦ THIS ALSO MEANS THAT 99.73% OF THE MEASUREMENTS WILL LIE WITHIN A TOLERANCE OF THREE STANDARD DEVIATIONS i e.

DIAM. = 1.000"± .001"

FIG. 2-8. Typical computation of standard deviation.

each one occurred, a chart resembling Fig. 2-9*D* would be obtained. It can be seen that there is a definite trend, as each new part is made, for the measured diameter to approach the upper tolerance limit. A probable assignable cause for this variation would be the dulling of the tool (see Fig. 2-9*D*).

From a preventive point of view, it would be desirable to make the change to a newly sharpened tool before this trend had reached the point where defective parts were made. If this were the only *assignable* cause for this particular operation, it would be necessary only to inspect a few pieces at stated intervals, plotting the results and watching the trend. This method of inspection would be far more economical and would produce products lying within the specified standards at as high or even a higher degree than full piece inspection. Furthermore, the preventive attitude, inculcated in operating personnel by the use of such a chart, would cause them to make tool changes before spoilage occurred.

Average. An average is the "mean" or middle point of a series of data. It has the symbol $\bar{X}$. If, in the example given above, all the readings of the measurement of the diameter be added together and this sum be divided by the number of readings, 1.000 in. will be obtained as the average value. It is important that this average value be examined and its reliability checked by determining its standard deviation (see Fig. 2-8).

Deviation of Individual Readings. If the difference between each reading and this average value be ascertained, this difference can be called the deviation of each observed value. It has the symbol $(X - \bar{X})$. It can be seen in the example that some of these deviations are positive and others are negative. If the deviations are added up, with due regard for signs, the sum would equal zero (see Fig. 2-7).

Standard Deviation. If, however, these deviations are squared, the squares can then be added together. If this sum is divided by the number of readings and the square root of this quotient is extracted, the result is a measurement called the "standard deviation." This standard deviation has the symbol σ and is called sigma. *The value of sigma does not equal zero* (see Fig. 2-8).

WHAT THE STANDARD DEVIATION MEANS. The standard deviation of an average tells how good the average is (see Fig. 2-8).

1. An amount of one standard deviation on each side of the average, or mean, will include 68.26 per cent of all readings composing the average.
2. An amount of two standard deviations on each side of the average, or mean, will include 95.46 per cent of all readings composing the average.
3. An amount of three standard deviations on each side of an average, or mean, will include 99.73 per cent of all readings composing the average.

Effect of Arbitrarily Tightening Tolerances. Going back to the original example involving the measurement of the diameter of the 100 small cylinders, assume that somebody decided to attempt to increase the quality of these cylinders by arbitrarily tightening the tolerances from plus or minus three standard deviations (see Fig. 2-9*A*), which represent the limits of variation inherent in the operation, to plus or minus two standard deviations.

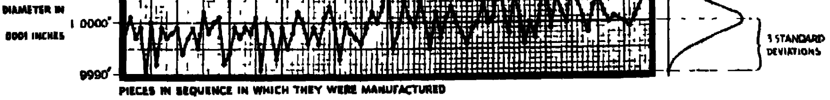


FIG. 2-9*A*. Quality-control chart—correct tolerancing of operations.

The new tolerances, positioned at the plus or minus two-sigma limits, are shown in Fig. 2-9*B*. These tolerance lines are now inside the lines of the three-sigma limits, and arbitrary spoilage now shows up.

Figure 2-9*C* shows a further arbitrary tightening of tolerances to the plus or minus one-sigma limits, with a consequent sharp increase in the number of rejected pieces.

It can be seen that, with the tolerances set too tight by arbitrary means, a needless amount of rejection will take place and that no amount of inspection will cure the variation which is inherent in the operation. This is a situation which can be corrected only by the improvement of the machinery or operations constituting the process or by changing the tolerance limits back to more realistic ones than those which were arbitrarily set. The records shown in Figs. 2-8, 2-9*A*, 2-9*B*, 2-9*C*, and 2-9*D* are called "quality-control" charts of the simplest kind. Such records enable supervision to control processes realistically. The intelligent use of these records will prevent the manufacture of defective material without recourse to 100 per cent inspection.

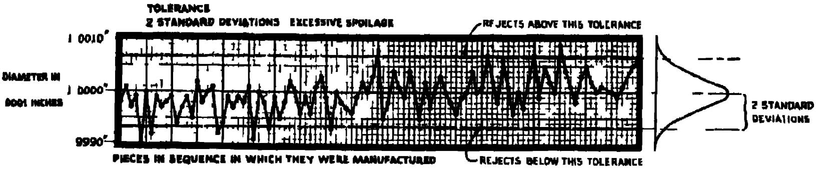


FIG. 2-9*B*. Quality-control chart—excessive spoilage due to tightening tolerances arbitrarily.

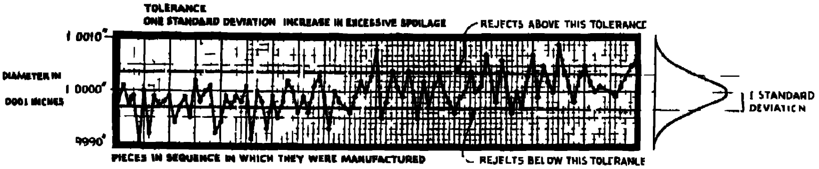


FIG. 2-9*C*. Quality-control chart—increase in spoilage by arbitrary increase in tightening of tolerances.

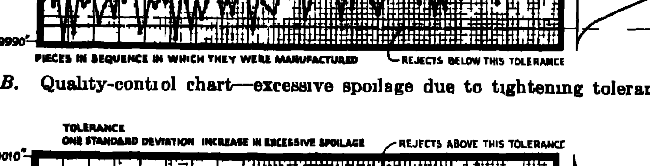


FIG. 2-9*D*. Quality-control chart—a trend of measurements due to probable assignable cause.

Mistakes Avoidable through Quality Control. The quality-control charts and quality-control philosophy enable supervision to avoid two very costly types of inspection mistakes. They are:

1. The process of looking for trouble when none exists and when the variation is the normal amount inherent in a process operation of material.
2. The failure to look for trouble, when it does exist, soon enough to prevent defective material or product being made or faulty operations being performed.

Quality-control Literature and Consultation. While the quality-control literature is scattered and highly technical, some acquaintance with this literature by supervision is extremely profitable. The advice and consultation of engineers versed in quality control or of statistical men connected with a supervisor's organization will help to clarify the more technical sections of this subject for line supervision. Consultation with these men is definitely recommended.

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WORK SIMPLIFICATION*

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Work simplification or motion study is used to analyze production and work methods so as to

1. Eliminate unnecessary steps
2. Arrange the necessary steps in the best order
3. Make each step as easy and economical as possible
4. Determine the easiest and most economical over-all pattern for the job
5. Place the method information in such a form as to facilitate its being carried into practice.

The development of a better way of doing the job, the development of a better method of production, and the development of an efficient method of doing new work are almost always the results of the application of a logical procedure consisting of the following steps:

1. Analysis of the details of the known method or methods of performing the work
2. Application of known general and specific principles of improvement
3. Formulation and detailing of the resulting suggested method of performing the work
4. Preliminary analysis of the suggested method of doing the work
5. Testing of the improved method by application and possible time measurement
6. Final application of the standard method

The main body of knowledge in work simplification consists of

1. Procedures for analyzing jobs into units possessing known characteristics
2. Check lists of desirable and undesirable features applicable to these analyses
3. Procedures for describing the new method prior to performance
4. Procedures for checking, prior to performance, the desirability of the new method
5. Procedures for setting down job methods in such a manner as to aid in training operators to follow the desired job method

Table 2-1. Possible Classes of Change through Motion Study

Class of change	Hand and body motions	Tools, workplace, and equipment	Process	Product	Raw material
1	New	Minor changes	Same	Same	Same
2	New	New (radical)	Same	Same	Same
3	New	New	New	Same	Same
4	New	New	New	Modified	Same
5	New	New	New	Modified	New

NOTE. The word "modified" appears in the product column to emphasize that this is only a modification rather than a different product for a different purpose Also in any row the item farthest to the right undergoing change indicates where the change must usually be initiated

The particular procedure used for analysis is a function of how far the analyst thinks desirable, or feels able, to carry the improvement Altogether, five classes of possible changes exist as shown in Table 2-1 Note that, as the complexity of change increases, more and more items are changed The higher the class of change, in most cases, the longer it takes to achieve, the more authority required, and the more people affected.

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* This section is abstracted from M E Mundel, "Systematic Motion and Time Study," Prentice-Hall, Inc., New York, 1947 All rights are reserved by Prentice-Hall Inc

The class of change finally sought is the main factor that determines which group of analysis procedures the procedure used will be chosen from.

The class of change sought is a function of

1. How much volume is there, or expected, and how often does the job occur?
2. How long will the job exist?
3. How much time per unit is spent on the job?
4. How much time is available for working up the change?
5. How much equipment is already invested in the job?
6. How much analysis time will be required?
7. How much loss of production would occur during a change?
8. How much retraining would be required?
9. What is the possible saving by means of the improvement?
10. What is the position of the analyst in the organization?
11. What is the nature of the organization and the personalities involved?
12. Is the product production or consumer goods? Will the change affect marketing?

The importance of each factor will vary from case to case. Their particular order as given here does not indicate their necessary relative importance. The analyst must weigh and evaluate them for each situation.

For class 1 and class 2 changes the following work-simplification techniques emphasizing man analysis are used:

1. Process chart—man analysis
2. Operation charts
3. Man and machine charts
4. Multiman charts
5. Multiman and machine charts
6. Micromotion study
7. Stop-watch time study (This last is mainly used as a means of determining expected performance, although it may be used as an aid to methods improvement.)

For class 3, 4, and 5 changes a work-simplification technique emphasizing product analysis is used. The one commonly used is

8. Process chart—product analysis

DEFINITIONS

Process Charts—Product Analysis. Process charts—product analysis—are a graphic means of portraying the separable steps of the procedure involved in performing the necessary work required to modify a product from one stage of completion to another.

Process Charts—Man Analysis. Process charts—man analysis—are a graphic means of portraying the step-by-step procedure a person goes through in doing a job and are particularly useful when the job involves moving from place to place.

Operation Charts. Operation charts are a graphic means of portraying the work a person does when performing a job that takes place essentially at one location. The operation chart usually involves a breakdown into steps of the work performed by each of the body members engaged in the task. This usually means the right and left hands. Where the feet or eyes are important factors, they may also be charted

Man and Machine Charts. Man and machine charts are a graphic means of portraying the work a person does when working with a machine or machines when the work of the machine or machines is a controlling factor.

Multiman Charts. Multiman charts are a graphic means of portraying the work of a crew, with or without machines, when the work involves coordinate work of the crew and not work in sequence.

Micromotion Study. Micromotion study is a technique that involves the use of a motion-picture camera and a timing device for the analysis of man activity. Any of the above man-analysis techniques may be used for a graphic presentation of the data, or a simo chart may be used with time as a factor and the motions broken down into therbligs.

How the man-analysis technique is selected is indicated in Table 2-2.

Table 2-2. Selecting the Analysis Technique for Man Analysis

Job characteristics	Technique recommended	Equipment required	Notes
Person moves from place to place while accomplishing job	Process chart—man analysis	Paper and pencil. Watch is sometimes used but not necessary	Rapid analysis
Person does job at one place	Operation chart	Paper and pencil. Watch is sometimes used but not necessary.	Rapid analysis
Man operates one or more machines. Machine time is an important factor	Man and machine chart	Paper, pencil, and watch are desirable	Rapid analysis
Several men work coordinately as a crew without or with machine or machines	Multiman chart and multiman and machine chart	Paper, pencil, and watch are desirable	Rapid analysis
Any job of any type of sufficient volume and duration to afford analyst's time. First choice on long-run short-cycle operations or where detailed training material is required. Is not suitable for one-time, short-lived jobs	Micromotion study	Moving-picture camera, filming equipment, timing device, and projection equipment	Analysis takes some time but is less costly for detailed analysis and where much methods work is done

CONVENTIONS

Tables 2-3 through 2-7 give the conventions used on the various types of charts.

Table 2-3. Symbols for Process Charts—Product Analysis

Symbol	Name	Used to represent
◯	Operation	A modification of a product which takes place essentially at one location
○	Movement	A change in location of a product from one place to another, not changing the product's characteristics
▽	Controlled storage	Storage of a product under controls such that a requisition or receipting is needed to withdraw it
⛛	Temporary storage	The storage of a product under conditions such that it may be moved or withdrawn from storage without a requisition, *i.e.*, material banked on skids at a machine
□	Quantity inspection	The verification of the quantity of a product against a standard
◇	Quality inspection	The verification of the quality of a product against a standard

Table 2-4. Symbols for Process Chart—Man Analysis

Symbol	Name	Used to represent
◯	Operation	The doing of something at one place
○	Movement	Change in location moving from one place to another
▽	Delay	Idleness or time spent either waiting or moving If the movement was not part of the job could have been spent in waiting
□	Quantity determination	A special form of operation involving the person determining the quantity of an item present
◇	Inspection	A special form of operation involving the person comparing an attribute of a product with a standard

Table 2-5. Symbols Used on Operation Charts

Symbol	Name	In gross breakdown used to represent	In fine breakdown used to represent
◯	Suboperation	Body member doing something in one place such as placing assembling turning, bending etc	Body member doing something at one place, such as taking hold, lining up, assembling etc
○	Movement	Get (reach for take hold of, and move to change location except if lever—reach for and take hold only) or aside (same as get but for the purpose of disposing of an object)	A movement of a body member toward an object or changing the location of an object
⛛	Hold	Body member maintains an object in a fixed position	Same as in gross breakdown
▽	Delay	Body member is idle	Same as in gross breakdown

Table 2-6. Symbols Used with Man and Machine Time Charts and Multiman Charts

Symbol	Name	With man activities is used to represent	With machine activities is used to represent
■	Suboperation	Body member or operator doing something at one place	Machine working
▩	Movement	Body member or operator moving toward or with an object	Not used
▧	Hold	Body member maintaining an object in a fixed position	Not used
□	Delay	Body member or operator is idle	Machine idle (down)

PROCEDURES

The step-by-step procedures for making the three simplest and most commonly used types of analyses follow:

Process Charts—Product Analysis

1. A form or a blank sheet of paper may be used.
2. The analyst should actually observe the production process if possible. If the product is not in process or if direct observation is inconvenient, a scale floor plan should be used. Actual observation is usually desirable inasmuch as deviations between what is supposed to be the process and what actually is the process are not uncommon.
3. Pick a convenient starting place for the analysis.
4. Classify the first step into the proper category by means of Table 2-3.
5. On the first line of the chart place the proper symbol and description. If the step is a movement, pace off the distance or measure it by some means. If the time for the step is desired, time the step with an ordinary watch or stop watch or consult the file of standard times if they are available.
6. On the second line place the proper symbol for the second step, etc.
7. Every time the product moves from one workplace to another, waits, is inspected or worked on, a separate entry should be made. However, movements on a workplace in the course of which the product is worked on should not usually be separately noted.

Process Charts—Man Analysis

1. A form or a blank sheet of paper may be used.
2. The chart may begin at any point in a cycle of work, considering a cycle as the complete set of movements necessary to bring a unit of the product to the degree of completion typical of the process. However, it is usually most convenient to begin with the first movement of the operation connected with a particular unit of the product and end on the last movement before the next similar unit is worked on.
3. The first work step should be carefully classified according to the categories of Table 2-4 and the symbol, explanation, and, if the step is a movement, the distance entered on the first line of the chart. The distance may be paced, estimated, measured on the actual factory floor, or scaled from a drawing. The explanation should be as brief as possible.

Table 2-7. Therblig Definitions and Symbols

Therblig	Symbol	Color	Eagle pencil	Definition
Select	ST	Light grey	734½	Begins when hand or body member touches several objects. Consists in locating an individual object from a group. Ends when the hand or body member has located individual object.
Search	SH	Black	747	Begins when hand or body member gropes or hunts for part. Consists in attempting to find an object. Ends when hand or body member has found location of object.
Grasp	G	Lake red	745	Begins when hand or body member touches an object. Consists in gaining control of an object. Ends when control is gained.
Transport empty	TE	Olive green	739½	Begins when hand or body member begins to move without load. Consists in reaching for something. Ends when hand or body member touches part or stops moving.
Transport loaded	TL	Grass green	738	Begins when hand or body member begins to move with an object. Consists in hand or body member changing location of an object. Ends when hand or body member carrying object arrives at general destination or movement ceases.
Position	P	Blue	741	Begins when hand or body member causes part to begin to line up or locate. Consists in hand or body member causing part to line up, orient, or change position. Ends when body member has part lined up.
Pre-position	PP	Sky blue	740½	Same as position except used when line up is previous to use of part or tool in another place.
Assemble	A	Heavy violet	742	Begins when the hand or body member causes parts to begin to go together. Consists in actual assembly of parts. Ends when hand or body member has caused parts to go together.
Disassemble	DA	Light violet	742	Begins when hand or body member causes parts that were integral to begin to separate. Consists in taking objects apart. Ends when hands or body member has caused complete separation.
Use	U	Purple	742½	Begins when hand or body member actually begins to manipulate tool or control. Consists in applying tool or manipulating control. Ends when hand or body member ceases manipulating tool or control.
Release load	RL	Carmine red	744	Begins when hand or body member begins to relax control of object. Consists in letting go of an object. Ends when hand or body member has lost contact with object.
Inspect	I	Burnt ocher	745½	Begins when hand or body member begins to feel or view an object. Consists in determining a quality of an object. Ends when hand or body member has felt or seen object.
Hold	H	Gold ocher	736½	Begins when movement of part or object which hand or body member has under control ceases. Consists in holding an object in a fixed position and location. Ends with any movement.
Unavoidable delay	UD	Yellow ocher	736	Begins when hand or body member is idle. Consists in a delay for other body member or machine when delay is part of method. Ends when the hand or body member begins any work.
Avoidable delay	AD	Lemon yellow	735	Begins when hand or body member deviates from standard method. Consists in some movement or idleness not part of method. Ends when hand or body member returns to standard routine.
Plan	PN	Brown	740	Begins when hand or body members are idle or making random movements while worker decides on course of action. Consists in determining a course of action. Ends when course of action is determined.
Rest for overcoming fatigue	R	Orange	737	Begins when hand or body member is idle. Consists in idleness necessary to overcome fatigue from previous work which is part of cycle. Ends when hand or body member is able to work again.

4. If the time for the step is desired, time it with an ordinary watch or stop watch.

5. The subsequent steps should be entered on subsequent lines, placing the information in such a way that the symbols, explanations, etc., form separate columns for easy reading.

6. Care should be exercised to make one entry for every separable phase of the work. Each time the worker moves from one place to another, an entry should be made. Each time the worker works at a workplace, an entry should be made. In some cases, two distinctly separate activities may follow each other at a workplace without an intervening movement, and two operation symbols, one after the other, may be convenient.

7. The steps should represent the activities of the worker, what he does to the product, and where he goes rather than what happens to the product.

Right- and Left-hand Operation Charts

1. They are best prepared by actual observation of a worker, although they may be prepared from a proposal for a method.

2. A form or a blank sheet of paper may be used.

3. Although the chart may begin at any point in the work cycle, it is usually most convenient to begin with the hand that makes the first movement in the cycle.

4. The first and last cycles of a work spell may be different from the other cycles. It is usually most desirable to chart the most typical cycle.

5. The first step of the hand that begins the cycle should be classified into the proper category according to Table 2-5 and an entry made on the first line of the chart. The symbols are usually drawn freehand.

6. All the steps of this hand should then be properly classified and plotted in order. It may be necessary to observe a considerable number of cycles in order to accomplish this and to check the completeness.

7. The other hand should then be plotted, care being taken to place each step for this hand opposite the step of the other hand during which it occurs.

8. Since step 7 often involves the squeezing in of lines and items to coordinate the chart properly, it is frequently necessary to redraw it so as to make it easily legible. However, since the chart is mainly an aid to the understanding of the method, excess draftsmanship is purely a waste of time and should be avoided.

CHECK LISTS

The check lists for application to the three simplest and most commonly used analysis techniques follow:

Check List for Process Charts—Product Analysis

1. Who could most conveniently do the job?
2. What does the job really accomplish?
3. Why is it done?
4. Where is it done?
5. Where could it be done?
6. Can the job be eliminated?
   a. As unnecessary
   b. By changing the order of work
   c. By slightly changing the product
   d. By changing the raw-material specifications
7. Should the job be combined with another?
   a. By new tools or workplace
   b. By new equipment
   c. By a new order of work
   d. By slightly changing the product
   e. By changing raw-material specifications

Check List for Process Charts—Man Analysis

1. Can any operation be eliminated?
   - *a.* As unnecessary
   - *b.* By changing the order of work
   - *c.* By new or different equipment
   - *d.* By changes in the layout
2. Can any movement be eliminated?
   - *a.* By leaving out operations
   - *b.* By shifting some operations to another job into which they fit more conveniently
   - *c.* By changing the layout
   - *d.* By changing equipment
   - *e.* By changing the order of work
   - *f.* By conveyors (Make sure they are economical.)
3. Can delays be eliminated?
   - *a.* By changing the order of work
   - *b.* By changing the layout
   - *c.* By new or different equipment
4. Can countings or inspections be eliminated?
   - *a.* Are they really necessary; what happens after they are done and the information obtained?
   - *b.* Do they provide unnecessary duplication?
   - *c.* Can they be performed more conveniently by another person?
   - *d.* Are they done at the best point in the sequence?
5. Can operations be combined?
   - *a.* By changing the order of work
   - *b.* With new or different equipment
   - *c.* By changing the layout
6. Can movements be combined?
   - *a.* By changing the order of work
   - *b.* By changing the layout
   - *c.* By changing the quantity handled at one time
7. Can delays be combined?
   - *a.* By changing the order of work
   - *b.* By changing the layout
   - *c.* If they provide rest, can they be grouped better?
8. Can countings or inspections be combined?
   - *a.* By changing the order of work
   - *b.* By changing the layout
9. Can any step be made safer?
   - *a.* By changing the order of work
   - *b.* By new or different equipment
   - *c.* By changing the layout

It may also be possible to make various steps easier. However, several other types of analyses are of greater help here. Hence, questions concerning easier ways of performing single steps do not appear in this check list.

Check List for Operation Chart

1. Can a suboperation be eliminated?
   - *a.* As unnecessary
   - *b.* By a change in the order of work
   - *c.* By a change of tools or equipment
   - *d.* By a change of layout of the workplace
   - *e.* By combining tools
   - *f.* By a slight change of material
   - *g.* By a slight change in product
   - *h.* By a quick-acting clamp on jig if jigs are used

2. Can a movement be eliminated?
   - *a.* As unnecessary
   - *b.* By change in the order of work
   - *c.* By combining tools
   - *d.* By a change of tools or equipment
   - *e.* By a drop disposal of finished material
3. Can a hold be eliminated (holding is extremely fatiguing)?
   - *a.* As unnecessary
   - *b.* By a simple holding device or fixture
4. Can a delay be eliminated or shortened?
   - *a.* As unnecessary
   - *b.* By a change in the work each body member does
   - *c.* By balancing the work between the body members
   - *d.* By working simultaneously on two items
   - *e.* By alternating the work, each hand doing the same job but out of phase
5. Can a suboperation be made easier?
   - *a.* By better tools
   - *b.* By changing leverages
   - *c.* By changing positions of controls or tools
   - *d.* By better material containers
   - *e.* By using inertia where possible
   - *f.* By lessening visual requirements
   - *g.* By better workplace heights
6. Can a movement be made easier?
   - *a.* By a change of layout, shortening distances
   - *b.* By changing direction of movements
   - *c.* By using different muscles

     Use the first muscle group in this list that is strong enough for the task
       - Finger
       - Wrist
       - Forearm
       - Upper arm
       - Trunk
   - *d.* Making movements continuous rather than jerky
7. Can a hold be made easier?
   - *a.* By shortening its duration
   - *b.* By using stronger muscle groups such as those of the foot in operating vises

STOP-WATCH TIME STUDY

Definition. A stop-watch time study is used to find the amount of time necessary to accomplish a unit of work, using a given method, under given conditions of work, by a worker possessing a specified amount of skill on the job and a specified aptitude for the job, when working at a pace that will produce, within a unit of time, a specified physical effect upon him.

Uses

1. As a basis for setting schedules
2. To determine supervisory objectives
3. To determine operating effectiveness
4. To set labor standards
5. To determine the number of machines a person may run
6. To balance the work of crews working coordinately or in a production line
7. To compare methods
8. As a basis for computing standard costs
9. To determine equipment and labor requirements
10. To determine synthetic times
11. To provide a basis for the setting of piece prices or incentive wages

Components. The activities required for taking stop-watch time studies may be grouped under five heads:

1. Determination of objective
2. Recording of method
3. Timing of an individual
4. Rating and adjusting
5. Allowances

Objective. The first step in the setting of time standards requires the substitution of some definite adjective for each use of the word *specified* in the generic definition previously given, such as:

A stop-watch time study is used to find the amount of time that will be necessary to accomplish a unit of work, using a given method, under given conditions, by a worker possessing sufficient skill to do the job properly, as physically fit for the job, after adjustment to it, as the average person who can be expected to be put on the job and working at the maximum pace that can be maintained day after day without harmful physical effects, to which 30 per cent has been added. (An infinite number of variations on this definition are possible.)

Definitions of Terms in Objective. The time obtained from a stop-watch time study will be called the standard time.

Skill is the ability to do a job in the proper manner; the ability to repeat a definite muscular pattern. The higher the skill, the faster the possible pace before the muscular coordinations fail.

Aptitude is the physical fitness for the job.

Physical effect is the muscular and mental deterioration as a result of work.

Method Record. There is considerable agreement in the field concerning the requirements of the method record. It must be complete. Any of the work simplification techniques may be used to describe the method or it may be described in paragraph style.

The essential criteria of any method description are

1. Is it sufficient so the job could be reproduced from it?
2. Is everything the worker has to do recorded?

The method record should also include

1. Products and material specifications
2. Workplace layout
3. Surrounding environmental conditions
4. Tools
5. Equipment
6. Feeds and speeds of machines, welding currents, etc
7. Services in the way of machine and tool maintenance, delivery and materials handling rendered to or required of the worker
8. The name and clock number of the operator observed
9. The date and time the time study is made
10. The department in which the job exists
11. Any other pertinent information

The method record should be broken down into elements which are as small as possible and yet large enough to time adequately The elements should be chosen in accordance with the following requirements·

1. Well defined as the first motion of the element, what it involves and where it ends
2. As small as is convenient to time
3. As homogeneous as possible
4. Hand time should be separated from machine time
5. Constant elements should be separated from variable elements
6. Regular and irregular elements should be separated

Timing. Times for stop-watch time studies may be obtained with any one of three devices:

1. Stop watch
2. Marstochron
3. Motion picture camera

Stop Watch. Stop watches may be used in one of three ways:

1. Continuous
2. Repetitive or snapback
3. Accumulative

Continuous. In the continuous method of using a stop watch the watch is started at the beginning of the first element of the first cycle being timed, and allowed to run continuously throughout the study. The times for each element are obtained later by successive subtractions.

Repetitive. In the repetitive method the watch is simultaneously read and snapped back to zero at the completion of each element.

Accumulative. In the accumulative timing two stop watches are used and the time is alternately accumulated on the two watches. When one is started the other is stopped and before starting again it may be snapped back to zero or merely restarted from the stopped position.

NOTE: In many places in the literature a method called *cycle-timing* is described. This method consists of timing the elements in groups of one less than the number in the cycle. It is followed by a mathematical procedure for the determination of the individual element times. This method does not produce the correct element times and it should not be used. It involves the assumption that the average value for each element computed on the basis of a small sample of cycles will be the same as the average value for each element, computed on the basis of a sample $(x - 1)$ times the size of the original sample, with x equal to the number of elements in the cycle. A huge number of recordings would be necessary for this to be a reliable assumption.

Marstochron. The Marstochron is an instrument which draws a narrow tape of paper at uniform speed under a set of finger-actuated pencils. By depressing these pencils at intervals corresponding to the terminal points of elements, the time for each recording may later be found by measuring the length of paper between successive pencil marks. This method may be highly accurate.

Motion-picture Camera. Recent practice in taking stop-watch time study has involved the use of a constant speed, one frame per second, motion-picture camera. This permits simultaneous recording of the method and time to a degree of preciseness not obtainable by other methods The reliability of the record is also enhanced.

Recent thinking in the time-study field suggests that some of the statistical quality-control techniques should be applied to time studies to ensure that the average values which are used as representative of each element are reliable averages. Reference should be made to standard works on statistics.

Whatever the timing device used, most present-day practice involves the determination of an average or mean value for each element.

Rating and Adjustments. To the preceding values, obtained by any one of the means given, some correction factor must be applied to relate the value observed to that called for by the definition or objective. Such an adjustment factor is called a rating.

Rating procedures may be divided into one of three groups:

1. Mathematical
2. Fully subjective
3. Semisubjective

Mathematical. No accurate, reliable mathematical means of performing the rating procedure is at present known.

Fully Subjective. Fully subjective techniques usually involve comparison by the time-study man of the pace of performance he observes to his concept of proper pace of performance for the job, and as such, is subject to the typical errors of unanchored human judgment. Commonly described procedures may involve a rating of skill, effort, conditions, and consistency; skill and effort; or effort alone.

Semisubjective. Semisubjective methods may involve comparison of observed rates of activity or pace of performance to recorded paces portrayed on endless loops of motion-picture film. Some such systems rate all jobs to a single pace and correct later for the nature of the job. **Present indications are that such procedures offer more reliable bases for performing this step.**

Allowances. Allowances added to a time study are usually percentages of time, or increments of time per hour, added to the base time for the job so as to give the worker personal time, time for tool or material handling, etc. Allowances may be accurately determined by means of *ratio delay* studies which are based on a sampling procedure or by recording sufficient values continuously to provide an accurate estimate of the time required by the activity for which the allowance is made.

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INCENTIVE WAGE METHODS

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OBJECTIVES

> The human donkey requires either a carrot in front or a stick behind to goad it into activity. It is fashionable at the moment to argue that the carrot is the more important of the two: "incentive" is the watchword, and all classes of the community are busy arguing that if only they are given a little bit more in the way of incentive (at the expense of the rest of the community) they will respond with more activity. From miners to company promoters, the basic argument is the same. And no doubt, within limits, it is quite correct: a bigger carrot would make the donkey move a bit faster.[1]*

It is evident that the emphasis has shifted during the past 60 years from punishment for disobedience, defective work, and loafing to reward for skill, effort, ingenuity, and competence. It is also evident that the method of reward—*viz.*, the wage-incentive plan—has frequently been ill-conceived, unfair, inaccurate, and misused.

> . . . the use of carrots as bait for the donkey has neither universally nor permanently solved all of the knotty problems in the relationship between the driver and the cart's prime mover, although the device may have served useful on many occasions. Likewise, wage incentives have been used with success under proper circumstances, but they have not proved to be a panacea for industrial ills.[2]

The theoretically simple principle of permitting maximum use of skill and effort and rewarding accordingly has had many pitfalls in actual practice. The decade preceding the Second World War saw considerable advancement in correcting the evils. Top management took increasing interest in wage-incentive plans, and organized labor demanded sound and fair incentive plans. Today, the value of the wage-incentive plan which achieves higher rate of output with lower unit cost and higher employee earnings is no longer questioned. The problems are those of determining the goal of industry in the matter of wage-incentive coverage and establishing the type of wage-incentive plan, if any, best suited for each set of conditions.

The application of wage-incentive plans should be restricted to those operations where both management and labor mutually benefit from the application of the incentive plan. The sound incentive system should not be installed to quench a fire kindled by labor dispute. It should be used only after methods have been highly developed and standardized. It should be used only after consideration is given to the wage rate and incentive relationship of the workers covered by the proposed installation to other workers in the department or workers performing similar work elsewhere in the company or area. It should be used only after consideration is given to the feasibility of placing the workers covered by the proposed installation on incentive. It should be used only after joint discussion with labor representatives and the employees affected of the need for and the type of incentive installation. It should be used only after the incentive standards have been accurately developed, the degree of accuracy depending largely upon their use. It should be used only when the development and application of the whole incentive plan is simple and understood by the employees affected. No incentive plan is far better than one poorly conceived and poorly maintained.

In the past, many types of wage-incentive plans have been developed to meet varying conditions. An analysis of the major classifications of wage-incentive plans will indicate guiding principles to be used for their development and installation.

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* Superior numbers refer to specific references listed at the end of this subsection.

DEFINITION OF TERMS

Definitions of terms, particularly wage-rate terms, vary considerably from industry to industry and from company to company; therefore, no attempt has been made to be all-inclusive.

Actual Hour. An actual hour is a unit of time taken to perform a given amount of work.

Standard Hour. A standard hour is a unit of time allowed to perform a given amount of work at a rate of output that a worker or group of workers qualified for, and skilled in, the particular work may be expected to attain for reasonable wage-incentive pay, given good supervision and supplied with an unlimited amount of work, and to maintain over long periods of time without physical or mental impairment.

Operating Performance, or Productive Efficiency. Operating performance is the ratio of standard hours to actual hours. It is generally expressed in per cent.

One Hundred Per Cent, or Expected Operating Performance. One hundred per cent operating performance is that level of operating performance attained when standard hours equal actual hours (or actual production equals standard production).

Occupational or Evaluated Rate. The occupational rate is the prevailing hourly rate for fulfillment of minimum requirements of a job. It is frequently determined by job evaluation and an appropriate rate curve, with due consideration given to rates paid for similar occupations in other departments and the community and to rates paid in the past. It is the rate that would be paid for the job without wage incentive. In many industries, the *base rate* is synonymous with the occupational rate.

Guaranteed Rate. The guaranteed rate is the minimum rate paid for a job and is generally the occupational rate.

Pay Performance. Pay performance is the ratio of actual hourly earnings to the occupational rate. It is generally expressed in per cent.

Expected Pay Performance, or Expected Earnings. Expected pay performance is that level of pay performance attained when operating at 100 per cent operating performance. It is a level of earnings at a specified percentage amount above the occupational rate.

Earning Curve. The earning curve graphically represents the relationship between pay performance and operating performance.

EARNING CURVES

Purpose of Earning Curves. The relationship between reward and effort, or amount of wages earned and production, is of primary importance, and for any wage-incentive plan this relationship can be represented graphically or expressed in formula or tabular form. Formula or tabular form is used in actual wage calculations for convenience and accuracy; the graphic form, however, is more satisfactory for development and comparison purposes. In graphic presentation, pay performance (actual hourly earnings in per cent of occupational rate) is scaled on the vertical axis, and operating performance (actual production in per cent of standard production) is scaled on the horizontal axis. The percentages permit the use of one curve to represent the relationship for jobs of varying occupational wage rates and production units; whereas the use of actual occupational wage rates and production units would require a family of curves.

In the following classification and analysis of wage-incentive plans by earning curves, the strong and weak incentive features of each type are indicated by the slope of the curve. They range from the strong wage incentive with the steeply sloped curve to the weak wage incentive with the flat curve. The flat curve approaches the horizontal axis along which earnings are constant regardless of production.

Classification by Earning Curves. There are various classifications of wage-incentive plans by their earning curves. One of the simplest classifications makes the distinction between straight-line (constant-sharing) and curved-line (variable-sharing) plans, with a breakdown of the straight-line group into three subdivisions: curves parallel to the horizontal axis, straight-line curves originating at the origin, and all other straight-line curves. In actual practice, however, the earning curve is frequently more complex because it is composed of a combination of segments of curves taken from the straight- and curved-line groups. In addition, step curves are occasionally used. In illustrating various major types, whenever convenient, the names and curves of a few of the more noted plans are used as being representative of their particular type. These plans usually bear the name of the engineer by whom they were developed.

Time-rate plans, represented by the horizontal line, have been included for the sake of completeness, although they are not considered a wage incentive as earnings are constant, regardless of production (Fig. 2-10). With the differential time-rate plan, however, earnings are related to production to the extent of the step or steps. As the number of steps in the differential time-rate plan increases, the shape of the curve approaches those of the constant- or variable-sharing types. A variation of the differential time-rate plan is called "measured day work." Under this plan, an individual hourly wage rate is composed of an occupational rate plus an incentive rate which is based on past individual operating performance. To determine the incentive portion, a long period (1 or 3 months) of past individual operating performance is reviewed. The incentive portion of the rate is developed, and a new individual hourly wage rate is established and used until the next periodic review (Fig. 2-11).

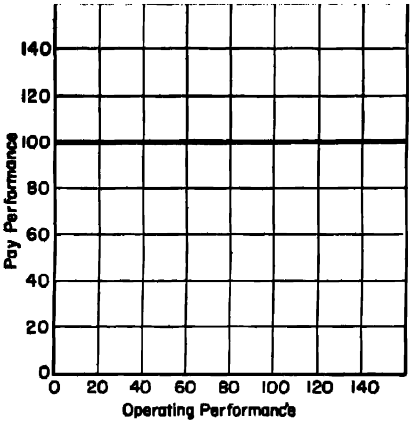

FIG. 2-10. Time-rate plan.

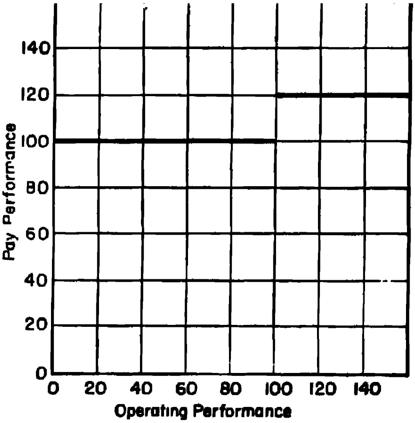

FIG. 2-11. Differential or step time-rate plan

The piece-rate plan is of the constant-sharing type where the earning curve passes through the origin; it is generally considered separately, however, because of its extensive use (Figs. 2-12, 2-13, 2-14, 2-15). Earnings under the constant-sharing plan may be considered to be made up of two portions—a time portion and an incentive portion. The area between the horizontal axis and a line parallel to it which crosses the vertical axis at the same point as the earning curve, when extended if necessary, may be considered the time or nonincentive portion (this area is zero for the piece-rate plan); while the portion between this line and the earning curve is the incentive portion and remains as constant sharing per unit of production throughout the production range (this area is zero for the time-rate plan). With constant-sharing plans, the (50-50) or (75-25) figure frequently shown indicates the point at which the straight-line curve, when extended if necessary, would cross the vertical axis (Figs. 2-16, 2-17, 2-18).

One classification of the earning curves of variable-sharing wage-incentive plans has been by the geometric form of the curve, such as hyperbolas and parabolas; further classification as to direction of concavity (upper or lower side), however, probably is more enlightening (Figs. 2-19, 2-20). Most variable-sharing curves are used in combination with constant-sharing curves (Figs. 2-21, 2-22). The straight-line curve frequently takes the form of the time guarantee for low levels of production, when used either with the constant- or variable-sharing curves or with some combination of them.

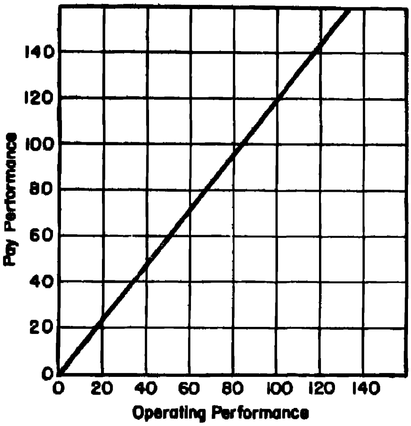

Fig. 2-12. Piece-rate plan.

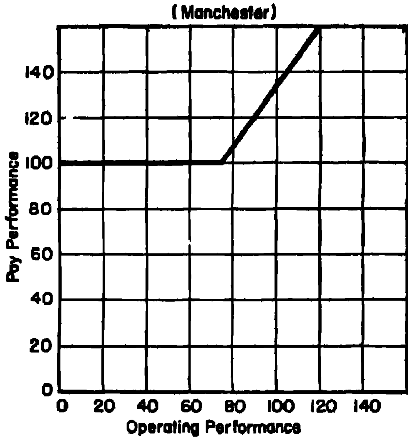

Fig. 2-13. Piece-rate plan with time guarantee.

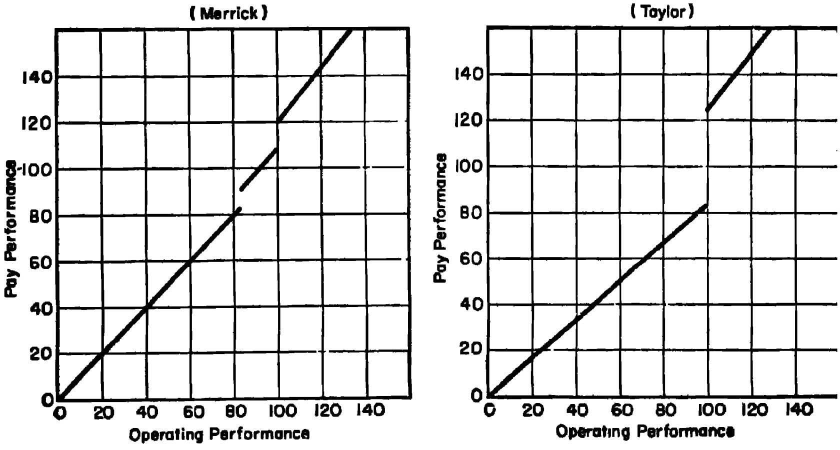

Fig. 2-14. Differential or step piece-rate plan.

The guaranteed and occupational rates, the slope or slopes of the curve, and steps in the differential type of plan vary according to the results desired and conditions surrounding a particular plan. Principles outlined under Guaranteed Period and Rate, Expected Earnings, and Analysis of Earning Curves indicate the normal range of earning curves.

Expected Earnings. In the discussion of the various classifications of wage-incentive-plan earning curves, no detail was given on slope or relative position of the earning curve. Determination of these factors is a question of policy as they must be established somewhat arbitrarily. Consideration should be given to the surrounding conditions and purpose of the plan, such as desired wage-incentive strength and other factors given under Analysis by Earning Curves. So many factors affect operating performance that isolation of the effect of one factor—the financial incentive—is exceedingly difficult. It is known that its effect is different under each set of varying conditions, if not even with each individual. Therefore, until better methods are devised for such determination, past performance is our chief guide. In actual practice, expected earnings, *i.e.*, earnings at expected or 100 per cent operating performance, range generally from 120 to 130 per cent pay performance. In recent years the trend has been toward the higher figure. In other words, 20 to 30 per cent bonus is considered as the necessary additional compensation to encourage workers materially to increase effective effort above normal nonincentive pace. When the definition

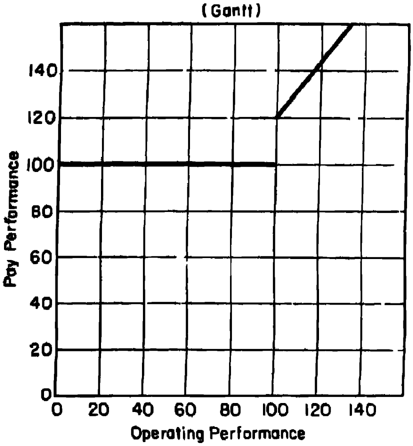

FIG. 2-15. Differential or step piece-rate plan with time guarantee.

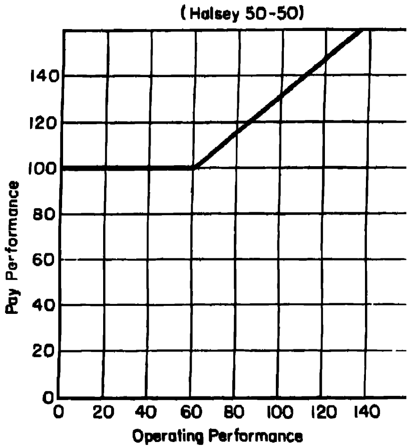

FIG. 2-16. Constant-sharing plan with time guarantee.

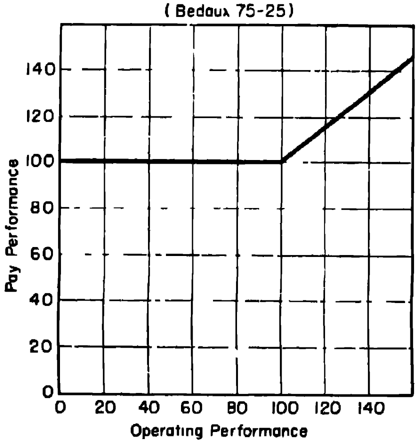

FIG. 2-17. Constant-sharing plan with time guarantee (expected operating performance 133).

of a standard hour is different from that defined above, the expected operating performance is not 100 per cent (*example* Bedaux).

Guaranteed Period and Rate. Some of the illustrations of earning curves shown include guaranteed rates (Manchester, Gantt, Halsey, Bedaux, Diemer, Baum, Emerson, Wennerlund, Knoeppel, Bigelow, and Rowan plans), whereas others do

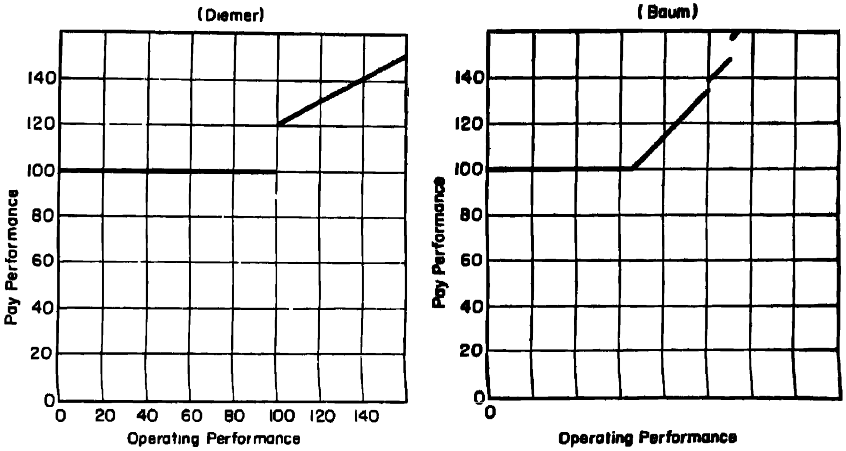

FIG. 2-18. Differential or step constant-sharing plan with time guarantee.

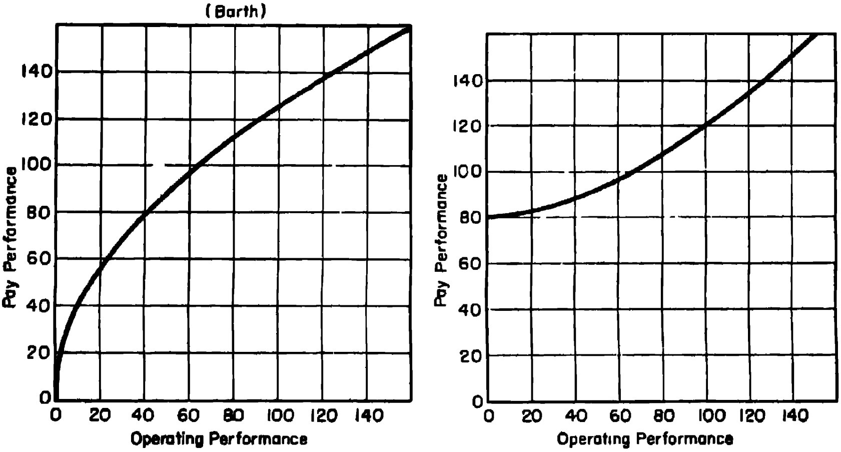

FIG. 2-19. Variable-sharing plan.

not (straight piece, Merrick, Taylor, and Barth plans). Theoretically, the lower the guarantee, the stronger the incentive. Nevertheless, in actual practice, the guarantee of occupational rates for incentive workers has become accepted. This guaranteed rate has become a standard feature because of the legal, social, and psychological requirements of a wage-payment plan. The guarantee is generally unconditional, assuring the employee his occupational rate for the hours worked regardless of the reason for the low output. In addition, present union contracts may call for guarantees above the occupational rate when earnings are affected by conditions beyond the employee's control.

The length of the guaranteed period depends upon the circumstance surrounding each installation. The job, day, or pay period (1 or 2 weeks) is the period for which the guaranteed rate is generally satisfactorily applied. Within reason, the shorter the period for which the guaranteed rate is applied, the stronger the incentive. For supervisory employees working under wage-incentive plans and for plant-wide plans, the guaranteed period is a month or more.

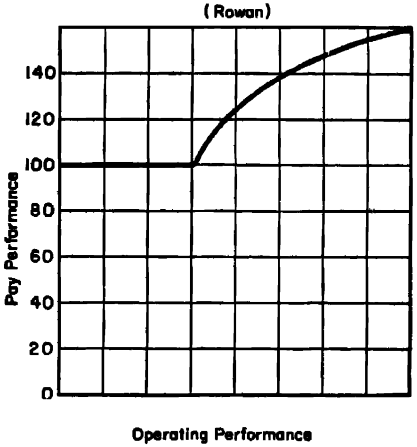

FIG. 2-20. Variable-sharing plan with time guarantee.

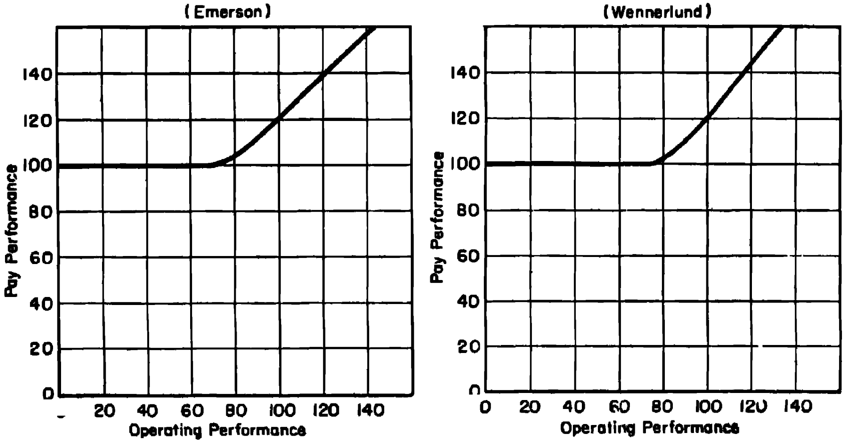

FIG. 2-21. Combination constant- and variable-sharing plan with time guarantee.

Analysis by Earning Curves. The rate of premium, or sharing ratio, on all production above the prescribed production level should be based upon the directness of the relationship between production in terms of the standards determinants and the skill, effort, and other factors affecting production over which the individual or group on incentive has control. Where the degree of control is relatively high, earnings should be in direct proportion to production.

There are no specific rules for use in selecting the best wage-incentive plan for a given set of conditions. The conditions surrounding each proposed installation cannot be defined in absolute terms but must be considered in relation to conditions of other wage-incentive installations. Analysis of these conditions, as factors affecting the selection of the earning curves, is a matter of degree. Selection of a proper wage-incentive plan is based upon the relative importance of each factor making up a given set of conditions to other factors in the same set and other sets of conditions. Those factors which appreciably influence selection of the earning curve and their effect on the earning curve follow.

Strong wage-incentive plans with steep earning curves are usually installed when one or more of the following conditions exist:

1. When standards for measuring performance can be accurately and practically established
2. When operating performance can be readily obtained
3. When job and methods are highly developed and standardized
4. When work is routine or at least elements of the work are repetitive

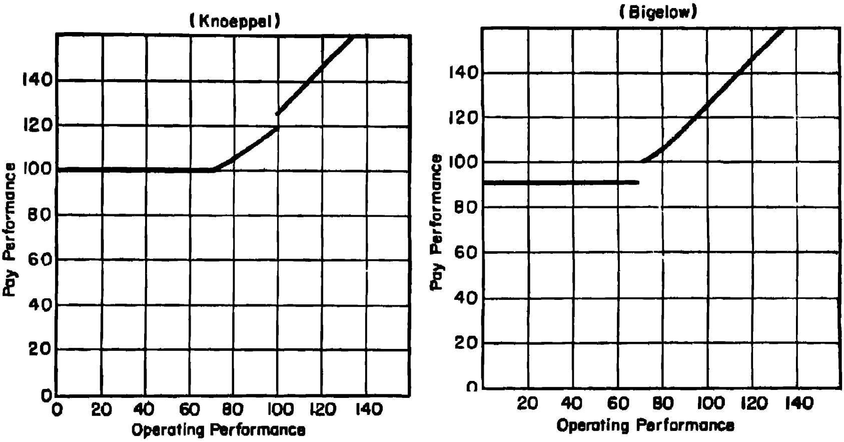

FIG. 2-22. Differential or step constant- and variable-sharing plan with time guarantee.

5. When operator has considerable control over production
6. When operating conditions and equipment are satisfactorily maintained to minimize delays, breakdowns, etc.
7. When ample backlog of work exists
8. When overhead costs are high in relation to labor costs so that emphasis is on production

Weak wage-incentive plans with flatter earning curves or time rates are usually installed when one or more of the following conditions exist:

1. When standards for measuring performance cannot be accurately or practically established
2. When skill and effort do not necessarily have a direct effect upon production
3. When considerable fluctuations in earnings might result from wide fluctuations in production uncontrollable by the employee
4. When job and methods are not developed or standardized
5. When work performed is not a key operation in controlling rate of production but is of a paced or service nature
6. When work is temporary in nature
7. When there is possibility of considerable unavoidable delay, breakdowns, and nonstandard work
8. When operating performance is low (minimum-wage guarantee)
9. When work is highly technical
10. When used to pay for delays, nonstandard work, etc. (occupational rates)

Wage-incentive plans with step earning curves or a combination curve which is weak in certain areas and strong in others are installed when the condition exists in which it is desirable to maintain production at a certain point, or area, above which little or no increase is desired.

INDIVIDUAL OR GROUP PLAN

The individual type of wage-incentive plan is used where one employee works independently; the group type of plan is used when two or more employees work on a common unit or group of units. The individual type is generally considered a stronger incentive because reward is directly associated with individual performance. With the group type, the earnings of one employee may be affected by performance of another, as each employee of the group shares the resulting earnings in proportion to his actual hours worked and his occupational rate. Under certain conditions, however, measurement of individual performance is not practical and so the group type of plan is used. Naturally, the smaller the group, the stronger the incentive. Historically, considerable success has been obtained with groups ranging from two to ten or fifteen employees. The effect of one's effort on larger groups is so diluted that it is generally considered to have little incentive value. The success of any group plan depends largely upon the degree to which the individuals can be mutually helpful, the teamwork of the group, their common interest, and the group spirit. Also, the work load must be properly and uniformly balanced among all members of the group.

There is no direct connection between the use of the individual or group plan and the earning curve. The principles governing the selection of earning curves apply equally for the individual or group type of plan. The formulas for the two types are frequently different, however, because of the man-hour-control feature of many group plans. Such a condition occurs when the size of the group is not necessarily constant and when group actual and standard hours are used instead of total of the individual hours.

In recent years there has been limited application of the group plan on a broader scale with group composition covering a department or even a whole plant. Under such an installation, particularly for a larger group, the additional earnings are frequently of the profit-sharing or Christmas-bonus type, in contrast to the physical type of wage-incentive plan where man performance is directly measured, and reward is made accordingly. When bonus earnings are based on such determinants as dollars of sales, per cent of net profit after or before taxes, and length of service, the factors influencing the amount of the bonus are so remote from the performance of an individual that the effect of the plan as an incentive is negligible. Other financial types based on dollars of sales per man-hour, dollars of sales per pay-roll dollar, actual per cent compared to standard per cent of labor cost to sales, or dollars of sales weighted by type of product per man-hour are a decided improvement over those totally lacking in man-hour control. In these an attempt is made to connect the bonus with total man-hours or labor cost. Many of these plans have claimed success where the department or plant is not too large and the group spirit strong. One advantage of many group plans—particularly department or plant-wide plans—is that they are easier to administer than individual plans; although even the plant-wide plan requires extensive service work.

DIRECT OR INDIRECT WORKERS

The same principles that apply to the determination of the earning curve and the use of individual or group type of plan for direct workers also apply to the development of incentive plans for other types of workers. In fact, the classification of employees as direct, indirect, clerical, supervisory, skilled, or semiskilled workers, to mention a few, has little or no bearing upon the general features of a wage-incentive plan. The type of operation performed by the workers is the important consideration.

Historically, management has been production-conscious; so, naturally, tools to implement production, such as wage incentives, would appear for direct workers first, especially in consideration of the fact that measurement of the performance of direct workers is generally more practical than measurement of the other classes. Gradually, other groups have been increasing their demand for wage incentives because semiskilled direct workers on wage incentives frequently receive higher wages than skilled indirect, clerical, or supervisory workers. The need for increasing the skill and effort of the indirect, clerical, and supervisory classes is probably as great as it is for the direct workers, if not greater. Management recognizes this need. However, it does not consider the wage-incentive tool, with the exception of the plant-wide plan, practical for most indirect, clerical, and supervisory workers.

Wage-incentive installations may be made for indirect workers when the functions of the indirect workers are not closely connected with direct operations. This group may include general maintenance workers, storeroom workers, and receiving and shipping workers. When standards can be accurately developed and practically maintained for these indirect operations, the indirect workers are paid on their own individual or group performance. Under such conditions, incentive development and application for indirect workers is identical to incentive development and application for direct workers.

Wage-incentive installations may also be made for indirect workers when the economical use of the indirect workers is second in importance to their influence on the performance of the direct workers. In this group may be included crane operators, transportation workers, assigned maintenance workers, assigned inspectors, and setup men. In establishing this type of indirect service workers' wage-incentive plan, the performance of the direct workers and the utilization of the indirect workers are important. Incentive payment of the indirect workers is made on the basis of the performance of the direct workers, provided a man-hour control establishing the ratio of direct to indirect workers is used. The indirect man-hours should be correlated to a measurable unit, such as direct man-hours or production. Whenever feasible, however, indirect workers should be solely or partly paid on their own individual or group performance. Payment to indirect workers of a bonus based solely on the performance of the direct workers without measurement of the productivity of the indirect workers amounts to a wage increase and destroys the value of the direct workers' incentive plan The relationship between performance and reward for indirect workers, as for direct workers, must be direct or else the value of the incentive is lost.

Thus, the standard objections to wage incentives under each of the above conditions is (1) the cost of development, maintenance, and application of incentive standards for most indirect jobs not closely connected with operations exceeds the value received, and (2) the performance of indirect workers servicing direct workers can have little direct effect upon production. The importance of the first objection diminishes, however, as operations become standardized and routine, and the importance of the second objection diminishes when an increase in man performance results in increased production and possibly in a decrease in service hours expended.

Learners and apprentices are protected by minimum guaranteed rates, and generally no additional wage-incentive features are considered necessary. However, when learners and apprentices are members of a group whose incentive earnings are determined as a group, the earnings of the other members of the group will be affected by inclusion of learners and apprentices in the group. Under such conditions, an allowance may be permitted the other members of the group.

Wage-incentive installations for clerical workers are based on the same principles as those for direct workers. Performance of some clerical workers, however, is difficult to measure as there are frequently many factors affecting performance, the relative importance of each of which is difficult to determine. Even more difficult to measure is the performance of technical workers. About the only incentive application that may cover this latter group is a plant-wide plan.

In consideration of the difficulty encountered in measuring performance of hourly rated employees, either in the shop or office, even more caution should be exercised in attempting to measure the performance of supervisory employees for wage-incentive purposes. When used, wage-incentive installation for supervisory workers is based on the same principles as those used for indirect service workers. The major difference lies in the standards and the guarantee period. The guarantee period is a month or longer, such as 3 months or a 3 months' moving average. Standards may be based solely on the performance of the employees supervised but more frequently include all factors over which the supervisor has control, such as maintenance expense, utilities, yields, quality, nonproducing labor, etc. Standard cost or variable budgetary-control standards are used frequently as the basis for such wage-incentive standards. Thus, the supervisor's performance is determined by the standard cost performance of the department or departments over which he has control.

Another type of worker frequently covered by wage-incentive plans is the salesman. In the past, by far the great majority of standards for measuring the performance of salesmen have been set by nonscientific methods. These sales standards, or quotas, as they more frequently have been called, were established by estimates of executives or based on past performances, sales records, or established by estimates of salesmen. A more scientific approach would include a thorough analysis of all factors that may have an effect upon the standard. The importance of the various factors may vary from industry to industry, or even from company to company, and what is practical to analyze in one situation may not be in another. The important thing is to achieve the greatest possible accuracy for those factors which have the greatest influence. A list of factors would include past sales experience, general business conditions, potential market, changes in sales policies, and competition.

SOME CHARACTERISTICS OF OPERATION AFFECTING PLAN

Unavoidable Delays. Delays should be reduced to a minimum before incentive development and installation are undertaken. Even after extensive precautions have been taken, employees on incentive occasionally are required to spend time waiting for material to be brought to them, for machinery to be repaired, for machinery to be adjusted, for resumption of work after power failures, or other unusual conditions which are no fault of their own. The type of delay will depend upon the nature of the work and the conditions that surround it, and only to a minor extent will the experienced operator, through better planning, have fewer such delays than the poor operator. As such unavoidable delays adversely affect the earnings of the employees, payment is generally made for all or part of the time the employees are delayed. Theoretically, such delays are an indication of poor management, and payment for them weakens the wage incentive.

For convenience, specified unavoidable delays of short duration are included in the standards because continual recording and approving of 2- and 3-min delays would require excessive foreman time for approval of payment for the delayed time. No additional payment is made for this type of delay. For delays of a long duration, allowances for which have not been included in the standards, payment is made on an actual occurrence basis. The length of the delay time to be included in the standards, the length to be paid for, and the rate of pay are determined in part by the nature of the delay. They vary considerably in actual practice. Some plans call for payment on an occurrence basis for the full duration of the delay, whereas others call for payment of the delay time beyond a specified period, such as 15 or 30 min. Under no conditions should the full delay time be paid for delays during this specified period because this would make it advantageous to the worker to prolong delays that are shorter than the specified period. A definite procedure should be established for payment of allowable delays, preferably at occupational or base rate, whether they are included in, or excluded from, the standard.

The higher the hourly rate used for payment during the delay period, the weaker the incentive. As this rate approaches average hourly earnings, a condition is reached where number and length of delays have no effect on earnings, thereby defeating one purpose of the incentive plan. The occupational or base rate is the most widely accepted rate. Some use has been made, however, of rates above the base rate, such as average earnings or a specified percentage of average earnings.

When part of the equipment operated by an individual is idle, such as one or more of a bank of machines, the determination of its effect on performance is difficult and generally impractical, and little or no attempt is made to obtain an accurate analysis. An arbitrary or developed adjustment may be made in the standards or rate allowed.

Man versus Equipment Performance. In operations where machines are used, the operator is frequently idle or at attention during part or all the machine elements. Ideally, this time is kept at a minimum through proper arrangement of equipment and operations. In actual practice, considerable idle or attention time occasionally exists. Where automatic machine time with set feeds and speeds is extensive, the operator should utilize his time during the machine elements on other equipment or at other operations. This frequently involves the operating of two or more machines by one operator. Where the operating cycle is short, manual elements usually overlap machine elements, and attention time is negligible. When the capacity of the equipment is greater than man's capacity, the standards are based largely on man performance; when man's capacity is greater than the capacity of the equipment, the standards are based on equipment utilization. A combination of man and equipment performance standards is frequently used when both conditions exist for a particular job. In the United States, because of a high wage level, emphasis is on full utilization of the man's time. The strength of the wage incentive is directly proportional to the degree of utilization of man's time.

Quality and Defective Work. Theoretically, an employee at fault should not be paid for defective work, and when payment is made it weakens the incentive plan. General practice has been to require an employee to correct the defective work, whenever possible, on his own time in order to receive payment, when the identity of the employee at fault could be established. When the defect could not be corrected, the employee was paid for acceptable product only. Frequently, the identity of the employee producing the defective work cannot be established, and payment for all product is made. However, many plans where such payment occurs claim to pay for acceptable product only. An employee should be paid for work on defective products resulting from faulty materials, etc., only when he is not responsible for the cause of the rejection.

In some industries there now appears to be a tendency to include incentive payment for all products, regardless of the reason for rejection. The trend away from expectation of highest quality, regardless of specifications, toward requirement of quality within specified limits, has caused a shift in thinking about payment for quality and defective work. With the present subdivision of work, better control of materials, and equipment, the feeling prevails that control over quality is management's responsibility and that no quality control should be tied in with the wage incentive. The poorer employees, after reasonable training and trial periods, should be transferred or released

A definite procedure should be established governing the payment for defective work. The rate, if any, used for defective-work payment varies with the surrounding conditions. The theoretically sound wage rate for time spent by an employee at fault working on defective material is zero, and the farther the actual rate deviates from the theoretical, the weaker the incentive. In present practice, the occupational rate is used frequently in payment for the time spent on defective work; under conditions where the employee at fault corrects the defect or where the identity of the employee is lost, payment is made through application of regular incentive standards. When the employee is not at fault, average earnings, approximate average earnings, or other rates above the occupational rate are used and justified on the basis that the employee should not be penalized for conditions beyond his control.

Nonstandard Time. Wage-incentive-plan installations are considered desirable for employees wherever accurate incentive standards can be practically established and maintained. Ideally, standards should be available prior to commencement of work for all the elements of work performed by an employee on incentive. Practically, however, certain work elements occur for which accurate standards development is impossible or not feasible. The time spent by an employee performing work for which no standards have been developed is considered as nonstandard time and arises either when an employee is temporarily assigned to a special job other than his regular job or is temporarily assigned to a regular job for which standards have not been developed. Naturally, the amount of nonstandard time should be kept to a minimum as the strength of the incentive varies inversely with the amount of such time.

Method of payment for nonstandard time varies considerably in actual practice. The proponents of a relatively high rate, such as average earnings, hold that the workers should not be penalized for conditions over which they have no control. On the other hand, the proponents of a relatively low rate, such as occupational rate, hold that the higher rates weaken the incentive because earnings are maintained regardless of performance on nonstandard work. With the lower rate, there is an incentive to get back to work covered by standards. When an employee is temporarily assigned to a special job, the reason for the transfer has considerable bearing upon the rate used. If there is no work available on the regular job when the transfer to the nonincentive job is made, the nonincentive rate is generally used. If the transfer is made because of the immediate need for an operator on the nonincentive job, although there is still regular work available, then average earnings probably would be the rate used for payment on the nonincentive job. When an employee is temporarily assigned to a regular job for which standards have not been developed, the issue is not so clear-cut. Generally, an attempt is made to use the occupational rate, although there is a recognized tendency to pay average earnings, a specified percentage above base, or a specified percentage of average earnings.

Equipment or Method Change. It is an almost universally accepted rule that standards will not be revised unless there is a change in method, equipment, or product. When there is such a change, the time for only those elements of the job which are affected is revised. Also, good faith on the part of management requires that method changes be legitimate and not deliberately sought because of an error in the standards. It is considered essential that standards be revised after a change whether the change was proposed by management or labor. If standards are revised only when management instigates a change, the rate structure soon becomes distorted. Employees proposing worth-while changes should be compensated by other means for their suggestions. The theoretically simple rule of revising standards for changes in method, equipment, or product is difficult to apply in actual practice. Minor changes which require an endless amount of servicing for proper rate maintenance are continually occurring. The practical limits to which one may go in making rate and standards changes vary for each set of conditions. Under present practice, an attempt is made to revise standards if the change has made an appreciable effect upon earnings, such as 5 per cent or more. When method changes occur, which require a lapse of time for the new methods to become standardized and proposed standards to be developed, payment is frequently based on temporary standards or average earnings.

The department responsible for maintaining standards is frequently not notified when minor changes occur. Even though aware of the changes, they often do not know the exact conditions before and after the changes and, consequently, the effect of the change. Even when these two conditions are known, the proposed standards are often subject to bickering, and the change leads to discontent. Revision is exceedingly difficult when standards are based on equipment utilization. Also, revision is made difficult when incentive payment is being made for low levels of man performance, thus introducing the problem of definition of a "fair day's work." Considerable attention has been given by industry to devising a method of routine revision of base rates and standards. Use of such a method would assure adequate payment for additional work or decreased work resulting from technological change. At present, there is no adequate solution to the problem of revision.

STANDARDS

Performance Measurement. Although systematically measured performance standards are associated most frequently with incentive compensation, there are other uses now being made of such standards. A brief recapitulation will illustrate some of the more important uses.

1. They serve as a basis for wage-incentive standards so that the compensation is contingent upon accomplishment.

2. They serve as a basis for developing the operating budgets. These in turn give rise to more orderly production planning and better production control.

3. They serve as a basis for developing standard costs. These in turn give rise to more accurate product costs, cost control, and estimating cost of new product.

4. They give the supervisors, as well as the workers, a yardstick for evaluating the worker's performance. Workers will be able to detect their own deficiencies; supervisors will be able to spot low performance efficiently and take corrective action, as well as spot creditable performance.

5. They supply required data for redesigning plant layout, equipment, method, and product.

Ideally, the same basic standards would be used for all purposes, whether they be for managerial control, cost control, or for wage payment. Here we are concerned only with the use of standards as a basis for wage payment.

When an employee is hired, whether he is hired as president or janitor, there should be available to him a description of the duties of his job and beside each specific duty there should be an indicated means of measuring performance. The subordinate, as well as his supervisor, should be able to answer the question, "How am I doing?"

There is no doubt that a comparatively accurate analysis and description of most jobs can be made, but the question of performance measurement is more difficult. Two main problems are encountered in performance measurement; these are the ability to measure and the practicality of attempting measurement. Any analysis of these two points will yield a wide divergence of opinion and a host of conclusions. Circumstances that make measurement practical in one industry may be totally lacking in another industry. Conditions are dynamic—what is practical today may not be tomorrow. The practicality of measurement will have to be governed by the conditions surrounding a given case.

As our performance-measurement techniques improve, our ability to measure performance more accurately will increase. Certainly, the most widely accepted performance-measurement technique today is the time study. It has, however, many recognized and unrecognized limitations. Some of the time-study techniques are not equal to the occasion. That we have overstepped the bounds of the practical use of time study in many instances in the past is obvious. Witness the lack of uniformity among the time-study men in the handling of allowances for fatigue and delays or in the handling of time-study ratings. What few definitions there are of a "fair day's work" or "average operator" or "average conditions" are vague and dissimilar. Results of going beyond practical limitations can be seen in the numerous examples of wild incentive earnings of 200 or 300 per cent bonus, although such earnings frequently result from lack of proper analysis and method improvement prior to measurement. Wage inequities frequently result where plans are applied without the knowledge of a fair rate of production or where the work load for a particular individual has been changed but where the individual is paid on the basis of group performance. More time should have been taken in the majority of the cases to correct operating conditions first and then develop, sell, install, and maintain sound standards.

Standards Development. In addition to the two basic requirements for standards development and application, the ability to measure performance and the practicality of attempting performance measurement, other requirements for successful application would include

1. A study involving the use of methods-improvement techniques. The methods should be highly developed and standardized prior to standards determination.

2. Establishment of fair, consistent, and accurate standards by time-study techniques. Development of uniform time-study practices and standards-development procedures will do much to achieve the desired consistency as the proper relationship is frequently more important than absolute value.

The technique of time study depends upon a clear-cut division of an operation into its component elements which can be recognized, measured, and eventually used for synthetic determination of standards. The first consideration in time and motion study is the studying of an operation by studying its elements. The finer the breakdown into elements, the better each individual element lends itself to close analysis; therefore, an operation should be broken down as minutely as is consistent with accuracy and practicality of describing the element and making the observation. Where there are many rapid movements to be timed, any one of which is too short to be observed and recorded by stop-watch time study, the technique employed in "micromotion" study involving the motion-picture camera provides an accurate means of observing and recording such times. The next step is determination of the necessary elements, their proper sequence, and where, how, and by whom they should be performed. It is only after such analysis leading to standardization with proper methods that wage-incentive standards should be developed.

Established time-study procedure should be followed in order to achieve the most consistent standards possible. Conditions under which the study is made should be specified. Stop-watch techniques, including type of watch and method of recording elemental readings, delays, missed elements, foreign elements, elements out of sequence, and unusual conditions should be uniform. Although what constitutes a representative number of elemental readings is difficult to establish, good practice calls for continual assembly of data to determine when sufficient information has been obtained. Within reason, the greater the number of observations, the more intelligently will the engineer be able to determine the standard time. For practical purposes, the influence on incentive earnings of a given standard should be directly proportional to the accuracy in its determination.

The next consideration is that of leveling or time-study rating. After the proper methods and conditions have been established, there are still factors that affect output, *viz.*, the skill, or proficiency, at following a given method and effort of the worker. As effective effort varies among workers, it is necessary to evaluate to a common level of skill and effort the various recorded elemental time values. The common level most frequently chosen is called 100 per cent, average, or expected performance. There are numerous definitions of average performance, such as "that rate of output which the average, skilled worker, qualified for the occupation, given good supervision, and allowed adequate rest, should be capable of maintaining for an extended period." Other definitions of average performance include considerable detail as to the qualifications of the average worker, such as average skill, effort, physique, and intelligence with respect to other workers doing the same type of work. Leveling is usually accomplished by estimating the percentage of performance of the worker in relation to average performance and adjusting the actual time by the estimated per cent. For machine elements, average performance may be defined as the rate that a machine unit attains when operating at established feeds and speeds. The leveling factor, as well as the selection of average conditions under which the operation is to be performed, are considered nonscientific parts of the time-study technique because the factor of human judgment is present. As they are a matter of personal opinion, training and experience in their application are essential in order to achieve the required consistency. To minimize error in leveling, time-study practice calls for the selection of as near average workers as possible because the farther away from average the performance of the worker under observation becomes, the more difficult it is to determine reasonable leveling factors. The maximum practical range for leveling calls for performances within approximately 25 per cent of average.

The normal time represents the time in which a worker may be expected to perform the designated elements without any interruptions. Separation of "variable" from "constant" elements aids in the determination of normal time. Time for constant elements, which may be constant with respect to turn, job, setup, or piece, is generally obtained by averaging. Variable elemental times are most easily analyzed by means of graphs. In the determination of a normal value for an element, all readings for the element are examined and the abnormal readings excluded. Many statistical and graphical methods for determining the normal value are used, but the straight average is the most common. Here, as in leveling, the selection of the readings to be used and the method of determining the normal is subject to opinion.

To the normal time must be added a certain amount of time to provide for personal needs, fatigue, unavoidable delay, and miscellaneous work elements. The category of unavoidable delay includes machine and electrical breakdowns, power failure, material delay, interference, and crane delay. Fatigue incident to an operation is influenced by the physical and mental requirements of the operation and the environment in which the operation is performed. Personal allowances include time for trips to the washroom, etc. Separate elemental consideration of miscellaneous work elements such as minor setup elements, cleaning machine, grinding tools, and machine adjustments, which are nonrepetitive and occur infrequently, is impractical. Allowances for unavoidable delays and miscellaneous work elements are determined by time study, whereas allowances for fatigue and personal needs are usually arbitrarily established. All are expressed in per cent and applied to the normal elemental values to give standard elemental values. Total allowances generally fall within the 10 to 20 per cent range. The factors included, however, vary considerably.

Table 2-8. Typical Allowances

Personal	3- 5%
Fatigue	5-15%
Unavoidable delays	0- 5%
Crane delays	2%
Mechanical and electrical	1%
Miscellaneous work elements	0- 5%
Clean machine	2%
Grind tools	2%

Standards Presentation. The presentation of standards in the form most convenient for application must be so clear that there is no chance for misinterpretation. This requires an operation description of each standard shown in the incentive plan. Such detail is particularly necessary when the individual elemental standard times have been grouped for presentation. To simplify application, those individual elements which always occur together should be grouped together. Whether they are constant or variable elements has no effect upon the grouping, since an element that may be considered constant for a given repetitive operation frequently is a variable element under nonrepetitive conditions. The more repetitive the nature of the work, the greater the degree of grouping individual elemental standards, the fewer the number of standards presented in the incentive plan, and the simpler the application. For highly repetitive work, the individual elemental standards are grouped for presentation into a few job standards; whereas for nonrepetitive work, the individual elemental standards are presented separately. Obviously, the standards for nonrepetitive work need to be developed to cover a complete range of sizes, weights, or types.

Standards for variable elements are expressed in graphic, formula, or tabular form because of the necessity for covering a whole weight, size, or type range for a given operation. The development and use of graphs and formulas minimizes the need for studying similar operations performed for a whole range on a single or several similar pieces of equipment. Also, standards so developed tend to be more consistent. Equipment specifications are frequently used in conjunction with time study to develop elemental times. For variable elements expressed in tabular form, it is desirable to have the time values broken down to a uniform per cent difference or a preferred number series. For presentation of standards, tabular form is preferred to graphic form as it permits more consistent and accurate application.

When the standards are expressed in monetary values, the time per unit, if it exists, is multiplied by the expected earnings rate per hour for the occupation to give the standard in dollars per unit. From a control standpoint, standard-hour plans are more desirable than monetary plans as operating performance is automatically obtainable.

Temporary Standards. Whenever possible, predetermined standards (standards determined before the start of work on an operation) should be applied. When existing standards do not apply to an operation, standards for it may be estimated by a member of management or developed directly from time study or elemental comparison with similar jobs, standards for which have been set by time study. Under these conditions, temporary standards are frequently used until regular ones can be established. The purpose of temporary standards is to increase incentive coverage. Temporary standards may be set on new jobs for which methods are incompletely developed or on jobs for which methods are temporary. Some temporary standards are applied to small orders which occur infrequently. Such temporary standards remain in effect until conditions warrant establishment of a revised temporary standard or a regular standard.

LABOR PARTICIPATION IN INCENTIVE PROBLEMS

General Labor Attitude. Opposition to wage incentives is a traditional part of the attitude of some unions. The prevalent notion that restriction of production will *temporarily lengthen employment has frequently proved to be true. However, it is* well known that such a *practice keeps prices high, a condition that invites competition* and thus defeats the practice in the long run. Opposition directed at the malpractices of the wage-incentive system rather than at its use has also occurred. The attitude of the workers will change as the malpractices are corrected and as the workers realize that management is honestly trying to develop and advance them and improve their standing, thus benefiting both management and the workers.

Some unions have abandoned their unfriendly attitude toward wage incentives and are moving toward cooperation and a correction of the malpractices. Van Dusen Kennedy, in a recent book,[3] indicates the willingness of the United Steel Workers of America to approve incentives:

> The United Steel Workers of America . . . does not take an official position against any and all incentives. It has many grievances against various wage methods in specific situations, but its policy is to work for correction of abuses and for protection of workers through participation in the regulation of such methods; it considers complete abolition of incentives from the steel industry an unrealistic aim.

Some unions, such as the United Electrical, Radio and Machine Workers of America, have strongly advocated the use of wage-incentive systems. The UE has published a book on wage-incentive systems,[4] stating in part that:

> Effective industrial unions can turn incentive systems from detriments to benefits for their members by securing for them a share in the financial benefits which accrue from the employee's increased productivity.

Some unions, such as the International Ladies' Garment Workers Union, have set up their own management-engineering department to assist in improving methods, in determining fair occupational rates, in training union representatives in sound time-study techniques, and in the development of standards for the industry. Gomberg, Director of Management Engineering Department, ILGWU, writing on labor's frequent objections to wage-incentive plans stated that[5]

> The principal technical barrier . . . is the shortcomings of the time study itself. It is not my objective to deny the usefulness of the time study but to point out the limitations of its accuracy. Thus far, no alternative method of measuring or estimating a reasonable day's work has appeared. Hence, the time study must stand until a better technique is found. Unions which denounce time study and refuse any participation in its use must either furnish a superior alternative method of measurement of a day's work or appear completely irresponsible.
>
> Acceptance of time study, however, does not mean acceptance of all the claims of its partisans, practitioners, users, and abusers. The profession, unfortunately, has failed to police itself. It has been content to denounce malpractice in general.

Fear of wage incentives on the part of the workers results from many beliefs, such as fear that the standards require a pace that will be injurious to health, that the standards will be cut, that failure to meet standards will bring criticism, that incentives will cause unemployment, and a general lack of understanding of standards development and incentive-pay determination.

Management has been reluctant to relinquish to labor its right to set and install standards. This is recognized by most unions; however, labor reserves the right to question standards through grievance procedure. Enlightenment on the part of both management and labor and the general labor movement toward cooperation will tend to alleviate both labor's fear of the incentive plans and the malpractices.

Negotiation and Grievance Procedure. The union contract that provides no grievance procedure for settling disputes arising during the contract term is now rare. Although a large and aggressive union usually has a more elaborate grievance procedure than a small union, discussion of grievance procedures in union contracts may be divided into four parts.

One part deals with the nature of the grievance. Most contracts specify the subjects that may be handled under the grievance procedure, such as differences arising over interpretation, application, or compliance with the provisions of the agreement or questions relating to hourly wages, wage incentives, or other conditions of employment. Such provisions are broad and permit discussion of almost any type of grievance except disputes involving demands for changes in terms of the agreement.

A second part deals with the manner in which the employee is to be represented. Agreements frequently differ in the extent to which the aggrieved employee participates in the processing of his grievance, although union representatives must by law be given an opportunity to be present. Frequently included are the number and method of selection of the union representatives, the protection allowed the representatives, such as seniority, and special features including remuneration and time allowed for processing disputes.

The third and most extensive part pertains to the procedure channels. Outlined in detail are the various steps for processing the disputes including the form and method of grievance presentation, the scheduling of time and place of meetings, the union and company representatives participating in each step of the procedure, and the channels to be followed in case of lack of settlement at any given point. Procedures usually call for four or five steps.

The fourth part covers the terminal point of the grievance procedure, which is almost always some form of arbitration.

Development and Installation. Joint discussion with labor representatives and the employees affected of the need for incentive application, the type of earning curve, the employees to be covered, and other important features including establishment and presentation of standards will bring about better labor acceptance of incentive wage-payment systems. Cooperation from labor should be obtained prior to the development of the plan. There seems to be a high degree of correlation between the extent of the standards presentation or willingness to deal with labor by the company and the freedom of management to set standards. Management is willing to explain new standards in great detail when assured of a receptive attitude on the part of labor and when assured that the proposed standards will not be subject to bickering, and vice versa.

The installation of an incentive plan is primarily a selling job and one which does not stop with top management or labor representatives but which must be sold to all workers affected. Before the workers will accept a plan, they must understand it, and therefore it must be simple enough to allow easy calculation of earnings by the workers themselves. Standards and sufficient explanatory data always should be available to each worker for calculation purposes.

Before a plan is made effective, it should undergo a considerable trial period in order to test its soundness, accuracy, completeness, and expectations. The results of such a trial run should be analyzed by both the workers and management. Regardless of the care taken in development, a trial-run period will usually indicate conditions not covered by the plan in its existing form. Although procedure is included in the plan for future revisions and additions, such additions and changes can be made more readily before a plan becomes effective. When revisions and additions are necessary, joint cooperation and understanding should be obtained to the same or greater extent than for original installation because of labor's fear of change.

Union-contract Incentive Detail. Because of labor's increasing participation in wage-incentive problems, the development and presentation of wage-incentive plans is becoming more formalized. The trend has been away from the inclusion of monetary piece rates in the company wage scale or the use of fly sheets to cover unusual conditions toward the use of a formal incentive-plan manual which includes, in addition to the standards, method of application, policies covering delays, equipment changes, etc. At present, practically all union contracts contain a section on rates which covers occupational rates and occasionally rates for delays and faulty materials, etc. A few of the more recent contracts include, in addition to rates, time-study procedure, allowances, use of temporary and regular standards, and changes in standards.

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ADMINISTRATION OF JOB-EVALUATION PLANS

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Job evaluation is a natural outgrowth of the development of scientific management. It is an attempt to reduce the complexities of an occupation to their fundamental components and to measure and weigh each component so that the sum will give the relative economic worth of that occupation. Let there be no mistake—no workable job-evaluation plan can measure a job as one may measure the dimensions of a room. One of the greatest causes of failure of these plans is a blind adherence to the system and the evaluations as they have been set up. In spite of the inaccuracies inherent in job evaluation (which will be pointed out later), it is still the most equitable method of dividing the pay roll which has yet been developed. If it does nothing else, it provides a framework for logical thought when assigning wage rates. Job evaluation is not an easy way to set wage rates, and there is no easy way to evaluate jobs. Painstaking and thorough research is required. Plans have been proposed as easy methods of job evaluation, but there is no easy cure for the inevitable disasters that will result from slipshod methods of analysis. No conclusions can be more accurate than the observations and data upon which they are based. If these are inaccurate or inadequate, the wage structure built from them will be intrinsically unsound and hopeless to administer.

The job components and their relative weights are of secondary significance to the care with which they are applied.

It is necessary in establishing a system of job evaluation first to define the objectives of the program. These objectives should be basically the following:

1. To describe and define completely the mental and physical requirements, the duties, and the responsibilities of every hourly rated occupation
2. To establish a ranking or grading of all occupations according to some breakdown of the skill, working conditions, and responsibilities that are required by each occupation
3. To establish certain key jobs with known market rates and from them to establish a wage-rate structure that will most equitably distribute the existent pay roll among the various occupations
4. To develop administrative procedures for evaluating new and changed jobs as changes occur and to maintain a close relationship between internal wage rates and the market-rate ranges for the same and similar jobs

Job descriptions are the heart of any program of job evaluation. It is that phase of job evaluation which is most frequently slighted and is most frequently a cause of future trouble. During the Second World War, especially, there was a tendency to establish these programs in a hurry. As a result, a great deal of job describing was done by filling out printed forms with short one-word answers. These forms generally follow the outline of the points in the actual job evaluation. Real job descriptions should be obtained by trained interviewers without a highly standardized list of questions to be asked. The interviews should be in the form of leading questions which encourage the worker to describe the true characteristics of his job. There are only a few general points to be covered, but they should be covered in detail. The resulting information should reveal what the worker does, what he has to know to do it, what material and equipment he must use to do it, what mental and physical exertion is required, and what his responsibilities are. These descriptions should be checked and rechecked for accuracy by foremen, supervisors, plant management, and industrial engineers. As they usually bring out phases of jobs that previously no one was aware of, the jobs will normally require a reorganization and possibly an elimination of certain phases of many jobs. Many jobs may be simplified, and some overlapping classifications should be entirely eliminated at this time. After this has been done, the evaluation plan to be used may then be selected.

The most popular type of plan today is the one which establishes a point valuation for the various predetermined job factors or characteristics. This type of plan is fairly easy to administer, and if the number of factors is fairly small, it is simple to explain and easily understood. The maximum number of factors which should be used is about eight. Some plans have been proposed that use as high as 20 different factors. Doing this does not add any real accuracy and in many cases only serves to remove attention from the important characteristics and to make agreement more difficult.

There are instances of some very successful plans which use no more than three factors. Table 2-9, which lists factors with their weighting, is a reasonable sample.

Table 2-9

1. Experience	21 %
2. Education . . .	18 %
3. Initiative and judgment	12 %
4. Dexterity and precision	12 %
5. Physical demand	12 %
6. Mental and visual demand .	6 %
7 Responsibility for equipment or process	12 %
8. Responsibility for others	7 %

When the plan to be used has been decided upon, certain key jobs are selected. These key jobs must have fairly definite market rates. They are evaluated and plotted on a chart of wage rate vs. point value. By statistical methods, a trend line is then drawn through these points. If some of the original key jobs are found to be too far out of line with the others, they should be discarded and others selected. When the other jobs have been evaluated, their wage rate or wage range will be established by this curve. The resultant wage structure should not cause a potential lowering of total pay roll or an appreciable increase in the present pay roll. This is based upon the assumption that the wage rates of jobs which fall above the curve will not be lowered immediately and those jobs whose rates fall below the curve will be raised to the proper level.

The foregoing is merely an outline of the general method of applying a job-evaluation plan. A great deal of literature is available on the actual methods of evaluation. However, very little has been written on the problems of administering such a system. Frequently, proponents of job evaluation prefer to ignore the existence of these problems. If job evaluation is to be utilized to its fullest advantage, these problems must be recognized and policies formulated to solve them. Poor administrative policies are responsible for the failure of more systems than a poor choice of factors or weighting of these factors.

MARKET RATES

One of the most important decisions to be made is determining the role of market wage rates. In some plans the effect of the labor market has been included as a factor to be evaluated. This procedure is not satisfactory in that it distorts the objective rating of job content or else it carries so little weight as to be of no value. It is also impossible to evaluate availability or market rates of skills that are unique to one plant. Therefore, a factor "market" is not common to all jobs.

A definite policy must be established as to whether the wage-rate structure is to be based primarily on internal standards or market rates. If the company is dependent upon a competitive labor market, some provision must be made for allowing rates to reflect going market conditions, or troubles of many types are bound to ensue.

Chronologically, the first problem to be met is what to do with the key jobs which were used to establish the wage curve. The key jobs will be located on a chart of point values vs. wage rates. The wage curve will then be drawn through them in such a way that the weighted quantity of those below the curve will be equal to the weighted quantity of those above the curve. The curve will also have to be drawn so that there is no net decrease in potential or actual pay roll if union acceptance is to be hoped for. It can be seen therefore that few, if any, of the key jobs—those with determinable market rates—will be on the curve. If internal standards are to prevail, these jobs must be either raised or lowered so that they do fall on the curve. Under present conditions of inflationary wage pressures, if the plant is unionized, all wage rates that are below the curve will undoubtedly have to be raised if the union is to accept the principle of job evaluation at all. This can, however, have an unfortunate effect upon the workers involved on the job. In some instances, it may be possible for the company to eliminate that work and contract it out to other firms paying the market rates and actually reduce the costs. This is most likely to occur in the service and maintenance functions. If it is not possible to eliminate these operations, the company will be put in an unfavorable cost position in relation to its competitors.

If, on the other hand, wage rates of key jobs above the curve are lowered to the curve, this will result in trouble in employing people to fill vacancies for those jobs or work stoppages and slowdowns.

The greatest trouble occurs where market rates of jobs change after a job-evaluation system has been in operation for some time. In order to avoid personnel troubles, wage rates must vary with important variations in market conditions. There are five general ways in which this may be done.

1. Reevaluate the job adding values where none exist in order to get the desired point value.
2. Apply an incentive bonus to an hourly job or relax standards on present incentive jobs.
3. Change the slope of the wage curve in order to decrease or increase differentials.
4. Keep knowledge of the existence of a job-evaluation system from the workers.
5. Admit by policy that market rates are more fundamental than internal standards.

The practice of reevaluating jobs to increase their point values in order to correspond to market rates is in common use. Although the problems it creates are, in the long run, more dangerous than the original problem, it is, in many cases, the only solution. This procedure, known in popular terms as "screwing the rates," will have the immediate effect of raising the job rate to correspond with the market rate. One representative of labor, however, admitted that usually they have a difficult time finding those last few points to raise the job another classification. The long-run effects of this type of action have a tendency to undermine the whole system. Each job that has been tampered with in this fashion is sometime going to be questioned by someone in the plant. Some worker is going to wonder why this job which is easier than his in all respects gets paid more money. That question is very difficult to answer. If enough of these jobs are present in a plant and enough of them questioned and found to be at fault, worker confidence may be so undermined as to require scrapping of the whole system and starting over again. That is a long and expensive process, and the entire procedure will not tend to improve relations between labor and management.

The second method, that of applying incentive rates to hourly jobs in order to increase wages, has fundamentally the same disadvantages of "screwing the rates" except that here it is not necessary to tamper with the actual point values. This protects the job evaluation from the brunt of the criticism from the workers. An example of this is the trouble one company had with a barrel-washing operation. The workers finally staged a wildcat strike in order to obtain their demands. Management satisfied them by applying an incentive rate which enabled them to increase their daily earnings by more than their original demands. The primary reason for the low rate was the weighting of the factor "working conditions" which was satisfactory for other jobs in the plant but did not allow sufficient points for this job. Because the job is dirty and unpleasant nobody wants to take it even at an increase of pay, and as yet no trouble has ensued from this action. This is not always the case. Frequently, during periods of a tight labor market it is desirable to increase facilities and hire many new employees, as, for example, during a war. In the same company, it was necessary to establish a new department performing simple, light repetitive work, suitable for women. In order to hire the necessary personnel and where a strong and well-integrated horizontal union is in operation. In such a case, job evaluation can have no effect in determining wage rates.

It is desirable from the standpoint of the company and the union that market rates be followed as closely as possible consistent with a stable wage structure. Under almost no conditions is it desirable to have a wage structure that is sensitive to short-run (less than 1 year) variations in wage rates. Inequities would be introduced to last for a long period if rates were raised on jobs that showed only temporary rises in market price, if the policy of not lowering rates for present incumbents were followed. If this policy were not followed (which in a strongly unionized plant is not likely to occur) the wage structure would be so unstable as to have an extremely detrimental effect on worker morale. A more satisfactory solution is to watch key jobs for significant changes which will have more than transient effect on the wages of the area.

JOBS "ABOVE THE CURVE"

Most of the literature on job evaluation mentions the handling of rates that are above the curve. However, very little concerns some very serious problems that arise from the principles. It is a generally accepted principle that all rates below the curve will be raised to the curve as of the date the plan is instituted. Frequently these raises are made retroactive to some prior date which has been agreed upon as the effective date of the job-evaluation plan.

The principles for handling rates above the curve, or "red-circle" rates, are as follows:

1. Wage rates of present incumbents shall not be lowered.
2. The rates of these jobs may be lowered by the following methods:
   - *a*. All workers subsequently hired for these jobs will be hired at the curve rate.
   - *b*. Workers on these jobs may be transferred to other jobs which carry the wage rate they now receive.
   - *c*. Normal turnover will slowly replace all present incumbents on these jobs.
   - *d*. Job content may be changed on these jobs thus substantially creating new jobs and eliminating the old ones.
   - *e*. Workers on these jobs may not take part in general wage increases until the inequities are ironed out.

One of the biggest problems is the definition of "present incumbent." One company, which had to go to arbitration to establish a wage curve, made an agreement with its union that the effective date of the job evaluation was the date that the petition for arbitration was approved. It was agreed that retroactive pay should be based from this date, and also present incumbents were defined as people hired before that date. Because the negotiation and arbitration extended over a period of about six months, many people were hired during that period for jobs that were later decided to be above the curve. On one job about fifteen per cent of the workers had been hired after the effective date and could not be counted as present incumbents. The situation was explained to these workers by a representative of management before their rates were cut, and it was accepted with good grace by all involved.

In another case present incumbents have been defined as all persons hired during the present contract year. This was done to put off the problem of hiring new workers at the curve rate to work with men who are getting the old higher rate. There are generally two unfortunate results to this type of situation. The old worker feels that the new man is "scabbing" on him, is out to cut his throat and take his job away from him. The new worker can't understand why he is doing the same work as the man next to him and doing it just as well, but getting only a little more than half his pay. No matter how well the reasons for this are explained there will usually be dissatisfaction on the part of one or both of the parties. The extent to which this dissatisfaction is generally made apparent will determine how successfully this situation can be handled. This company is anxious not to create this problem, at least during the present contract year. At present, with the continuing pressure for higher wages, it is not unlikely that this policy will be continued for the next contract year.

Another problem arises in defining present incumbent where promotion sequence is involved. One of the generally accepted advantages of job evaluation is that it defines more exactly the sequence of promotion and provides job incentives. If, for example, a group of jobs in promotional sequence are all found to be above the curve, how can a satisfactory definition of present incumbent be made? If only those persons working on the job at the time are to be present incumbents, why then the man on the next lower job will find that, if he makes his promotion, he will have to take a reduction in pay or at best no increase in pay at all. Under those conditions, he probably would not accept the promotion and would be extremely disgruntled about the whole affair. Under these conditions, the most satisfactory solution is to include all persons working at any one of the sequence of jobs as present incumbents for all of them. With this sort of policy, great care must be taken to define precisely the series of jobs which can be included in the promotional sequence. If this concept is accepted, it is evident that promotion must be made entirely within the framework of these jobs. This is probably a desirable thing to do in any case. This group of jobs will be in the class of "red-circle" rates for a much longer period than is generally accepted as being satisfactory, but in spite of this, this solution is the only satisfactory one in many cases.

The first method of lowering the rates of red-circle jobs, that of hiring all subsequent workers at the curve rate, is, of course, dependent upon the definition given for present incumbents. The problems involved have been discussed above and are sometimes of major importance. The union of one company has suggested that the only suitable time to bring wages down to the curve is during a general increase or decrease in the over-all wage level. However, in most cases, this principle is accepted if the union is honest in its belief that these rates should be brought down to the curve.

In order to hire workers at the lower rate, vacancies must be provided in these jobs. One method of providing vacancies is to transfer workers to other jobs which carry the same rate as that which they are now getting. In cases where the skills involved are of a general nature and the work is not radically different, this may take care of a high percentage of these people. If the job is one which requires a unique skill which is not required in other parts of the plant, this policy may offer no solution. If the work on the new job is very different in its nature or the environment less attractive, workers may be opposed to the transfer. At any rate, workers will generally be opposed to enforced transfers if they are fairly content on their old jobs. All those workers who for one reason or another cannot be transferred to other jobs will eventually be replaced through normal turnover. Where turnover is slow, this may be a long process, but it is eventually as sure as death itself.

In some cases it may be possible, by a change in process, to eliminate the job entirely or to change it so that a reevaluation is necessary. If the union admits that jobs which have been changed should receive the new rate as determined by evaluation, this procedure will also give management an added incentive to modernize and improve existing methods and processes. In some cases, however, this is impossible without completely changing all existing manufacturing procedures. This is especially true in the chemical industry where changing processes is sometimes actually impossible at the time, or investment in equipment and setup for present methods prohibits any changes.

The fifth method, that of excluding workers whose rates are too high from general wage increases, can be quite satisfactory if it is used with discretion. Management should adopt the attitude that their wage curve represents a long-run goal and should not insist on excluding these workers entirely from increases for too long a period. The workers whose rates are above the curve will usually accept without question the initial raises which were given to bring the low jobs into line. However, there may be trouble with subsequent raises if they are not given some share in them. One company suffered a 6 months' strike because of this very principle and was finally forced to grant the general increase across the board equally to all employees. It is quite possible that the strike would not have occurred if the company had been content to grant a portion of this increase to red-circle rates even though there would have been a less substantial decrease in intraplant inequities.

It has been argued that it is extremely difficult to bring jobs into line during a period of rising wages, the upward pressures being too high to allow segments of this wage structure to be held back. One union felt this so strongly that it was included as an issue in a case before the War Labor Board, although there was no possibility of a general increase at the time. They contended that the only way that wages could be brought down to the grade maximum was through death or separation from the company. The company felt that these jobs should not share in general wage increases. The issue was referred to the parties for negotiation as no wage increases were imminent. This is indicative of the fact that strong opposition to this procedure may be met.

A final method of getting around the problem of rates above the curve, but which is not to be recommended, is that of building the job description and point values around the existing rates. If this is done, the same results will occur as when rates are tampered with to correspond to market conditions. One company,[*] which did build its job evaluation around the existing rates, found the union was soon demanding reevaluation of all jobs or a complete scrapping of the whole system. When this happens, even if the plan is salvaged, it will be suspected by the union as a whole and will be, for that reason, less valuable. This procedure will, in the long run, create more problems than the red-circle rates.

There are two reasons why it is undesirable to retain these high rates. The first is that the company is forced to pay a higher total pay roll than would otherwise be acceptable to labor. The second is that these high rates will, as long as they exist, be a source of confusion and dissatisfaction to the workers. It was the experience of one company,[†] which had to retain these rates over an excessive period, that the union requested an entirely new job-evaluation plan. They claimed this was necessary because many jobs were higher than the maximums by from 2 to 31 per cent, which created inequities which they could not explain to the union body. If these high rates exist for too long a time, their original meaning will be lost sight of, and the union leaders will have difficulty in convincing the union body that there is a reason for them. Their continual existence will disturb the functioning of the job-evaluation system.

The only way this problem can ever be completely avoided is by granting a general increase at the time the system is established. This serves to lift all jobs up to the same relative level as the too-high jobs. This method could not be accepted by management, however, unless such a general increase was imminent anyway.

TECHNOLOGICAL CHANGES

Decreasing the rates on jobs that have been simplified by some change in methods has been objected to on the grounds that workers should be paid on the basis of productivity. This contention is diametrically opposed to the principle of paying on the basis of the skills required and the arduousness of the work to be done. The concept of pay based on productivity will break down logically in the extreme cases. Certainly, the operator of a highly automatic machine, who is required to have no special skill beyond pushing a button, should not be paid more than a highly skilled tool- and diemaker even though the daily output of the button pusher may far exceed in value the output of the machinist. If this policy were followed out, it would lead to the development of many highly paid sinecures. This is certainly not a goal to be desired. Jobs that have been simplified, even though their productivity may have increased, should have their pay rate decreased. The savings should be distributed to the workers as a whole in the form of general increases or to the general public in the form of reduced prices. The changed job should be paid on the basis of job evaluation.

If it can be agreed by union and management, the most satisfactory policy is to lower the rate on the job as soon as it has been changed and the new evaluation has been made. This is frequently difficult to manage if the job is changed only slightly and there is no change of personnel doing the work. If this is the case, it is frequently necessary to call the job a red-circle job and handle it by some method as outlined previously. In some instances, however, it will be possible to lower the rate immediately and for those people who have been working on the job all along. This can be established only by agreement between union and management. Another company has had considerable success with the policy of adding new duties to the job when it has been simplified in order to increase its point rating to the original value. If this is not practicable, the company increases the worker's merit rating on the basis of the residual skills the man has but is not using. In a small company, where the problem of replacing absentees is quite troublesome, these residual skills represent a form of insurance to the company which they feel they are warranted in paying for. In plants where a merit rating is in use, this solution has practical value and is theoretically sound.

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[*] Cuno Engineering Corp., Meriden, Connecticut, and United Automobile Workers of America, Local 753, AFL Case No. 111-13250, Mar. 31, 1945.

[†] Brown and Sharpe Mfg. Co., Providence, Rhode Island, and The International Association of Machinists, Brown and Sharpe Lodges 1088 and 1442, Case No. 13-447-D, Nov. 27, 1944.

If job evaluation is to have current significance, it is necessary that no changes be made in job content without notifying the parties responsible for reevaluation. This can best be accomplished by the use of three policies or practices:

1. Close cooperation of department foremen and methods department with the job-evaluation committee
2. Rapid channeling of wage grievances to the evaluation administrator
3. Periodic audit or check of the descriptions of all jobs

Understanding of the significance of job evaluation and the role it plays in the company's wage structure is a necessity for all foremen. They must be trained to watch processes closely in order that job content may not be changed without their knowledge. Whenever a new job must be introduced or a present job changed, the job-evaluation administrator must be notified immediately. It is usually the best policy to establish a temporary rate until the job content has been permanently established. Retroactive payments will have to be made when the temporary rate is set too low. If job evaluation is administered entirely by management, it is probably best to have the job-evaluation administrator set the temporary rate as he is most familiar with the other jobs in the plant. If this is impractical, it would be satisfactory to have temporary rates set by the industrial engineering department if they are separate departments. In companies that have efficient methods engineering and where processes are tightly controlled from a standpoint of quality control or incentive rates, there should be little difficulty in quickly spotting any changes in jobs. Where this is true, it is advantageous to have job evaluation very closely tied to methods engineering if not actually in the same department. When job evaluation is handled by the personnel department, especially in a large organization, it is liable to become too far divorced from the actual manufacturing operations which it is evaluating.

The handling of wage grievances should be greatly simplified by the establishment of a job-evaluation plan. Wage grievances should be channeled directly to the job-evaluation committee, whether or not it operates under the personnel department. They should be treated differently from other types of grievances. It has been possible in some plants to establish a time, to occur at regular intervals, when grievances, or requests for reevaluation, may be submitted to the job-evaluation committee. This policy will have the effect of decreasing total grievances as it will automatically give the worker a "cooling-off" period. If labor-management relationships are not in a strained state, where the union is hunting for grievances, this should be agreeable to the union. When relationships are smooth, individual wage-rate grievances are as much a headache to the union committee as to management because, if one man gets more, it means that another is getting relatively less. The alternative solution is to accept wage grievances at any time. This has the advantage of not allowing inequities to exist after they are discovered. However, if the times that grievances are accepted occur frequently, this advantage is of doubtful significance. The form of the grievance slip is important. If an agreement is made by each worker that the reevaluation will be accepted even if it is found that the rate is already too high

there will be added insurance that the grievances will be legitimate. If job evaluation has the backing of the union, this principle will probably be accepted.

Application for Job Reevaluation

We, the undersigned employees now working on the job, wish to have the job evaluation reconsidered on

____
(Name of job)

for the following reasons ______________________________

____

____

____

____

____

In requesting this reconsideration we understand that there is an equal possibility of a reduction as well as an increase in evaluation and we agree to abide by the findings of the Job-evaluation Committee even though the reduced evaluation places the job in a lower wage-rate group

Signed ______________________________

Grievances will be of two general types:

1. The job was incorrectly evaluated originally
2. The job has been changed in such a way that the point value should be increased

In either case, care must be taken in the reevaluation to see that it is done accurately and objectively and that it is not influenced by pressures from the group presenting the grievance. It is in situations of this type that joint union-management job-evaluation committees display their inherent weaknesses. Grievances that are claimed to be based on changes in job content may actually be based on external conditions such as market rate, and under these circumstances, it is easy to read new values into a job which are not actually there. The support of really complete job descriptions makes it easier to avoid error in such circumstances.

To ensure the complete contemporaneousness of job evaluation, some sort of periodic review or audit of job descriptions is necessary. This may be done piecemeal throughout the year or all at one time but should be accomplished for each job at least once each year. This procedure will catch minor changes before they become serious and require drastic changes in the rates for any one job. This procedure is especially necessary where control of standards is fairly loose and changes are less likely to be caught by foremen or the methods department.

UNION PARTICIPATION

The basis for employee participation in job-evaluation plans is well expressed by Eugene Benge who said, "Success depends not only on selling the program to employees, but also upon the extent that employees believe in it. That belief is greatly enhanced by the opportunity to participate."[2]

The extent to which employees should participate is not a matter that can be decided categorically. The degree of participation should be decided upon the basis of the attitude of the workers and their general level of education and intelligence. A policy that may be quite successful in one case may fail completely in another. In one case where management had to deal with 14 different trade unions, a plan was set up independent of the workers, purely for management's information. If some sort of employee aid and participation is not enlisted, this will be the degree of effectiveness of most plans.

The degree of union participation may vary from the above type to a plan in which job evaluation is a joint function and responsibility of union and management. The most important weakness of this sort of plan is that job evaluations will be, to a large degree, negotiated, and there will be a greater tendency to attempt to fit the evaluation to the present rate. For example, in one company with this type of plan, no rates were found to be above the curve when the plan was established. It does not seem to be within the realm of possibility that this could happen with a fair and objective evaluation of all jobs. The evaluations were evidently built around the existing rates. The dangers of this action have been discussed under Jobs above the Curve. It is probable that the actual job evaluations are changed to reflect group pressures and market conditions as no red-circle rates of any kind are in existence. The second disadvantage is that union members of the evaluation committee may be removed by the union body. This leads to instability on the committee and has a tendency to force the union members of the committee to influence their decisions by what they feel to be the desires of the majority of the workers. This is true no matter how objective they may try to be. The presence of these rates is a continued threat to the very existence of the job-evaluation program.

If these disadvantages make a joint union-management plan unsatisfactory, to what extent should union participation be allowed, and what should be the role of collective bargaining in determining wage rates? In the long run, unions will not allow job evaluation to remove individual wage rates from the realm of collective bargaining. On the other hand, a typical management attitude is exemplified by W. H. Frater, Director of the Work and Wages Analysis Section of General Motors Corporation, who states that negotiation of individual wage rates will prostitute any job-evaluation program and nullify it in so far as the establishment of a rational wage structure is concerned.[3] This attitude assumes that job evaluation can measure and rank jobs with absolute accuracy and that labor must accept a rate for all time once it has been established. The labor movement will not admit that the plant's wage structure can be a unilateral decision by management and will reserve the right to change its mind concerning ranking of jobs and wage differentials.

In reality, there should be no fundamental conflict between collective bargaining and job evaluation. The actual point evaluations should not be subject to negotiation; the dangers of this have been pointed out. The job-evaluation program should provide common ground for the discussion of individual wage rates. Even without the existence of a job-evaluation system, rates are negotiated on the basis of what the worker must do and what he must know in order to do it. Job evaluation presents these facts completely and rationally and should make it much easier for union and management to come to an agreement on the pay for individual jobs. Job evaluation provides a framework for negotiating the rates of unique, new, and changed jobs in the company. Job evaluation should not attempt to establish, by management fiat, the rates of all jobs in the plant. In setting up a job-evaluation program, management and the union should agree that the actual evaluation of jobs is the prerogative of management, subject to normal grievance procedure.

In spite of the fact that the evaluation should be done by management, a certain amount of employee participation should be incorporated in the establishment of the program. The procedure of job evaluation and what it means to the individual worker should be explained, using terms and examples that will be understood by the average employee. Lengthy dissertations on point-valuating procedures and factor weighting will probably not be understood by the average employee and may serve to make him antagonistic. It is necessary only that he be instructed in what the plan is designed to accomplish and its general method of operation. The practice of questioning the worker on the job as to what his work actually involves will not only yield information but will increase his interest in the program. All efforts should be directed toward convincing the employee that the plan is being fairly and objectively administered and that it is to his benefit that a wage structure be established on some rational basis.

CHECKING RESULTS

In any phase of business activity, it is important to have some way of checking the results of expenditures of effort and money. In the field of industrial relations, this is a difficult thing to do. It is next to impossible to determine the effect of job evaluation on the yearly profit-and-loss statement. The only way that the results of job evaluation may be tested is by examining its effect on labor-management relations and its acceptance by the rank and file of workers. Comparing labor turnover and wage-rate grievances before and after job evaluation will usually provide a general idea as to its effects.

There are many characteristics of labor turnover which could invalidate any data which were gathered from these figures. First, it is usually impossible to find out why people quit a job. They may just never show up or they may actually lie when asked why they are quitting. Second, there are too many reasons that they might quit which have no relationship to job evaluation, and lacking knowledge of their reason for quitting, it is unsound to make any assumptions as to the effectiveness of job evaluation based on these figures. The soundness of a single rate may be determined with more accuracy than the whole structure, but even here the results are doubtful.

An examination of the amount and types of grievances is a better indication of the effect of the rates established, but even here there are many other factors that affect grievances other than wage rates. If union-management relations are better, shop stewards will discourage grievances. This change in attitude may be caused by a general wage increase or some other factor not even remotely related to job evaluation.

As these procedures are merely methods of determining worker reaction, probably the best method of determining this is by asking the workers themselves. If this seems unsuitable, it is possible to get the reactions of foremen and other supervisory people who are in a position to know the degree of worker acceptance. The union committee will probably be anxious to air worker reactions if they are not favorable. It should always be remembered that the only criterion of success of a job-evaluation system is whether the resulting rates are acceptable to labor.

CONCLUSIONS

Job evaluation, if properly used, can be an extremely useful tool of management. However, if jobs are hastily and inaccurately evaluated and described or the system haphazardly administered, it can become an expensive headache. The most basic pitfall to avoid is the belief that any job-evaluation plan known today can with absolute accuracy measure the relative worth of any group of jobs. Job evaluation can measure only those factors which are common to all jobs in some degree and which can be described and measured in concrete terms. Even those factors which are generally included are frequently next to impossible to evaluate rationally. The problem of measuring responsibility is one which has never been satisfactorily solved. At a job-evaluation meeting in one plant, a foreman gave his idea of responsibility: "Two men drive trucks of identical make, year, and model down the same road the same number of miles each day. One has a dump body and is loaded with sand; the other has a bus body, is used as a school bus, and is loaded with children." The difficulty of measuring the responsibility in this case and applying it to a wage differential is obvious. The foreman's final comment was that the man hauling the sand probably would be paid more anyway. There is still something to be desired in obtaining a really objective and adequate measurement of the relative worth of jobs. Therefore, it can be seen from the inherent inadequacies of any such plan, job evaluation should not be the ultimate authority but a guide in setting wage rates.

The following general policies will usually give the most satisfactory results:

1. The plan itself should be as simple and flexible as is consistent with reasonable accuracy. Wage-rate ranges for a scale of point values should be established rather than a single-rate system which gives a different rate for any small variation in point values. The accuracy of any system does not warrant using the latter method.

2. Because of the importance of going market rates, they should be recognized as more fundamental than internal standards, and suitable agreements should be drawn up to handle market variations. For this reason, job evaluation will be most valuable in setting rates for unique, new, or changed jobs.

3. Job descriptions and evaluations must be kept up to date. Labor must allow simplified jobs to be reevaluated and the rates eventually lowered if the plan is to

succeed. Close cooperation of industrial engineers, foremen, and the job-evaluation administrator, in addition to at least annual audits, is necessary to maintain current data.

4. Job evaluation should not exclude the possibility of negotiating wage rates but should provide information so that collective bargaining may be carried on more intelligently. Point values themselves should not be negotiated as this will result in compromised values with the resulting loss of objective point ratings. Management should establish rates and then, if they are found unacceptable to the workers, should bargain collectively over them.

5. The effective use of job evaluation requires a cooperative attitude on the part of labor. The plan must be sold and its method of application agreed upon before it is put into effect if it is to be of greatest service.

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SECTION 3

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FOUNDATIONS

BY F. W. STUBBS, JR.

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This section calls attention to some of the important problems arising in the design and construction of any structure, *viz.*, the foundations. At the outset, it may be well to suggest that, because of the importance of the foundation to any structure, an engineer competent in this field be selected to design and supervise the construction of all foundation jobs. Experience with the problems is essential. (This statement applies elsewhere.)

The essential problem in the design of foundations for buildings, bridges, towers, stacks, etc., is so to proportion the foundation that the settlement will be within the permissible limits. Actually the effect of the structure on the foundation material is considered. The problem of runways and taxiways of airports, highways, parking lots, and storage areas is similar, and in addition, the effect of the foundation material or subgrade on the wearing surface of such projects must be considered. The wearing surface must be so designed that applied loads do not cause failure of the subgrade. Also, the subgrade must be so selected that it will not cause failure of the wearing surface through frost action or other causes.

Structures may be divided into two parts, the superstructure and the substructure. That portion of a building above ground is considered the "superstructure," and that portion below the ground is called the "substructure." Likewise the trusses and deck of a bridge are called the "superstructure," and the supporting piers and abutments are generally referred to as the "substructure." The steel frame of a transmission tower is the superstructure, and the concrete footings are the substructure.

The substructure is commonly made of concrete, plain or reinforced, steel, or timber. That part of the substructure which transmits the load to the soil or rock is the foundation and in some cases may be called the "footing." The material on which the foundation is placed is called the "foundation material." Those portions of the substructure of a building above the footings are known as the "foundation walls." The substructure above the foundation of a bridge is the pier, abutment, or bent, although in some cases the various parts merge into one, and no distinction can be made, as in a pile bent. In highway, airport, and similar construction, the term "subgrade" refers to the foundation material on which the wearing surface or pavement is placed. In some types of construction the wearing surface merges into the subgrade, and there is no definite line of demarcation.

SOIL AND ROCK

The terms "soil" and "rock" may have different meanings depending upon the profession or field in which they are used. Geologists refer to the solid portion of the earth's crust as rock, using the more restricted terms of bedrock for the consolidated material and mantle-rock for the unconsolidated portion. The engineer refers to the unconsolidated portion as soil or earth and to the consolidated portion as bedrock. In agriculture, soil is used to designate that thin layer of unconsolidated material which supports plant life, and subsoil is used to designate the underlying material.

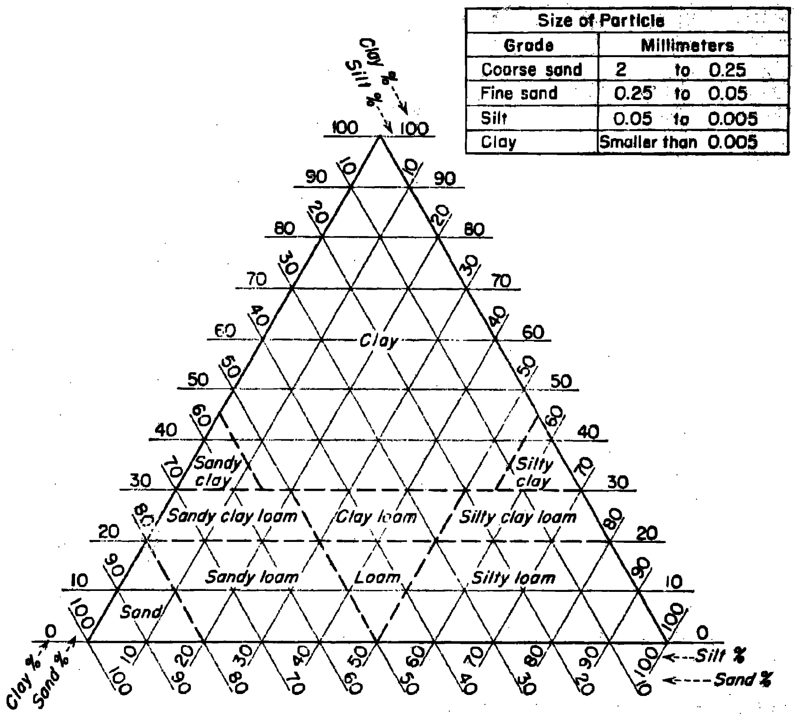
Table 3-1

Gravel . .	Over 2.00 mm
Sand . .	2.00 mm 0.05 mm
Silt	0.05 mm 0.005 mm
Clay	Smaller than 0.005 mm
Colloids . .	Smaller than 0.001 mm

Soil Classification. Soils may be classified according to the grain size of the granular mineral material as indicated Table 3-1. There is not complete agreement as to the limits of the size of the various constituents.

The separation into grain size by sieving is relatively simple down to 0.074 mm, which corresponds to a 200-mesh standard sieve. Below that size the gradation must be made by determining the rate of settlement in water, which is in part a function of the grain size. Some difficulties exist in the test procedures, producing results that are not entirely accurate.

Water is an important constituent of soil and may occur as free water or be absorbed. Water enters the soil by gravity flow from rain water and melting snow or by capillarity from an underlying source. Some inorganic chemical compounds occur in soils and have some importance because certain ones tend to flocculate fine-grained soils, while other compounds tends to deflocculate these soils. The mineral composition of fine-grained soils may be as important as the grain size.



Size of Particle	
Grade	Millimeters
Coarse sand	2 to 0.25
Fine sand	0.25 to 0.05
Silt	0.05 to 0.005
Clay	Smaller than 0.005

Fig. 3-1. Textural classification of subgrade soils. (After chart prepared by Public Roads Administration.[8])

Natural deposits of soils may consist of any combination of the several constituents. Here again, there is not complete agreement as to name, and owing to the diversity of combinations soils may be called blue clay, yellow clay, stiff blue clay, sandy loam, sandy clay, etc. These terms may mean one thing to one investigator and another to a second. The soil classification by "texture" is given in Fig. 3-1. It will be noted that the term "clay" here applies to a soil that may consist of varying percentages of sand, silt, and clay, as previously defined by particle size. The percentage ranges are sand, 0 to 50; silt, 0 to 50; and clay, 30 to 100.

Properties of Soils. *Particle Size and Shape.* The shape and size of soil particles influence or produce certain properties. Sand grains are somewhat spherical in shape, while clay grains are elongated or flat. The size and shape influence the arrangement of the particles in the soil mass and the resultant properties. Sand is less compressible than clay, and gravitational water flows through sand more readily than through clay.

Permeability. Permeability is the rate at which gravitational water flows through a soil and depends upon the size and continuity of the pores, the hydraulic gradient, and other factors

Shearing Strength. The supporting power of a soil is its ability to resist penetration or lateral flow. The shearing strength is made up of two elements: the friction between the soil particles—internal friction—and the attraction of the soil particles, or cohesion. Cohesion in turn consists of two elements: true cohesion—the molecular attraction of particle to particle—and apparent cohesion—the molecular attraction of water. Internal friction is greatest in sand and least in clay, while cohesion is greatest in clay and least in sand.

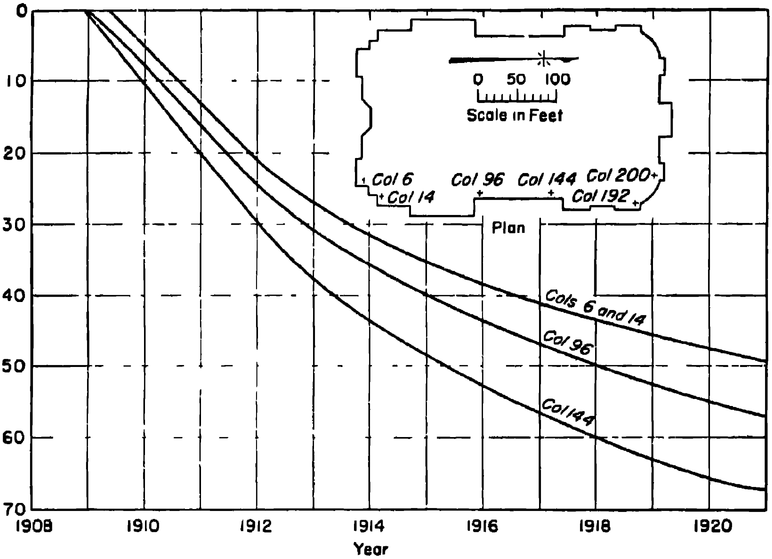


FIG. 3-2. Settlement curves of various columns of National Theater, City of Mexico, Mexico. (*Earths and Foundations, Proc ASCE, May,* 1933)

Capillarity Capillarity is that property of a soil which enables it to draw water from some source and transmit it in all directions against the force of gravity. Surface tension is the force that pulls the water against gravity and occurs when both air and water are present in the soil and the pores are of appropriate size The height to which water will rise depends upon the affinity of the particular soil for water and the diameter of the pores. For equal affinities the height is inversely proportional to the pore diameter. The rate of rise as well as the height is important. The rate of rise is inversely proportional to the diameter of the pores.

As the capillarity water evaporates, the surface tension causes a compressive force in the soil, and the soil will shrink When the surface tension is destroyed by "flooding," the soil expands. The large cracks that develop when "mud" flats dry out are examples of shrinkage.

Density. Density of a soil represents the extent to which the soil has been consolidated. It is a function of the relation of the volume of voids and the volume of soil particles in a given volume of soil. The specific gravities of soil particles vary over a relatively small range, and as there may be as much as 75 or 80 lb per cu ft difference in the weight of soils, these differences in weight are due to the variation in voids. The ratio of the volume of voids to the volume of soil is called the "percentage of voids," and the ratio of the volume of voids to the volume of soil particles is called the "voids ratio." Voids are sometimes called "pores."

The voids in sand are of relatively large size and therefore offer little resistance to the flow of water. The voids or pores in clay are relatively small and offer high resistance to the flow of water, although the total volume of voids may be more in clay than in sand.

Compressibility. Compressibility of a soil is a measure of the reduction in volume as load is applied. Generally the unit pressures are so small that the deformation of the individual soil particles is negligible, and therefore the settlement is due to the reduction of the volume of voids or lateral flow. As the volume of voids is reduced, water or air (or other gases) will be expelled from the voids.

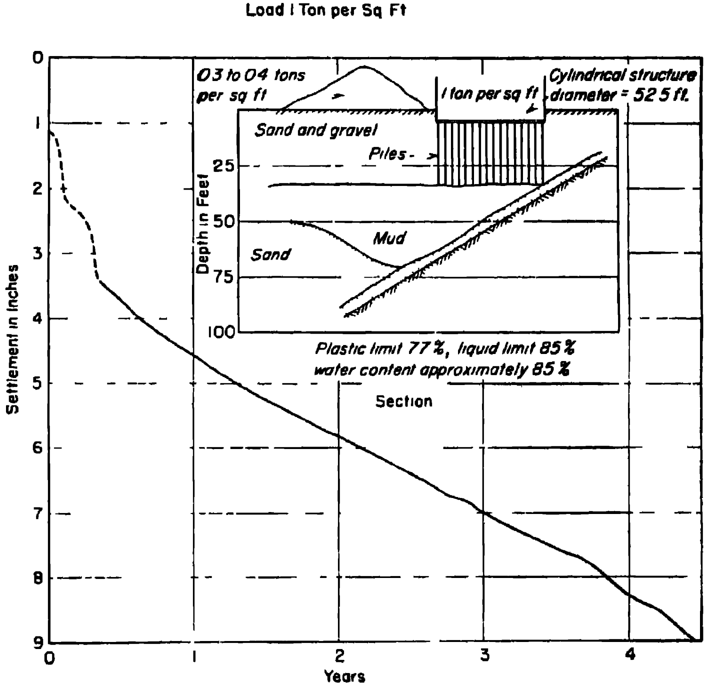


FIG. 3-3. Settlement of oil tank on wooden piles due to consolidation of mud layer. (*Earths and Foundations, Proc. ASCE, May,* 1933.)

The settlement of sands is relatively small unless there is a rearrangement of the soil particles due to a sudden shock. The settlement of clays, on the other hand, may be considerable and may take place over an extended period of time because of the high resistance to the flow of water or gas offered by the clay (see Figs. 3-2, 3-3, and 3-4).

Elasticity. Elasticity is that property of a soil which causes it to rebound after an applied load is removed. Sands are not elastic, while clays under low load are.

Plasticity. Plasticity is that property of a soil which enables it to undergo a marked change in shape without change in volume. Sands are not plastic.

Soil Tests. The foundations of buildings or similar structures, when founded on unconsolidated soils, are placed at some depth and on soils that are relatively undisturbed, whereas the subgrade for a highway, airport, or storage area is considerably disturbed by the very nature of the construction methods. The shear and consolidation tests on the soil of the first case are therefore generally made on "undisturbed" samples. The various tests on subgrade soils are generally made on disturbed samples.

The more common tests on subgrade soils are liquid limit, plastic limit, shrinkage limit, centrifuge-moisture equivalent, field-moisture equivalent, and density. Part or all of these are covered by ASTM standards.

The liquid limit of a soil is the minimum amount of moisture that will cause the soil to flow under a small force. The plastic limit is the smallest moisture content at which the soil can be rolled into a ⅛-in. diameter thread without breaking. The difference between the liquid limit and the plastic limit is the plasticity index. The plasticity index is a measure of the cohesion of the soil.

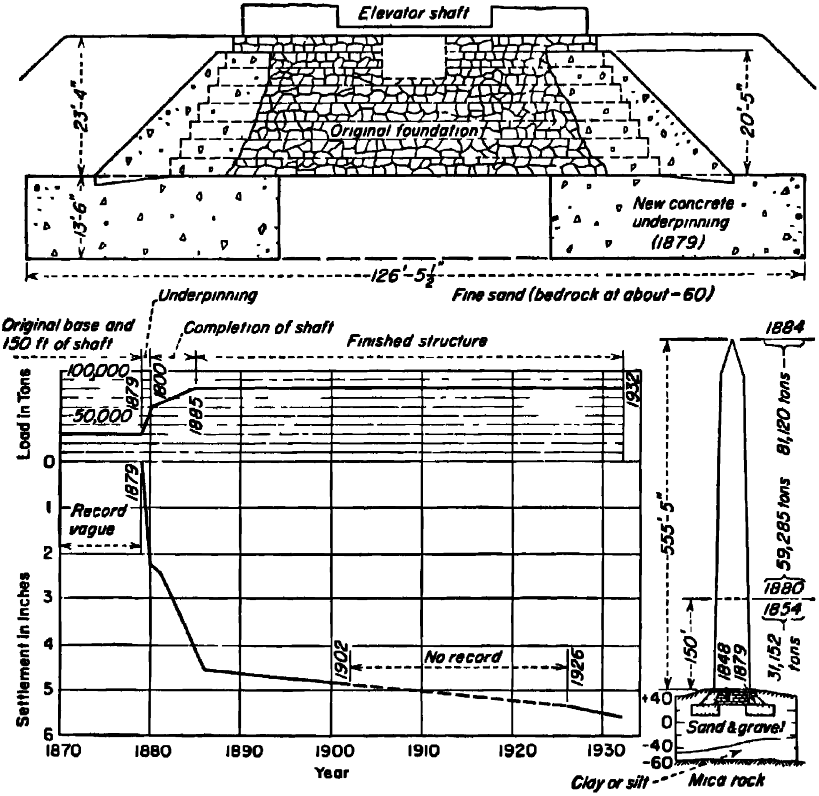


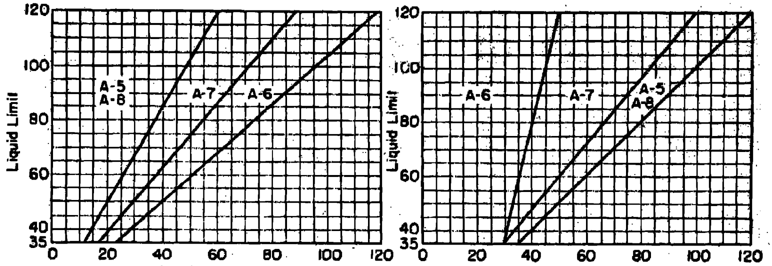
FIG. 3-4. Settlement record of Washington Monument. (*Earths and Foundations, Proc. ASCE, May*, 1933.)

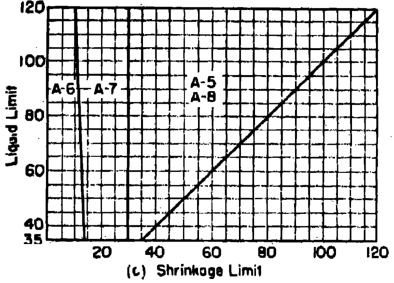
The shrinkage limit is the moisture content at which further evaporation does not cause additional shrinkage. This is the point at which the capillarity force equals the resistance of the soil particles to further consolidation.

The centrifuge-moisture equivalent is the moisture content of a soil that was previously saturated and subjected to a specified centrifugal force for a specified period of time. The centrifuge-moisture equivalent is indicative of the capillarity and permeability of the soil.

The field-moisture equivalent is the minimum moisture content at which a drop of water placed on a sample will not be immediately absorbed. The field-moisture equivalent is the starting point for determining volume change when the moisture content is reduced to the shrinkage limit. A volume change of 15 to 20 per cent between these moisture contents appears to be the dividing line between stable and unstable subgrade soils.

The Public Roads Administration has established eight classifications of soils. A summary of the characteristics and suitability for use of these soils under various conditions is given in Table 3-2 and Fig. 3-5.[1]





GROUP	A-1	A-2	A-3	A-4	A-5	A-6	A-7	A-8
Coarse material	0-65	+	+	+	+		+	
Soil mortar total sand	70-85	55 min.	+	55 max.				
Coarse sand	45-60	+	+	+	+		+	
Silt	10-20	+	+	+	+		+	
Clay	5-10	+	+	+	+		+	
Percentage passing No. 200 sieve	+	+	0-10	+	+		+	
Liquid limit	14-35	35 max.	N.P.	20-40	35 min			
Plasticity index	4-9	N.P.-15	N.P.	0-15	See chart (a)			
Field moisture equivalent	+	+	+	30 max.	See chart (b)			
Centrifuge moisture equivalent	15 max.	25 max.	12 max.	+	+			
Shrinkage limit	14-20	+	+	20-30	See chart (c)			

Note – Only the A-1 materials with values of liquid limit not greater than 25 and values of plasticity index not greater than 6 are suitable for use in base courses for thin flexible surfaces.

+ Not essential N.P. Nonplastic

FIG. 3-5. Range of soil characteristics. (*Courtesy of Public Roads Administration, Federal Works Agency.*)

FOUNDATION TYPES

The type of foundation selected for a particular structure depends upon the loads to be transferred to the foundation bed, upon the material on which the structure is founded, and upon the method of placing the foundation, which in turn depends upon the material. The purpose of the foundation is, of course, to distribute the load of the structure so that the settlement will not cause excessive maintenance or impair the usefulness of the structure.

The foundation of a light building may be simply a foundation wall founded upon rock or hardpan. This would be possible where a suitable foundation bed is near the surface or near the line of basement excavation.

When soil of lower bearing capacity is encountered, it may be necessary to increase the bearing area by the addition of a footing, as indicated in Fig. 3-6*b*, in order to reduce the pressure so that undue settlement will not take place. On simple structures the footings are usually made of plain concrete, and in such cases the projection b' (Fig. 3-6*b*) should not be greater than the depth of the footing and preferably less, say a ratio of b' to d of 2 to 3. When the projection b' is greater than the depth, transverse reinforcement is required, and the footing should be designed as a cantilever beam. The footing must be so proportioned that the shearing stress is within the limits set for beams without web reinforcement (Fig. 3-6*d*).

Table 3-2. Summary of Soil Characteristics and Classification

Group	A-1	A-2		A-3	A-4	A-5	A-6	A-7	A-8
		Friable	Plastic						
General stability properties	Highly stable at all times	Stable when dry, may ravel	Good stable material	Ideal support when confined	Satisfactory when dry, loses of stability when wet or by frost action	Difficult to compact, stability doubtful	Good stability when properly compacted	Good stability when properly compacted	Incapable of support
Physical constants:									
Internal friction	High	High	High	High	Variable	Variable	Low	Low	Low
Cohesion	High	Low	High	None	Variable	Low	High	High	Low
Shrinkage	Not detrimental	Not significant	Detrimental when poorly graded	Not significant	Variable	Variable	Detrimental	Detrimental	Detrimental
Expansion	None	None	Some	Slight	Variable	High	High	Detrimental	Detrimental
Capillarity	None	None	None	Slight	Detrimental	High	High	High	Detrimental
Elasticity	None	None	Some	None	Variable	Detrimental	None	High	Detrimental
Textural classification. General grading	Uniformly graded coarse-fine, excellent binder	Poor grading, poor binder	Poor grading, inferior binder	Coarse material only, no binder	Fine sand, cohesionless silt, and friable clay	Micaceous and diatomaceous	Deflocculated cohesive clays	Drainable flocculated clays	Peat and muck
Approximate limits:									
Sand, %	70–85	55–80	55–90	75–100	55 (maximum)	55 (maximum)	55 (maximum)	55 (maximum)	55 (maximum)
Silt, %	10–20	0–45	0–45	*	High	Medium	Medium	Medium	Not significant
Clay, %	5–10	0–45	0–45	*	Low	Low	30 (minimum)	30 (minimum)	Not significant
Physical characteristics									
Liquid limit	14–25†	35 (maximum)	35 (maximum)	NP‡	20–40	35 (minimum)	35 (minimum)	35 (minimum)	35–400
Plasticity index	4–9†	NP–3‡	3–15	NP‡	0–15	0–60	18 (minimum)	12 (minimum)	0–60
Field-moisture equivalent	Not essential	Not essential	Not essential	Not essential	30 (maximum)	30–120	50 (maximum)	30–100	30–400
Centrifuge-moisture equivalent	15 (maximum)	12–25	25 (maximum)	12 (maximum)	Not essential	Not essential	Not essential	Not essential	Not essential
Shrinkage limit	14–20	15–25	25 (maximum)	Not essential	20–30	30–120	6–14	10–30	30–120
Shrinkage ratio	1.7–1.9	1.7–1.9	1.7–1.9	Not essential	1.5–1.7	0.7–1.5	1.7–2.0	1.7–2.0	0.3–1.4
Volume change	0–10	0–6	0–16	None	0–16	0–16	17 (minimum)	17 (minimum)	4–200
Lineal shrinkage	0–3	0–2	0–4	None	0–4	0–4	5 (minimum)	5 (minimum)	1–30
Compaction characteristics									
Maximum dry wt, lb cu ft	130 (minimum)	120–130	120–130	120–130	110–120	80–100	80–110	80–110	90 (maximum)
Optimum moisture, percentage of dry wt, approx	9	9–12	9–12	9–12	12–17	22–30	17–28	17–28	
Maximum field compaction required, percentage of maximum dry wt, lb cu ft	90	90	90	90	95	100	100	100	Waste
Rating for fills 50 ft or less in height	Excellent	Good	Good	Good	Good to poor	Poor to very poor	Fair to poor	Fair to poor	Unsatisfactory
Rating for fills more than 50 ft in height	Good	Good to fair	Good to fair	Good to fair	Fair to poor	Very poor	Very poor	Very poor	Unsatisfactory
Required total thickness for subbase, base, and surfacing in	0–6	0–6	2–9	0–6	9–18	9–24	12–24	12–24	

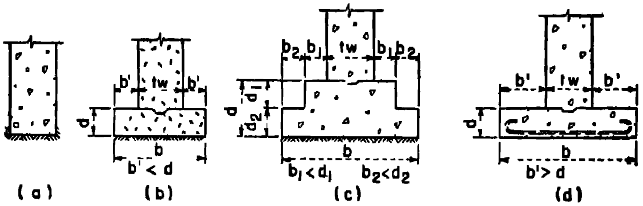
* Percentage passing No. 200 sieve, 0–10

† When used as a base course for thin flexible surfaces the plasticity index and liquid limit should not exceed 6 and 25, respectively

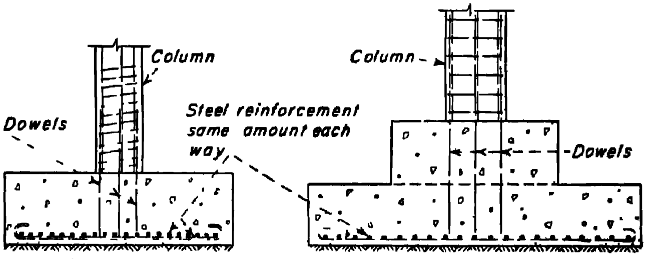
‡ NP = nonplastic

Courtesy—Public Roads Administration, Federal Works Agency

Where a considerable projection is required and it appears undesirable to use reinforcement, some saving in the quantity of concrete may be made by stepping the footing. In such a case, each projection must be proportioned as above. While there is some saving in the quantity of concrete in the stepped footing, the increased cost of forms may amount to more than the saving in the cost of the concrete (Fig. 3-6c).



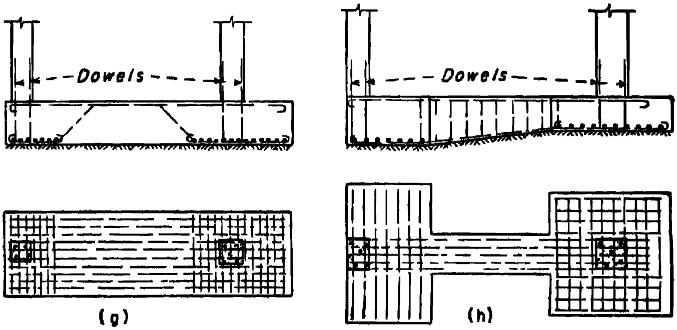
Cross-sections of foundation wall footings



(e) plain

(f) stepped

Simple spread footings (usually square in plan)



Combined footing

Cantilever footing

Fig. 3-6. Typical footings.

Spread footings may be used to distribute moderate column load to the foundation bed. They may appear as separate footings under individual columns, combined to distribute the load of two columns, or as a raft where several columns are supported on a continuous slab (see Figs. 3-6e, f, g, and h).

While the footings shown in Figs 3-6e and g are sketched as made of reinforced concrete, they be made of timber or steel and are then generally referred to as "grillages." Grillages may also be used to distribute the load from steel columns to concrete piers.

Piles are used to distribute the load to greater depths of soils of relatively low bearing values or to transfer the load through such soil to a more desirable material at lower depths. Piles may also be used to consolidate the material into which they are driven. The load is transferred from the pile to the foundation material by shearing forces along the pile, friction, or by bearing at the end of the pile, or by both friction and bearing. When the load is transferred primarily by friction, the pile is called a "friction pile," and when the load is transferred primarily in bearing, the pile is called a "bearing pile."

Timber piles may be used where the driving is relatively easy, and untreated piles should be cut off below the permanent ground-water level. Timber piles are susceptible to marine borers. Several species of wood provide satisfactory piles, especially Douglas fir, cedar, oak, and southern pine. Timber piles should not be overdriven.

Concrete piles may be cast in place or precast. Precast piles should be designed to withstand stresses caused by handling and driving as well as other applied loads. Cast-in-place concrete piles are generally made by driving a metal casing, with or without a mandrel, and filling the shell with concrete. In some cases the shell may be withdrawn as the concrete is poured. Also, hollow steel cylinders may be driven to the desired depth and the material within the shell removed by the use of compressed air and the cylinder filled with concrete. This type is called a "pipe pile." The diameter may vary from 6 or 8 in. to 36 in. or more. The larger sizes are sometimes referred to as "piers." Various methods have been developed for driving the cylinders.

Where long piles are required and the ground water is at a considerable depth, a combination timber and concrete pile may be used. The timber portion is driven to ground level by ordinary means, and then a follower is used to force the timber to the desired depth. The follower is withdrawn and the hole filled with concrete. A casing around the follower may or may not be required. A positive connection between the timber and the concrete is necessary.

Steel H-piles may be used where heavy loads are involved and where difficult driving may be encountered, such as through gravel or shale. H-piles do not displace as much material as timber or concrete piles.

METHOD OF PLACING FOUNDATIONS

While the method of placing foundations is listed here under a separate heading, it cannot be entirely separated from the type of foundation or from the particular conditions at a given site. The type of soil and the presence or absence of water are controlling factors both as to type and methods of placing and also influence the depth to which the foundations must be carried. The requirements for the support of adjacent structures must be considered.

Column footings for a light mill building without a basement may be placed in relatively shallow pits and generally present no serious problem. Buildings with one or more basements may create difficult problems. In relatively dry ground the pit may be excavated with sloping sides or by the use of sheathing If water is present, it may be controlled by pumping from sumps or by the installation of a system of well points whereby the water table is lowered over the entire area of excavation. If buildings or streets adjoin the new structure, bracing will frequently be required to prevent the caving of the ground with resultant damage to existing structures.

The columns of heavy buildings that are to be erected on a site where there is a considerable depth of stiff clay may be carried on concrete piers or caissons extending to rock or other suitable foundation material. In such cases the piers are usually constructed by the open-well or poling-board method (sometimes called the Chicago method). The diameter of the pier depends upon the column load. A section 4, 5, or 6 ft deep is excavated, and vertical sheathing fitted in the form of a cylinder is placed and held by channels bent to appropriate radius. The next lower section is then excavated and the lagging placed. The procedure is repeated until the excavation is of proper depth, *i.e.*, until suitable foundation material is reached. The open cylinder is then filled with concrete. Moderate amounts of water can be handled by pumping. Care should be exercised not to disturb existing structures by excessive pumping. In some cases it may be desirable to cap the cylinders and complete the excavation under air pressure.

Cofferdam. A cofferdam is an enclosure around an area in which the foundation is to be placed. The enclosure may be made of any material, but it is frequently made of steel-sheet piling or timber. It may be constructed with a single wall or with a double wall. Usually the cofferdam does not become a part of the foundation.

The cofferdam may be pumped out and the excavation carried on in the dry or the excavation may be made by dredging through the water. Likewise, the concrete may be placed in the dry, or a seal may be placed by pouring the concrete through the water with a tremie and then dewatering the cofferdam. The remaining concrete is then placed in the dry. The latter method is usually employed on bridge piers and similar structures.

Open Caisson. The open caisson is more generally used for the construction of bridge piers than for the foundations for buildings It might be a suitable method of construction for the foundation for a heavy stack under appropriate conditions.

An open caisson consists of a boxlike structure made of several vertical cells and generally provided with a false bottom in each cell. The caisson is placed in position and enough of the false bottoms removed to permit the caisson to sink. Excavation is carried out through the cells by means of an orange-peel or clamshell bucket, and by dredging in the various cells the caisson can be maintained in an upright position. As the excavation progresses, the caisson sinks, and additional height is added by pouring concrete in the walls or some of the closed cells. The alternate operations of excavation and concrete pouring continue until the bottom of the caisson reaches the desired depth. A diver then usually examines the foundation material and cleans and prepares it for final concreting. The bottom of the caisson is sealed by tremie concrete, and after that concrete has attained sufficient strength the various cells are unwatered and the remaining concrete placed in the dry.

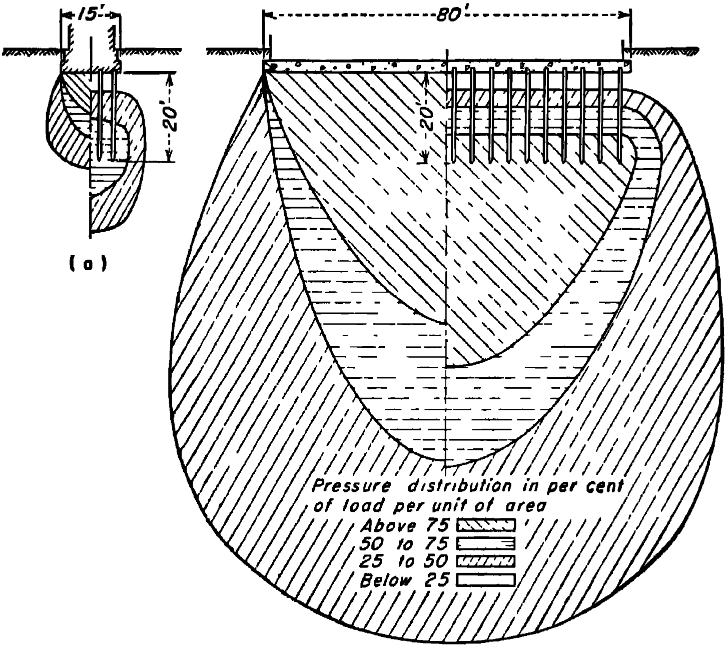
Pneumatic Caisson. A pneumatic caisson is essentially a boxlike structure, open at the bottom and closed at the top, in which compressed air is used to balance the water pressure on the outside. The excavation and placing of the concrete is done in the dry. Workmen are admitted to and excavated material is removed from the working chamber through air locks. As excavation progresses and the caisson sinks, additional height is added to the top. When the final grade is reached, the foundation bed is cleaned, and a concrete seal is poured while the necessary air pressure is maintained. After the concrete seal has attained sufficient strength, the air is released and the remaining concrete poured under atmospheric pressure.

FOUNDATION PRESSURES

The load from a foundation will be distributed in some manner to the foundation material, and pressures of varying intensities will develop in the material somewhat as indicated in the left half of Fig. 3-7a. The pressure lines form what is called the "bulb of pressure." The extent of the pressure bulbs depends upon the width of the loaded area. The pressure lines represent some fractional part of the unit pressure $p = P/A$, where P is the applied load and A is the area of the foundation. Lines of equal pressure are called "isobars." Studies by Terzaghi[2] indicate that for an ideal material (a perfectly elastic, isotropic, semi-infinite solid) about 80 per cent of the settlement of a foundation can be accounted for by the distortion of the material within the $0.2p$ pressure line. The depth of the $0.2p$ pressure line below the bottom of the foundation is about 1.5 times the width of the foundation. The load from piles develops similar pressure bulbs.

The pressure bulbs of closely spaced footings overlap and form a large bulb for the foundation as a whole. Likewise, the pressure bulbs of the individual piles in a group overlap and form a bulb for the group. The load capacity of a group of footings or of a group of piles is not directly proportional to the number of footings or the number of piles. The total is less. The settlement of a foundation covering a large area may be as much with piles as without piles. In other words, in some cases there may be no advantage in using piles. This situation is illustrated in Fig. 3-7.[3]

Settlement. The principal part of the settlement of a structure is due to two elements—the lateral and possible upward movement of the soil and the compression of the soil under the foundation. For footings near the surface the lateral movement of the soil may produce heaving around the foundation. The depth of the footing below the surface is a factor in lateral movement. Foundations placed on firm sand or gravel underlain with a strata of peat or other soft material may settle a considerable amount owing to the lateral flow of the soft material, and the heaving due to lateral movement may occur at some distance from the foundation.



(a)

(b)

FIG. 3-7. Pressure distribution under a foundation. (*Terzaghi, Charles, The Science of Foundations—Its Present and Future, Trans. ASCE, Vol. 93, 1929.*)

Distribution of Pressure. The distribution of pressure under an isolated footing placed at some depth below the surface on clay varies from a maximum at the edge of the footing to a minimum at the center. For a similar footing on sand the pressure varies from a minimum at the edge to a maximum at the center. The greater the depth, the more nearly are the edge and center pressures alike. Uniform pressures are usually assumed for design purposes. For isolated footing the assumption appears to be on the safe side, but for combined or mat foundations the assumption may be on the unsafe side.

Bearing Pressures. Elaborate investigations of the soil properties at a given site can be justified only in the case of a large, expensive structure. The cost of complete investigation for a minor job is prohibitive. Accordingly, the design of the foundation is usually based upon some allowable soil pressure which is the result of experience on similar sites. Building codes give a set of bearing pressures permitted on various soils. There is not complete agreement among the listed values.

Examination of the surface is generally not sufficient to determine the character of the soil at various depths, and unless data from adjoining structures are available, borings should be made to evaluate the kind of material that exists to a depth of about $1\frac{1}{2}$ times the width of the foundation The soil near the surface may be compact sand or gravel, which would indicate a satisfactory material If, however, a strata of soft material exists below the firm material, considerable settlement may be expected unless the appropriate bearing pressures are used Some of the more recently revised building codes recognize this situation

Table 3-3. Suggested Permissible Bearing Values

Type of soil	*Kips/sq ft*
Soft clay	2
Fine wet confined sand	4
Sand and clay mixed	4
Dry coarse sand	6–8
Hard clay	6–8
Gravel	10–12
Hardpan	16–20
Rock	30–40

Pressure on Planes at Various Depths The computation of the intensity of pressure on planes at varying depths below the foundation is most conveniently done by the use of influence charts which have been developed by N M Newmark[4]

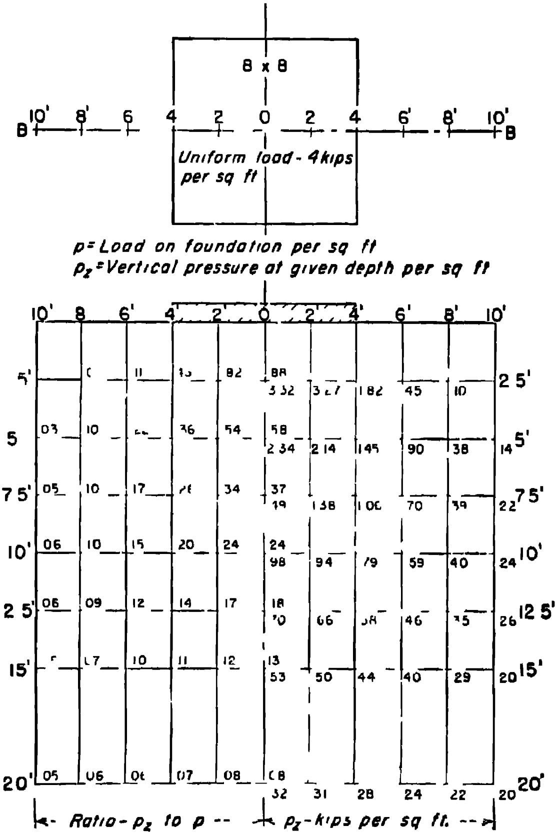
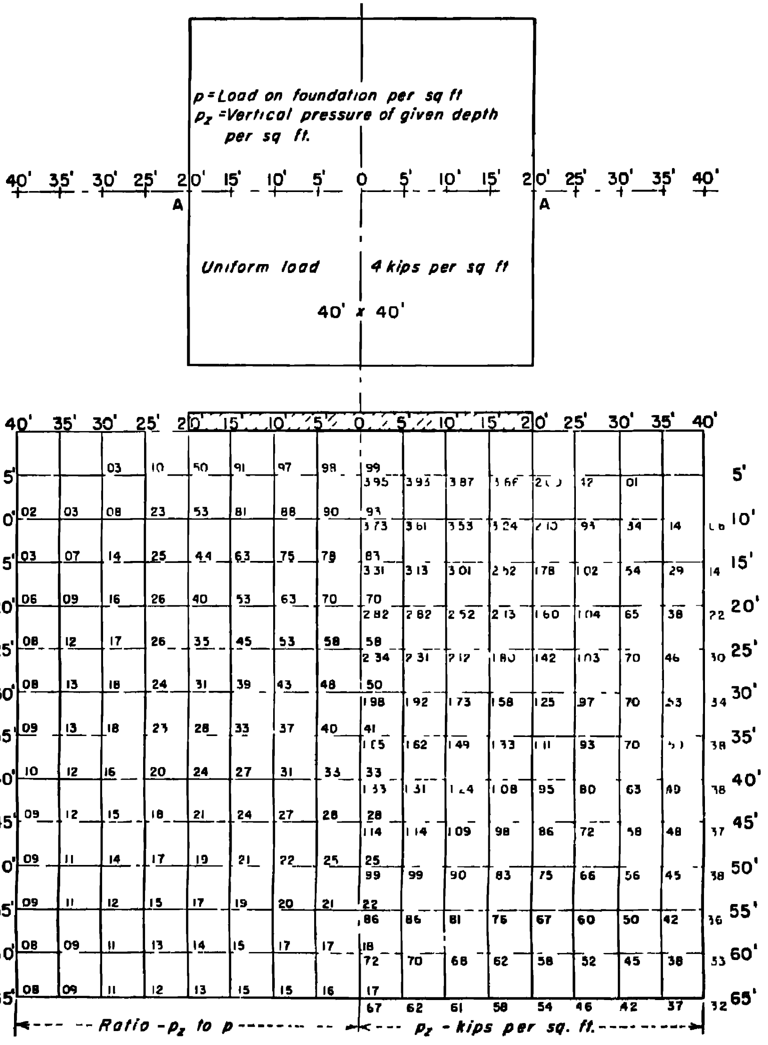


Fig 3-8 Vertical pressures on horizontal planes along center line of 8- by 8-ft footing. Uniform load, 4 kips per sq ft

Figure 3-8 gives the intensity of vertical pressure on various horizontal planes under a single spread footing 8 ft square The values on the left half of the chart are the ratios p_z/p of the intensity of vertical pressure at a given point to the intensity of loading. The values on the right half of the chart are the vertical pressures in kips per square foot at various points due to a load of 4 kips per sq ft. Figure 3-9 gives similar values for a foundation 40 ft square loaded uniformly with 4 kips per sq ft. Note the ratio of p_z to p in both Figs. 3-8 and 3-9 at a depth of about 1½ times the width. Note also that for the 8- by 8-ft footing the values are given for a depth of 12.5 ft and for the 40- by 40-ft foundation the depth is 60 ft. Comparison at a depth about equal to the width of the loaded area may also be made.



	40'	35'	30'	25'	20'	15'	10'	5'	0	0	5'	10'	15'	20'	25'	30'	35'	40'
5'			03	10	50	91	97	98	99	3 95	3 93	3 87	3 66	[illegible]	47	01		
10'	02	03	08	23	53	81	88	90	93	3 73	3 61	3 53	3 24	2 10	93	34	14	06
15'	03	07	14	25	44	63	75	78	83	3 31	3 13	3 01	2 52	1 78	1 02	54	29	14
20'	06	09	16	26	40	53	63	70	70	2 82	2 82	2 52	2 13	1 60	1 04	65	38	22
25'	08	12	17	26	35	45	53	58	58	2 34	2 31	2 12	1 80	1 42	1 03	70	46	30
30'	08	13	18	24	31	39	43	48	50	1 98	1 92	1 73	1 58	1 25	97	70	53	34
35'	09	13	18	23	28	33	37	40	41	1 65	1 62	1 49	1 33	1 11	93	70	51	38
40'	10	12	16	20	24	27	31	33	33	1 33	1 31	1 24	1 08	95	80	63	40	38
45'	09	12	15	18	21	24	27	28	28	1 14	1 14	1 09	98	86	72	58	48	37
50'	09	11	14	17	19	21	22	25	25	99	99	90	83	75	66	56	45	38
55'	09	11	12	15	17	19	20	21	22	86	86	81	76	67	60	50	42	36
60'	08	09	11	13	14	15	17	17	18	72	70	68	62	58	52	45	38	33
65'	08	09	11	12	13	15	15	16	17	67	62	61	58	54	46	42	37	32
	Ratio - p_z to p									p_z - kips per sq. ft.								

FIG. 3-9. Vertical pressures on horizontal planes along center line of 40- by 40-ft footing. Uniform load, 4 kips per sq ft.

SPREAD FOOTINGS

The design of wall and column spread footing is usually based upon the recommendations given by A. N. Talbot.[5] These recommendations have been frequently incorporated into various codes. The current Building Regulations for Reinforced Concrete (ACI 318-41)[6] makes allowance for bending moments approximately 15 to 20 per cent greater than those computed under the recommendations of Talbot.

The following design computations for plain wall footing, reinforced wall footing, and spread footing for a single column are based on the recommendations of the 1941 Building Regulations for Reinforced Concrete of the American Concrete Institute. The combined footing is designed under the recommendations of the 1940 Report of the Joint Committee on Standard Specifications for Concrete and Reinforced Concrete.

Wall Footings. Specifications: 1941 ACI Building Regulations.

$f_c' = 2{,}000$ psi $\qquad j = 1 - 0.143 = 0.857$

$f_s = 18{,}000$ psi $\qquad p = \dfrac{0.429 \times 900}{2 \times 18{,}000} = 0.0107$

$n = 15$

$k = 900 \div 2{,}100 = 0.429 \qquad R = \dfrac{900 \times 0.429 \times 0.857}{2} = 165$

18-in. concrete wall and applied load = 25,000 lb per ft of wall
Allowable soil pressure = 5,000 psf

Plain Concrete Footing

Tension in concrete $\gtrless 0.03 f_c' = 60$ psi

25,000 ÷ 5,000 = 5 ft Use 5 ft 6 in.
−1 ft 6 in.
4 ft 0 in. ÷ 2 = 2 ft 0 in. projection each side

$M = (25{,}000 \div 5{,}000) \times 2 \times 1 = 9{,}100$ ft-lb

$d^2 = \dfrac{9{,}100 \times 12 \times 6}{12 \times 60} = 910$

$d = 30.2$ in. Use 30 in. Wt = 375 psf

25,000 ÷ 5,000 = 4,550
+ 375
4,925 psf OK

Diagonal tension OK $d >$ projection

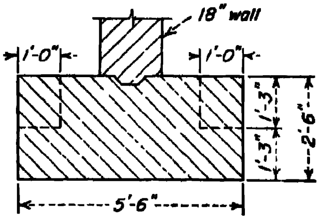


FIG. 3-10.

NOTE: Some saving in concrete may be made by stepping footing as indicated by dotted lines. Added cost of formwork may be greater than saving in cost of concrete.

Reinforced Footing. Same wall, load, and bearing pressure as above.

25,000 ÷ 5,000 = 5 ft 0 in. Use 5 ft 3 in.
− 1 ft 6 in.
3 ft 9 in. ÷ 2 = 1 ft $10\frac{1}{2}$ in.

25,000 ÷ 5.25 = 4,760 psf

$M = 4{,}760 \times 1.87 \times 0.93 = 8{,}300$ ft-lb

$V = 8{,}900$ lb

$d = \sqrt{\dfrac{8{,}300 \times 12}{165 \times 12}} = 7.1$ in.

Try 7 in.
+ 3 in. cover
10 in. over-all

$A_s = 7 \times 12 \times 0.0107 = 0.91$ sq in. per ft

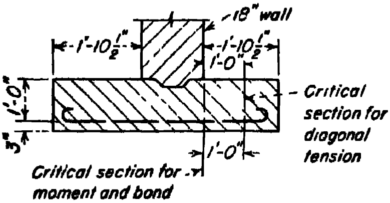

18" wall
1'-10½"
1'-10½"
1'-0"
1'-0"
3"
1'-0"
Critical section for diagonal tension
Critical section for moment and bond

FIG. 3-11.

$\Sigma 0 = \dfrac{8{,}900}{\frac{7}{8} \times 7 \times 112} = 13$ in. per ft Too much

Increase d to 12 in. 15 in. over-all

$\Sigma 0 = 13.0 \times 7/12 = 7.7$ in. per ft

$$A_s = \frac{8,300 \times 12}{0.86 \times 12 \times 18,000} = 0.54 \text{ sq in. per ft}$$

Use ½-in. diam, 3-in. centers $A_s = 1$ sq in. per ft $\Sigma 0 = 8.0$ in. per ft

$V = 4,760(1.87 - 1.00) = 4,150$

$$v_c = \frac{4,150}{7/8 \times 12 \times 12} = 33 \text{ psi}$$ Diagonal tension OK

v, allowable $= 60$ psi

$$\begin{array}{r} 4,760 \\ +\ \underline{190} \\ 4,950 \end{array}$$ psf OK

Spread Footing. Specifications: 1941 ACI Building Regulations

Column load = 325 kips

Column = 18 in. × 18 in.

Allowable pressure = 6 ksf

$f_c' = 2,000$ psi

$f_s = 18,000$ psi

$R = 165$

$u = 112$ psi, deformed and hooked

$A = 325 \div 6 = 54.2$ sq ft $7.5 \times 7.5 = 56.3$ sq ft $325 \div 56.3 = 5.78$ ksf

$M = 7.5 \times 3.0 \times 1.5 \times 5.78 \times 0.85 = 166$ ft-k

$V_{\text{bond}} = 7.5 \times 3.0 \times 5.78 \times 0.85 = 110$ kips

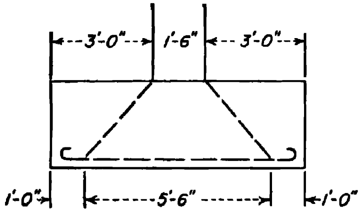


Fig. 3-12. Spread footing section for diagonal tension.

$$d = \sqrt{\frac{166,000 \times 12}{90 \times 165}} = 11.6 \text{ in.}$$

Say 12 in.

$$A_s = \frac{166 \times 12}{0.86 \times 12 \times 18} = 10.7 \text{ sq in.}$$

$$\Sigma 0 = \frac{110,000}{0.86 \times 12 \times 112} = 94 \text{ in.}$$

Too much

Make $d = 24$ in. Over-all depth = 28 in.

$$A_s = \frac{10.7 \times 12}{24} = 5.3 \text{ sq in.}$$

$$\Sigma 0 = \frac{94 \times 12}{24} = 47 \text{ in.}$$

Use 20 ¾-in.-diam bars $A_s = 8.8$ sq in. $\Sigma 0 = 47.2$ in. OK

$$V = \frac{7.5 + 5.5}{2} \times 1.0 \times 5,780 = 37,600 \text{ lb}$$

$$v = \frac{37,600}{66 \times 24 \times 0.86} = 27 \text{ psi}$$

OK. Allowable = 60 psi

$5,780 + 350 = 6,130$ psf Little high, OK

Length of dowels—column steel ⅞ in. diam:

$$l = \frac{16,000 \times 7/8}{4 \times 112} = 31 \text{ in.}$$

Use 26 in. × 26 in. × 6 in. pedestal

$$\text{In column } l = \frac{16,000 \times 7/8}{4 \times 150} = 24 \text{ in.}$$

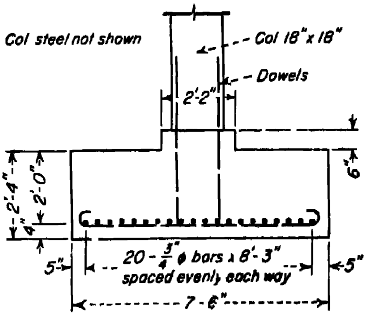


Fig. 3-13. Spread footing.

Transfer of compressive stress in concrete at base of column:

Column steel = 12 – ⅞ in. diam $\quad A_s = 7.2$ sq in.

$$f_c = \frac{325{,}000 - 7.2 \times 16{,}000}{324} = 645 \text{ psi}$$

Allowable on pedestal:

(*a*) Full area loaded = 0.25 × 2,000 = 500 psi

(*b*) ⅓ or less loaded = 0.375 × 2,000 = 750 psi

$$\text{Permissible ratio of loaded area to total area} = 1 - \left[\frac{(645 - 500)(1 - 0.33)}{250}\right] = 0.61$$

Minimum area of pedestal = 324 ÷ 0.61 = 530 sq in.

Area provided = 26 in. × 26 in. = 672 sq in. $\quad$ OK

Combined Footing. Specifications: 1940 Joint Committee Report.

$f_c' = 2{,}000$ psi $\quad R = 165$

$f_s = 18{,}000$ psi $\quad p = 0.0107$

$n = 15$ $\quad u = 112$ psi

$k = 0.43$ $\quad v = 60$ psi

$j = 0.86$ $\quad$ Use end anchorage.

Exterior column No. 11 $\quad$ 16 in. × 16 in. $\quad$ load, 176 kips

Interior column No. 21 $\quad$ 18 in. × 18 in. $\quad$ load, 272 kips

Maximum distance from property line to center line of exterior column No. 11 is 1 ft 1 in

Columns are 14 ft 0 in. on centers

Allowable soil pressure = 5 ksf

$$\text{Column No. 11 to center of gravity of footing} = \frac{272 \times 14}{448} = 8.5 \text{ ft}$$

Length of footing = (8.5 + 1.1) × 2 = 19.2 ft $\quad$ Use 19 ft 0 in.

Estimated weight of footing = 50 kips

Area required = (176 + 272 + 50) ÷ 5 = 99.6 sq ft

19 ft × 5 ft 3 in = 99.7 sq ft

Longitudinal Beam

$$d_M = \sqrt{\frac{484{,}600 \times 12}{165 \times 63}} = 24 \text{ in.}$$

$$v_c\text{, 2 ft to left of column 21} = \frac{112{,}700}{0.86 \times 24 \times 63}$$

= 87 psi $\quad$ Too high without web reinforcement

Make d = 30 in. $\quad$ Over-all depth = 31 in.

$$v_c\text{, 2 ft 6 in. to left of column 21} = \frac{100{,}900}{0.86 \times 30 \times 63}$$

= 62 psi $\quad$ OK. 60 psi allowable

$$A_s = \frac{484.6 \times 12}{0.86 \times 30 \times 18} = 12.5 \text{ sq in.}$$

$$\Sigma 0 = \frac{159{,}900}{0.86 \times 30 \times 75 \times 1.5} = 55 \text{ in.}$$

Use 17 1-in.-diam bars, negative steel in top between columns

A_s = 13.4 sq in. $\quad \Sigma 0$ = 53.4 in. $\quad$ Anchor bars

Interior end, right of column 21:

$$A_s = \frac{130 \times 12}{0.86 \times 30 \times 18} = 3.35 \text{ sq in.}$$

$$\Sigma 0 = \frac{76{,}700}{0.86 \times 30 \times 75 \times 1.5} = 26.5 \text{ in.}$$

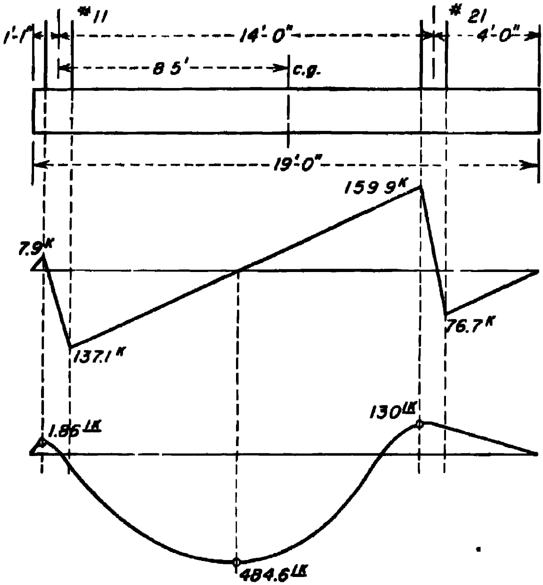


FIG. 3-14. Combined footing shears and moments.

Use 14 $\frac{5}{8}$-in.-diam bars in bottom
$A_s = 4.34$ sq in. $\Sigma 0 = 27.4$ in. OK

Exterior end, left of column 11:

$$f_c = \frac{1860 \times 12 \times 6}{63 \times 30 \times 30} = 2.4 \text{ psi} \qquad \text{No steel needed}$$

Transverse Beams

At exterior column 11: Assume effective width of 24 in. for compression

$$M = \frac{176 \times 1.96 \times 1.96}{5.25 \times 2} = 64.3 \text{ ft-k}$$

$$V = \frac{176 \times 1.96}{5.25} = 65.6 \text{ kips}$$

$$d = \sqrt{\frac{64{,}300 \times 12}{165 \times 24}} = 14 \text{ in.} \qquad \text{OK.} \qquad 30 \text{ in. available}$$

$$A_s = \frac{64.3 \times 12}{0.86 \times 30 \times 18} = 1.67 \text{ sq in.}$$

$$\Sigma 0 = \frac{65{,}600}{0.86 \times 30 \times 75 \times 1.5} = 22.6 \text{ in.}$$

Use 12 $\frac{5}{8}$-in.-diam bars $A_s = 3.41$ sq in. $\Sigma 0 = 23.5$ in.
Diagonal tension OK

At interior column 21: Assume effective width of 36 in. for compression

$$M = \frac{272 \times 1.88 \times 1.88}{5.25 \times 2} = 91.5 \text{ ft-k}$$

$$V = \frac{272 \times 1.88}{5.25} = 97.3 \text{ kips}$$

$$d = \sqrt{\frac{91{,}500 \times 12}{165 \times 36}} = 14 \text{ in.} \qquad \text{OK.} \quad 30 \text{ in. available}$$

$$A_s = \frac{91.5 \times 12}{0.86 \times 30 \times 18} = 2.37 \text{ sq in.}$$

$$\Sigma 0 = \frac{97{,}300}{0.86 \times 30 \times 75 \times 1.5} = 33 \text{ in.}$$

Use 17 $\frac{5}{8}$-in.-diam bars $A_s = 5.27$ sq in. $\Sigma 0 = 33.3$ in.
Weight of footing = 19 × 5.25 × 425 = 42,000 lb OK. 50,000 assumed

Transfer of stress at base of columns

Column 11:

$$\text{Area of pedestal} = \left(\frac{675}{500}\right)^2 \times 256 = 626 \text{ sq in.} \qquad \text{Use } 26 \text{ in.} \times 26 \text{ in.} = 676 \text{ sq in.}$$

$$\text{Length of dowels into column} = \frac{16{,}000}{4 \times 150} \times \frac{7}{8} = 23 \text{ in.}$$

$$\text{Length of dowels into footing} = \frac{16{,}000}{4 \times 100} \times \frac{7}{8} = 35 \text{ in.}$$

Use pedestal 26 in. × 26 in. × 5 in.

Column 21:

$$\text{Area of pedestal} = \left(\frac{675}{500}\right)^2 \times 324 = 794 \text{ sq in} \qquad \text{Use } 28 \text{ in.} \times 28 \text{ in.} = 786 \text{ sq in.}$$

$$\text{Length of dowels into column} = \frac{16{,}000}{4 \times 150} \times 1 = 27 \text{ in.}$$

$$\text{Length of dowels into footing} = \frac{16{,}000}{4 \times 100} \times 1 = 40 \text{ in.}$$

Use pedestal 28 in. × 28 in. × 10 in.

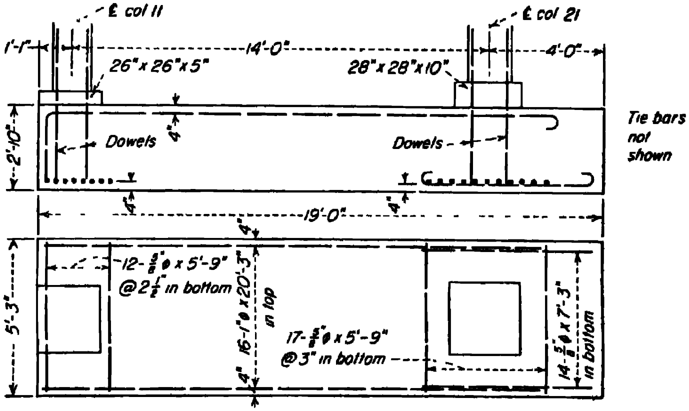


Fig. 3-15. Combined footing.

PILE-DRIVING FORMULAS

Formulas have been developed for computing the safe load on a pile based upon the observed penetration under stipulated conditions, the properties of the driving hammer, and the properties of the soil into which the pile is driven. There are many uncertain elements, and different formulas give different weights to these elements. Comparison of the ultimate load capacity of piles as computed by various formulas and the actual static loads at failure show erratic results.[7] At best, the formulas can be used only as a guide to the judgment of an experienced foundation engineer. It should be emphasized again that the load capacity of a group of piles is not the test load on a single pile multiplied by the number of piles in the group.

BASES AND SURFACES FOR STORAGE AREAS AND FLIGHT RUNWAYS

Table 3-4. Estimated Minimum Thickness of Concrete Runway Pavements[8]

Static wheel load, lb	Minimum slab thickness	
	Excellent subgrade support, in.	Inferior subgrade support, in.*
10,000	6	7
25,000	8	9
40,000	10	11
80,000	12	14

* Excellent support is considered to be furnished by soils of the A-1, A-2, or A-3 groups or by a foundation course of these materials from 12 to 18 in. thick placed on soils of the A-4, A-5, A-6, or A-7 groups. Inferior support [illegible]dered to be furnished by soils of the A-4, A-5, A-6, and A-7 groups without base courses under the pavement.

FLEXIBLE BASES AND SURFACES

Table 3-5. Total Combined Thickness of Subbase, Base Course, and Surface Course, In.[8]

Wheelloads, lb	10,000	25,000	40,000	80,000
Class of soil	Thickness, in.			
A-1	0-6	3-6	3-9	4 12
A-2 (friable)	0 6	3-6	3 9	4 12
A-2 (plastic)	2 8	4 10	6 12	8 15
A-3	4-6	5 8	6 9	8 12
A-4	9 18	15 25	18 30	24 36
A-5	9 24	15 30	18 36	24-48
A-6	12-24	18 30	24-36	30 54
A-7	12-24	18-30	24-36	30-54

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REINFORCED CONCRETE

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Late Professor of Structural Engineering, Cornell University

MANUFACTURE AND PROPERTIES OF CONCRETE

Definition. Concrete is a mixture of cement, fine aggregate, coarse aggregate, and water. The water reacts chemically with the cement, causing the water-cement paste to harden and thus bind the aggregate particles together. Concrete mixtures are defined by the relation between the amount of cement, the amount of fine aggregate, and the amount of coarse aggregate, *i.e.*, a 1:2:4 concrete contains 1 part cement, 2 parts fine aggregate, and 4 parts coarse aggregate. These proportions may be specified by weight or by volume, the latter being the more common.

Portland Cement. Portland cement is the principal binding agent or active ingredient in concrete. It is made by pulverizing the clinker obtained in burning an intimate and properly proportioned mixture of argillaceous and calcareous materials with no additions subsequent to calcination except water and calcined or uncalcined gypsum. Portland cement is classed as a hydraulic cement since it will harden under water. In order to be considered satisfactory for concrete work, portland cement should pass such standard specifications as those of the ASTM (see *ASTM Standards*, 1933).

Natural Cement. Natural cement is a hydraulic cement obtained by grinding to a fine powder the clinker resulting from the calcination of an argillaceous limestone (cement rock) at a temperature below fusion. Natural cement costs about the same as portland; it is not so strong as portland, it sets slower, and it is not so uniform in quality. Some natural cements are more resistant to variations of temperature and moisture conditions than portland. The practice of combining a high-grade natural cement with portland, in proportions from 1:6 to 1:3, for use in making concrete for structures with severe exposures is slowly coming into favor.

Special Cements. White portland cement is made of selected materials, low in iron oxide content, burned with iron-free fuel. Its physical properties are practically the same as those of normal portland cements. It is used principally in ornamental structures in which the white color is desired in the architectural treatment.

High-early-strength Portland Cements High-early-strength portland cements are made from mixtures with high lime content and are ground very fine (about ninety-seven per cent passing through a 200-mesh sieve as compared with about eighty-five per cent for normal portland cements). The time of setting is about the same as for normal portland cements, but they harden and gain strength much more rapidly. Concrete made with high-early-strength portland cement will attain in 72 hr a strength equal to that attained in 28 days by a similar concrete made with normal portland cement.

Fine Aggregate. Fine aggregate usually consists of sand, although stone screenings, or other inert materials with similar characteristics, having clean, hard, strong, durable, uncoated grains free from injurious amounts of dust, lumps, soft or flaky particles, shale, alkali, organic matter, loam, or other deleterious substances, may be used. At least 85 per cent of the fine aggregate should pass

Table 3-6

	Percentage by weight
Passing through a 3/8-in. sieve..	100
Passing through a No. 4 sieve .	85 100
Passing through a No 16 sieve.	45 80
Passing through a No 50 sieve.	2-30
Passing through a No 100 sieve	0-5
Removed by decantation... ...	Not more than 3

through a No. 4 sieve (4 meshes per linear inch) and not more than 5 per cent should pass through a No. 100 sieve. Between these two limits the fine aggregate should be well graded. The gradation recommended by the ASTM is as shown in Table 3-6.

Coarse Aggregate. Coarse aggregate should consist of well-graded crushed stone, washed gravel, or other approved materials of similar characteristics, having clean, hard, strong, durable, uncoated particles free from injurious amounts of soft, thin, or laminated pieces, alkali, and organic or other deleterious matter. Coarse aggregate includes all particles larger than ¼ in.; the maximum size allowable depends upon the class of work for which the resulting concrete is to be used. In small reinforced-concrete work, *i.e.*, slabs, beams, and thin walls, a maximum of ¾ in. is usually specified. For larger reinforced-concrete units, a maximum of from 1½ to 2 in. may be used. For mass concrete, derrick stones, or "plums," varying in size up to large boulders which must be handled by a derrick, are satisfactory and desirable.

Coarse aggregate should be well graded from fine to coarse. A common recommendation is that at least 95 per cent should pass the screen designating the largest size; not less than 40 or more than 75 per cent should pass a screen of half the largest size; not more than 10 per cent should pass a No. 4 sieve or more than 5 per cent a No. 8 sieve.

Artificial Aggregates. The artificial aggregates include well-burned boilerhouse cinders, air-cooled blast-furnace slag, and burned clay. Cinders produce a permeable, nailable, low-strength, and lightweight concrete (90 to 110 lb per cu ft, as compared with 140 to 150 lb for stone or gravel concrete). In the best practice the cinders are screened into two or three size groups—fine, medium, and coarse—and the groups are mixed in proper proportions to form a well-graded aggregate. Sometimes natural sand is used as the fine aggregate with coarse cinders as the coarse aggregate, but the resulting concrete is somewhat heavier than an all-cinder concrete, and the nailability is impaired.

Slag from blast furnaces is a hard though porous material of high compressive strength, which offers a very rough surface for the adhesion of the cement. Slag concrete is satisfactory in many cases for massive construction, but generally, on account of its porosity, it should not be used in thin sections exposed to the action of water. The weight will vary materially, depending upon the grading of the slag and the method of manufacture, from about 70 to 100 lb per cu ft.

Burned-clay aggregates produce concretes of practically any desired strength, averaging about one hundred pounds per cubic foot in weight. Burned-clay aggregates, fine and coarse, are sold under various trade names, such as Haydite, Gravelite, etc. The aggregates themselves weigh from 40 to 65 lb per cu ft.

Water. Water used for concrete should be clean and free from oil, acid, alkali, organic matter, or other deleterious substances. Sea water is not so desirable as fresh water, although, where it has been used in plain concrete structures subject to the weathering action of sea water, no greater disintegration has resulted. Sea water should not be used in reinforced-concrete work because of the corrosive effect of salt water on the steel reinforcement.

Proportioning. It is a generally accepted fact that the strength of mortar or concrete depends entirely upon the water-cement ratio, provided (1) that suitable cement and aggregates are used, (2) that the amounts of the aggregates are such as to produce a workable mix, and (3) that proper curing conditions are maintained during the hardening period. Three problems are involved, therefore, in the design of a concrete mixture: (1) the selection of suitable materials, (2) the selection of the proper water-cement ratio, and (3) the determination of the proper quantities of fine and coarse aggregates. A fourth factor, the method of curing, must be considered after the mixture has been placed properly in its final position.

Proportioning is done either by weight or by volume. Proportioning by weight is more accurate owing to the fact that dampness in fine aggregate, and to a lesser extent in coarse aggregate, has a material effect on the volume of the aggregate. Proportioning by weight is expensive on a small job, however, owing to the cost of the necessary weighing hoppers. If proper allowance is made for "bulking," proportioning by volume can be done with a reasonable degree of accuracy. In the following discussion, except where otherwise stated, proportions are given by volume. In any case, one bag of cement weighing 94 lb is considered to be 1 cu ft. Concrete strengths are invariably specified in terms of the unit ultimate compression strength at the age of 28 days of 8- by 16-in. cylinders, prepared, cured, and tested in accordance with ASTM Standard C39-44. In actual construction, concrete strengths will vary from 1,500 psi for mass concrete where great strength is not required to 4,000 psi for the more important structural units, with the most common values ranging from 2,000 to 3,000 psi. Mixtures will vary from as lean as 1:3:6 for mass construction, such as large abutments and gravity retaining walls, to a rich 1:1½:3 or 1:1:2 for comparatively thin and heavily reinforced structural elements where greater workability, durability, and strength are required. In general, the required strength being known (as assumed in the structural design), the proper water-cement ratio (see below) must be determined, and the proportions of fine and coarse aggregate must be selected so as to produce a plastic mass having the desired workability.

Tests should be made during the progress of actual construction work to check the strength of the concrete which is being used in that work. Quite frequently it is impractical to wait for the results of a 28-day test. In such cases samples may be tested at the age of 7 days, and the probable 28-day strength may be computed from the formula

$$f_{28} = f_7 + 30\sqrt{f_7}$$

in which f_{28} = probable 28-day strength, and f_7 = actual 7-day strength.

Water-Cement Ratio. The selection of the proper water-cement ratio for any desired strength under average job conditions may safely be made from the specification for strength of the American Concrete Institute (1928), which is illustrated graphically in Fig. 3-16. The amount of surface moisture contained in the aggregates must be determined and considered as a part of the mixing water.

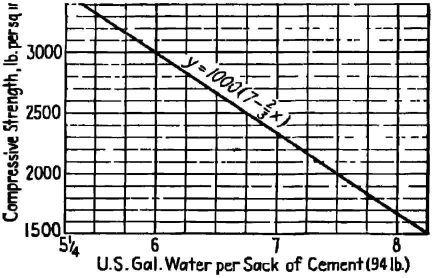

Fig. 3-16.

Quite frequently the strength requirement alone cannot be used as the determining factor in selecting the water-cement ratio. With present-day cements and fair curing conditions, a 2,000-lb concrete, which might economically satisfy ordinary design requirements, may be obtained with a very lean mixture, but the resulting concrete will be neither watertight nor capable of withstanding variations in weather conditions. Maximum water-cement ratios for concretes in various types of constructions and with various conditions of exposure are recommended by F. R. McMillan in "Basic Principles of Concrete Making." These vary from 5½ gal of water per bag of cement for thin sections with extreme climatic exposures to 7½ gal of water per bag of cement for normal structural elements protected from the weather.

Workability. A workable concrete is one that can be worked into all corners of the forms and around the reinforcement without excessive puddling, without segregation of the ingredients, and without the collection of free water on the surface. Workability is influenced by the nature and amount of the fine and coarse aggregates, the water content, and the amount of cement in the mix. A method of measuring the workability, or more properly, the consistency, is the slump test (*ASTM Tentative*

Standard D 138-32T), which is made as follows: A conical shell of 16-gauge galvanized metal with a base 8 in. in diameter, a top 4 in. in diameter and a height of 12 in. is filled to overflowing with concrete rodded into the shell in three separate layers, each receiving 25 strokes of a $\frac{5}{8}$-in. bullet-ended rod 24 in. in length. The excess is carefully struck off, and the mold is at once slowly lifted vertically. The amount of drop of the top of the mass below the original 12-in. height, measured in inches, is known as the "slump." Experience has shown that, to be workable in various types of construction, concrete must have minimum slumps as indicated in Table 3-7.

Table 3-7. Recommended Slumps for Proper Workability

Type of concrete	*Slump, in.*
Mass concrete	3–5
Reinforced concrete:	
Thin sections and columns	6–7
Heavy sections	3–5
Confined horizontal sections	6–9
Pavement concrete	1–4
Mortar for floor finish	2

With a given water-cement ratio, the larger the desired slump, the less is the amount of aggregate that can be included in the mix. Hence, for a given quantity of cement the yield is reduced by an increase in slump. To obtain the maximum economy consistent with placing requirements, the concrete should therefore be made as dry as possible, within the limits of successful construction. Mechanical vibrators, especially in thin-walled sections, will permit the use of slumps somewhat less than those recommended in Table 3-7, and in large constructions the additional cost of vibrating as compared with hand puddling will be more than offset by the saving in materials, and honeycombing will be reduced to a minimum.

Mixing. Practically all concrete is machine-mixed. The materials for each batch are carefully measured or weighed and placed in the rotating drum of the mixer. Most specifications require that the mixing of each batch shall continue for at least 1 min after all the materials, including the water, are in the mixture.

Hand mixing is expensive and difficult, and the resulting concrete is usually not so good as that which is mixed by machine. Hand mixing should be permitted only when a small quantity of concrete is required. Hand mixing is accomplished by placing the proper quantities of cement, fine aggregate, and coarse aggregate on a watertight platform and thoroughly mixing these before the water is added. When the dry mixture shows a uniform texture, the correct amount of water is added and the entire mass is again turned and mixed to a uniform consistency.

Placing. Concrete is conveyed to the forms from the mixer by means of wheelbarrows or two-wheeled buggies or by inclined chutes or pipes. Chutes should ordinarily have a slope not flatter than 1:3 or steeper than 1:2, depending upon the wetness of the mix. The slope of the chute and the consistency of the mix should be so adjusted to each other that the concrete will travel fast enough to keep the chute clean but not so fast as to cause segregation of the materials.

Concrete should not be dumped in one place and allowed to flow horizontally, but should be deposited in approximately uniform layers. Forms should be tight so that cement will not be carried away by the escaping water. In hot, dry weather, wooden forms should be wetted thoroughly on the contact faces to prevent them from absorbing water from the mortar faces in contact with the forms; such absorption causes these faces to be pulled away with the forms upon stripping. Where fresh concrete is to be placed on old concrete or on rock, the old surface should be wetted thoroughly before the new concrete is placed, but it should be free from standing water.

The filling of the forms should be continuous where possible. Where such continuous deposition is impossible, the surface of the old concrete should be cleaned, roughened, and wetted before placing is resumed. Where forms have considerable height, some means should be provided for depositing the concrete without dropping it through too great a distance. Spading or puddling of the concrete is necessary to cause it to spread laterally after it has been deposited in the forms. This puddling should be done in such a manner as to force the coarse aggregate against the forms. In a properly designed mix, sufficient mortar will follow and encase the coarse aggregate to ensure a good finish, but if all the coarse aggregate is forced into the interior, away from the forms, the surface is less durable.

Vibrators are sometimes used to aid in the flow of the concrete in the forms.

They are operated either by compressed air or by electricity. They are used both internally and against the outside of the forms. Since they allow the use of a very sluggish mixture, with its consequent lower water-cement ratio, higher strength concretes may be expected.

Placing under Water. When concrete is placed under water, it is essential that the water be still and not flowing and that the concrete be dumped as a batch in its final position so as to avoid the segregation that would take place if it were allowed to fall freely through several feet of water. For underwater placement, drop-bottom buckets or, preferably, the tremie method may be used. A tremie consists of a vertical pipe, with the lower end resting in the freshly deposited concrete, and with a hopper at the upper end which is above the water. The tremie is kept at least partially filled with fresh concrete during the entire placing operation. From time to time the tremie is lifted slightly to allow part of the concrete in the pipe to flow out the bottom. A stiff but thoroughly plastic mix should be used.

Cold-weather Precautions. If concrete is to be placed during cold weather, particularly in near-freezing temperatures, the materials should be heated before being placed in the mixer, and provision should be made for keeping the concrete warm after it has been placed in the forms. Heating of aggregates on large jobs can be accomplished by placing perforated pipes under the storage piles and forcing steam through the pipes. On small jobs, wood fires built in sheet-iron pipes buried in the storage piles will usually prove satisfactory. The heating should be sufficient to ensure that the temperature of the concrete will be at least 50°F when it reaches the forms.

Curing. Concrete hardens because of chemical reactions which take place between the cement and water. These reactions continue indefinitely, so long as the temperatures are favorable and moisture is present to complete the hydration. The strength and quality of concrete, therefore, are dependent upon its age and the conditions under which it cures.

The strength of concrete, under favorable curing conditions, increases with age This increase is very rapid in the early ages but continues more slowly for an indefinite period The increase in strength from 7 to 28 days is about equal to the increase from 6 months to 2 years. If concrete is cured under dry atmospheric conditions, some of the water in the concrete will evaporate before the chemical reactions have continued for a sufficiently long time to obtain a reasonable proportion of the maximum strength attainable. In general, good practice requires that all concrete should be protected against premature drying out for at least 1 week and for a longer time if the temperature is near the freezing point. This may be done by sprinkling the concrete with water at intervals or by covering it with wet burlap, canvas, or straw.

Tests show clearly that concrete gains strength very slowly, if at all, at near-freezing temperatures and that the strength of similar mixes at any given age increases with an increase in the curing temperature. These facts emphasize (1) the desirability of placing concrete in warm weather if possible, (2) the importance of heating the aggregates and water in cold weather, and (3) the necessity of keeping the concrete warm after it has been placed in the forms when the surrounding air temperature is close to the freezing point. Low temperatures may so delay the hardening that the water in the concrete will evaporate before even a reasonable strength is obtained. When the water has evaporated, no further increase in strength is possible. If, however, evaporation of the water can be prevented and the concrete can be kept from freezing, the concrete will ultimately gain strength when the temperature rises sufficiently to permit the hardening process to continue at its normal rate.

Proper curing, in addition to increasing the strength, has a marked effect on other qualities of concrete, particularly watertightness and resistance to wear. Building up the internal structure of the concrete through more complete hydration of the cement is one of the most effective and economical means of making it impervious to water, a characteristic especially important in walls, tanks, and other elements subject to hydrostatic pressure or severe weather conditions. Continuous damp curing, particularly in the early stages of hardening, helps to produce a hard, dense surface and to prevent checking and dusting, especially important in floors, pavements, and other surfaces subject to abrasive wear.

Modulus of Elasticity. Concrete is not a perfectly elastic material, there being a slight decrease in the ratio of stress to strain (deformation) as the stress increases. Concrete also shows a permanent set under the smallest loads, but within working limits there is a fairly constant relation between stress and strain, which may be considered the modulus of elasticity of the concrete. This varies with the strength and the age of the concrete. In reinforced-concrete design the principal use of the modulus of elasticity is in determining the relative amount of stress carried by the steel and the concrete, assuming that there is perfect bond between the two materials. The value of the ratio n between the modulus of elasticity of the steel and that of the concrete, as ordinarily used in design for both tension and compression, is given by the relation

$$n = 30{,}000/f_c'$$

in which f_c' is the ultimate compressive strength of the concrete at 28 days in pounds per square inch.

Contraction and Expansion. Concretes expand as the temperature is raised and contract as the temperature is lowered. The coefficient of expansion per degree of temperature change increases somewhat with the richness of the mix, but the range of values is small. A value of 0.000006 is usually used in design.

Concretes expand in volume if kept wet or immersed in water and contract if exposed to air. This property is not confined to freshly placed concrete but is characteristic of concretes of many years' service. A concrete that dries out in air may be expected to contract from 0.02 to 0.05 per cent and when immersed in water may expand at least one-half of this amount.

This tendency to change in volume with different moisture conditions and changes in temperature does, of course, set up stresses of both tension and compression in a restrained reinforced-concrete structure. The tensile stresses often exceed the amount that the concrete can sustain, and cracks result. Shrinkage or temperature reinforcement is placed near the surfaces of concrete walls to minimize the formation of such cracks

Bond. The adhesion of new concrete to work previously placed is an important consideration in certain classes of construction. The old concrete should be thoroughly wetted, cleaned, and roughened and coated with cement grout before the new concrete is placed. If the old concrete is not thoroughly wetted, it will draw the moisture from the new concrete, often leaving insufficient moisture in the new concrete for a normal consistency. The result is weak concrete near the joint as well as an inefficient joint. Tests indicate that mere wetting of the old surface results in a joint efficiency of about forty per cent; wetting and roughening about fifty per cent; wetting, roughening, and coating with cement grout about eighty per cent. Several commercial compounds are available which, when mixed with water and brushed on the old surface, result in joints which are just as strong as the concrete in the mass.

Resistance to Fire. Concrete not only is incombustible but also is a poor conductor of heat. Hence it is a splendid fire-resisting and fireproofing material. The low coefficient of expansion of concrete lessens its tendency to crack when heated, and the low conductivity prevents the transference of the heat of the fire to the interior of the mass and to the reinforcing steel.

Weight. The weight of concrete varies with the proportions of the mix, the consistency, and the character of the aggregates. Weights per cubic foot of various types of concrete as usually used are as follows: gravel or crushed-stone concrete, 145 lb if unreinforced, 150 lb, if reinforced; cinder concrete, 110 lb; burned-clay concrete, 100 lb. Patented methods of aerating concrete are in use to some extent. Finely divided chemicals, which generate gases, are added to ordinary concrete, causing it to fluff up so that upon hardening it is full of air spaces. Weights as low as 50 lb per cu ft may be obtained by aerating.

Effect of Admixtures. Materials other than cement and aggregate are sometimes added to concrete (1) to increase the workability, (2) to reduce the freezing temperature of the mixture, (3) to accelerate the hardening, or (4) to make the concrete more watertight.

Small quantities of hydrated lime (up to 15 per cent by weight of the cement) improve the workability, particularly for the leaner mixes. If the lime is added without decreasing the amount of cement or increasing the amount of water, the strength is increased somewhat, but if some of the cement is replaced by the lime, a reduction in strength can be expected.

Sodium chloride (common salt) and calcium chloride lower the temperature at which the water will freeze. The addition of sodium chloride lowers the freezing point about 1°F for each 1 per cent of salt added to the mixing water; up to 12 per cent it is beneficial to the strength of concretes cured at freezing temperatures, but it is detrimental to the strength of concretes cured at normal temperatures. The set is retarded in all cases, and in reinforced concrete the salt is likely to cause corrosion of the steel. Calcium chloride up to 4 per cent is beneficial to the strength of concretes cured at any temperature, and the setting is accelerated. Percentages are expressed in terms of the weight of sodium or calcium chloride as compared with the weight of the mixing water.

The accelerating admixtures that are most generally used are calcium chloride and calcium oxychloride. They do not react alike with all portland cements. The quantity of calcium chloride should not exceed from 2 to 4 per cent by weight of the cement, and the quantity of calcium oxychloride should not exceed from 7 to 10 per cent. The admixture should be dissolved in the mixing water before it is placed in the mixer.

Various waterproofing compounds in powdered or liquid form are sometimes used to make a more impervious concrete. They are either added to the mixing water, mixed with the cement on the job, or added to the cement during its manufacture. Their function is to fill the voids or pores of the concrete with a more or less soapy, insoluble filler. The results obtained are varied, and practically all the compounds in use detract from the strength of the concrete. In general, it is better to depend for watertightness upon correct methods of construction, with the use of sound aggregates of low porosity, a limited amount of mixing water, thorough mixing, proper placing, and favorable curing conditions.

Surface Treatments. Surface treatments are used on concrete floors in industrial buildings to prevent "dusting" and to protect the concrete against corrosion by some oils and acids. Even where comparatively small abrasive forces are present, as in offices and corridors, an untreated concrete surface will dust owing to the grinding effect of the abrasive forces. Thoroughly hardened concrete is unaffected by mineral oils such as ordinary petroleum or pure engine oils. Various animal and vegetable oils may have a slight disintegrating effect. Some acids, such as acetic, carbolic, carbonic, lactic, phosphoric, and tannic, have a slight disintegrating effect, while others, such as hydrochloric, hydrofluoric, muriatic, nitric, sulphuric, and sulphurous, have a marked disintegrating effect. Some salts and alkalies, particularly nitrate and sulphate solutions, have a marked disintegrating effect on concrete cured in the ordinary manner.

Dusting may be minimized by painting the surface with any of the commercial cement paints which are available. To prevent lime burns in the paint film, concrete surfaces less than 6 months old should be treated before painting with a solution of 3 lb of zinc chloride per gallon of water. The surface should be wetted thoroughly with the solution, allowed to dry for at least 24 hr, and then rinsed thoroughly with cold water before painting. Dusting may also be reduced by hardening the surface with two or more applications, applied as above, of a solution of magnesium fluosilicate or zinc fluosilicate. The first application should contain 1 lb of the crystals per gallon of water and the subsequent applications 2 lb per gal.

Treatments required for protection against acids, oils, and alkalies vary with the degree of disintegrating effect. In some cases the magnesium fluosilicate treatment may be sufficient. Better protection is obtained by applying two or three coats of boiled linseed oil, preferably hot, after the magnesium fluosilicate treatment. Sodium silicate (water glass) of about 42.5° Baumé gravity, diluted in proportions of about 1 gal with 4 gal of water and applied in two or more applications, is satisfactory for mild effects. To resist very strong acids it may be necessary to cement vitrified brick or tile units, glass, lead, or sheets of resin, rubber, or synthetic rubber to the concrete with special adhesives.

REINFORCED-CONCRETE BEAMS AND SLABS

Types of Reinforcement. Square and round bars of structural- or intermediate-grade steel, varying in size from ¼ to 1¼ in., are used for reinforcing beams, slabs, walls, and columns. Wire fabric and expanded metal in various forms are used to a considerable extent in slabs, walls, and other thin sections. Bars are almost invariably made with projections or indentations or both on the surfaces to increase the adhesion between them and the concrete. These bars are called "deformed bars,"

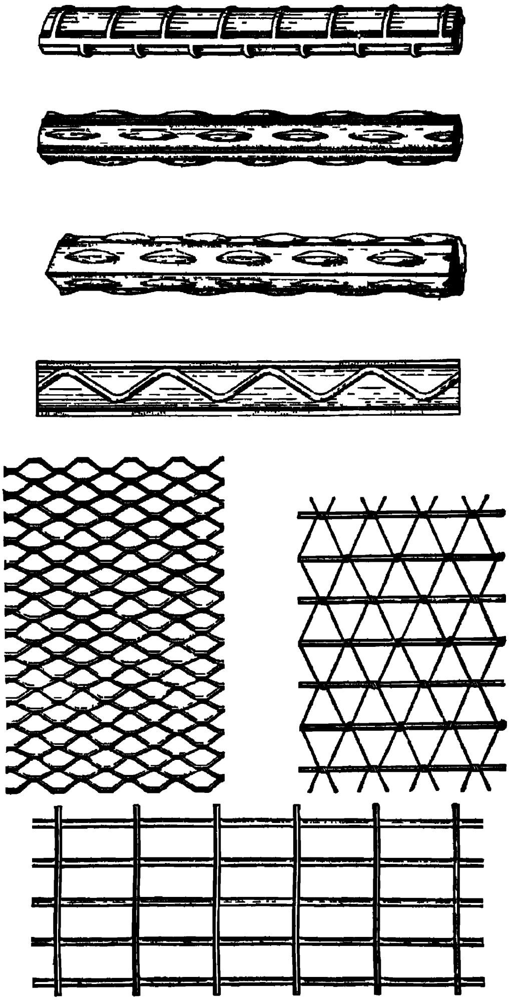

Fig. 3-17. Standard bar and mesh reinforcements.

several types of which are shown in Fig. 3-17. Standard bar sizes are given in Table 3-8. Some of the types of mesh reinforcement are shown in Fig. 3-17.

Table 3-8. Standard Reinforcing Bars

Round bars					Square bars				
Size, in.	Area, sq in.	Perimeter, in.	Weight, lb/lin ft	Size extra, cts/100 lb	Size, in.	Area, sq in.	Perimeter, in	Weight, lb/lin ft	Size extra, cts/100 lb
¼	0 0491	0 785	0 17	100	½	0 2500	2 00	0 85	20
⅜	0 1104	1 178	0 38	40	1	1 0000	4 00	3 40	0
½	0 1963	1 571	0 67	20	1⅛	1 2656	4 50	4 30	0
⅝	0 3068	1 964	1 04	10	1¼	1 5625	5 00	5 31	0
¾	0 4418	2 356	1 50	0					
⅞	0 6013	2 749	2 04						
1	0 7854	3 142	2 67						

Placing the Reinforcement. Most specifications require that the minimum clear distance between parallel bars shall be 1½ times the diameter for round bars and 2 times the side dimension for square bars, but in no case shall the clear spacing between bars be less than 1 in. or less than 1⅓ times the maximum size of the coarse aggregate. The reinforcement of footings and other principal structural members in which the concrete is deposited against the ground shall have not less than 3 in. of concrete between it and the ground contact surface. At surfaces of concrete exposed to the weather, the reinforcement shall be protected by not less than 2 in of concrete. In structures in which the fire hazard is limited, reinforcement at surfaces not exposed directly to ground or weather shall be protected by not less than ¾ in. of concrete for slabs and walls, 1½ in. for beams and girders, and 2 in. for columns. In structures with an abnormal fire hazard, the minimum covering for slab reinforcement shall be 1 in., for beam and girder reinforcement 2 in., and for column reinforcement 3 in.

Standard Design Notation. For convenience of reference, the standard notation, as used in the following equations for the design of reinforced-concrete beams and slabs, is given below:

A_s = effective cross-sectional area of steel in tension, sq in.
A_s' = effective cross-sectional area of steel in compression, sq in.
A_v = total area of web reinforcement in any one plane, sq in.
b = width of rectangular beam, or width of flange of T beam, in.
b' = width of stem of T beam, in.
d = depth from compression surface of beam or slab to center of longitudinal tension reinforcement, in.
d' = depth from compression surface of beam or slab to center of compression reinforcement, in.
E_c = modulus of elasticity of concrete in compression, psi
E_s = modulus of elasticity of steel in tension = 30 million psi
f_c = compressive unit stress in extreme fiber of concrete, psi
f_s = tensile unit stress in longitudinal reinforcement, psi
f_s' = compressive unit stress in longitudinal reinforcement, psi
f_v = tensile unit stress in web reinforcement, psi
j = ratio of lever arm of resisting couple to effective depth d
jd = arm of resisting couple, or distance from center of gravity of tension forces to center of gravity of compression forces, in.
k = ratio of distance from compression surface to neutral plane to effective depth d
kd = distance from compression surface to neutral plane, in.

l = span length of beam or slab
M = bending moment
M_c = resisting moment as governed by the compression forces, in.-lb
M_s = resisting moment as governed by the tension forces, in.-lb
n = ratio of modulus of elasticity of steel to that of concrete = E_s/E_c
Σ_0 = sum of perimeters of bars in one group at a given section, in.
p = ratio of effective area of tension reinforcement to effective area of concrete in beams = A_s/bd
p' = ratio of effective area of compression reinforcement to effective area of concrete in beams = A_s'/bd
r = ratio of unit stress in tension reinforcement to the simultaneous compressive unit stress in extreme fiber of concrete in beams = f_s/f_c
s = spacing of web reinforcement members, measured at the plane of the lower reinforcement and in the direction of the longitudinal axis of the beam, in.
t = thickness of flange of T beam, in.
u = bond stress per unit of area of surface of bar, psi
v = unit shearing stress, psi
v_c = unit shearing stress that can be resisted by concrete alone, psi
V = total shear, lb

Allowable Unit Stresses. The unit stresses recommended by the "Joint Committee"* of the engineering societies are stated in terms of the ultimate unit compressive strength f_c' of the concrete at the age of 28 days, as determined by tests of 6- by 12-in. or 8- by 16-in. cylinders. Concretes used in ordinary constructions have ultimate compressive strengths varying, with the type of member involved, from 1,500 to 4,000 psi, the most common values ranging from 2,000 to 3,000 psi. The required strength of the concrete is obtained by the proper proportioning of the cement, aggregate, and water.

A. Direct Compression	
Bearing on plain concrete piers and pedestals	$0.25f_c'$
B. Compression in Extreme Fiber	
Extreme fiber stress in flexure	$0.45f_c'$
C. Shear	
1. Beams with no web reinforcement	
a. Longitudinal bars anchored	$0.03f_c'$
b. Longitudinal bars not anchored	$0.02f_c'$
2. Beams with web reinforcement	
a. Longitudinal bars anchored	$0.12f_c'$
b. Longitudinal bars not anchored	$0.06f_c'$
D. Bond	
1. Between concrete and plain bars	$0.04f_c'$
2. Between concrete and deformed bars	$0.05f_c'$
E. Reinforcement	
Tensile or compressive unit stress not to exceed	
1. Structural-steel-grade bars	18,000 psi
2. Intermediate- and hard-grade bars	20,000 psi
F. Modulus of Elasticity	
Values of $n = E_s/E_c$	
For $f_c' = 2{,}000$	$n = 15$
For $f_c' = 2{,}500$	$n = 12$
For $f_c' = 3{,}000$	$n = 10$
For f_c' = any value	$n = 30{,}000/f_c'$

Rectangular Beams with Tension Reinforcement. The equations for designing or reviewing reinforced-concrete beams are derived in a manner similar to that used for any homogeneous beam, except that the difference between the elastic properties of steel and concrete and the method of stress distribution require certain obvious modifications in the derivations. The tensile strength of the concrete is disregarded because of its low value, and hence all the tension force is concentrated in the steel; the amount of this tension is equal to the area of the steel in square inches (A_s) multiplied by the unit stress in the steel in pounds per square inch (f_s), as shown in Fig. 3-18, which represents the elevation of a portion of a simply supported rectangular beam and a cross section of the beam. It is assumed that the unit compression force is a maximum at the extreme fiber and decreases uniformly to zero at the neutral axis. The total compression is therefore equal to the area of the shaded triangle in Fig. 3-18 times the width of the beam. This total compression is assumed to be concentrated at the center of gravity of the triangle, which is one-third of the distance *kd* from the upper (or compression) surface. The internal moment in the beam, as determined by the compression forces, is therefore equal to the total compression times the distance *jd* between the centers of gravity of the compression and tension forces. Hence

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* The Joint Committee on Standard Specifications for Concrete and Reinforced Concrete. Its Report on Recommended Practice and Standard Specifications for Concrete and Reinforced Concrete is known as "Joint Code" or "Joint Committee Report."

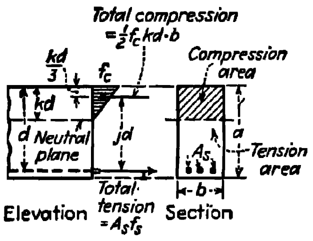


FIG. 3-18.

$$M_c = (\tfrac{1}{2}f_ckd\cdot b)jd = \tfrac{1}{2}f_ckjbd^2$$

Similarly, the internal moment, as determined by the tension forces, is

$$M_s = A_sf_sjd$$

Formulas for the value of *k*, which locates the neutral axis, and for *j*, which determines the lever arm of the resisting couple, and the formula giving the relation between the unit stress in the steel and the unit stress in the concrete, are derived from the usual assumptions made in the theory of flexure, complicated by the fact that the moduli of elasticity of the steel and the concrete are not the same ($E_s = nE_c$). A complete derivation of all these equations is given in any recognized text on reinforced-concrete design. The necessary equations follow (see page 122 for notations):

$$M_c = \tfrac{1}{2}f_ckjbd^2 \qquad (1)$$

$$M_s \quad A_sf_sjd = pf_sjbd^2 \qquad (2)$$

$$f_c = f_sk/[n(1-k)] \qquad (3)$$

$$\sqrt{2pn + (pn)^2} - pn \quad \text{(review only)} \qquad (4)$$

$$k = n/(n+r) \quad \text{(design only)} \qquad (5)$$

$$j = 1 - k/3 \qquad (6)$$

Method of Design. Compute the bending moment caused by the external loads. Assume the weight of the beam and add the bending moment due to this assumed weight to that caused by the external loads to get the total bending moment in the beam. This must be revised later if the actual weight of the beam does not agree with the assumed weight. Determine *k* [Eq. (5)], *j* [Eq. (6)], cross section of beam [Eq. (1)], area of steel [Eq. (2)]. Unless it is governed by architectural or other limitations, make *b* from one-half to three-fourths of *d*, and keep *b* in multiples of 2 in.

Method of Review. Determine *k* [Eq. (4)], *j* [Eq. (6)], M_c or f_c [Eq. (1)], M_s or f_s [Eq. (2)]. If the safe load is required, use the smaller of the values M_c or M_s, deduct the moment due to the weight of the beam, and compute the live load which will produce the remaining moment.

Design of a Rectangular Beam with Tension Reinforcement. Determine the required cross section and area of steel for a simply supported rectangular beam with a span of 18 ft 0 in., which is to carry a uniform live load of 1,000 lb per lin ft. A 2,500-lb concrete and reinforcing bars of structural-grade steel are to be used.

Assume weight of beam 220 lb per lin ft. Total load = 1,220 lb per ft.

$$M = (\tfrac{1}{8} \times 1{,}220 \times 18^2) \times 12 = 593{,}000 \text{ in.-lb}$$

From Eq. (5), $k = 0.430$, and from Eq. (6), $j = 0.857$. Then from Eq. (1),

$$593{,}000 = \tfrac{1}{2} \times 1{,}000 \times 0.430 \times 0.857 \times bd^2$$

$$bd^2 \text{ (required)} \quad 2{,}860 \text{ in.}^3$$

Select $b = 10$ and $d = 17$ in. The total cross section of the beam (see Fig. 3-19) $= 10 \times 20$ in., and the weight $= (10 \times 20/144) \times 150 = 210$ lb per lin ft, approximately as assumed. From Eq. (2),

$$593{,}000 = A_s \times 18{,}000 \times 0.857 \times 17$$
$$A_s \text{ (required)} = 2.26 \text{ sq in.}$$

Three 1-in. round bars will be used, as shown in Fig. 3-19. The purpose of the stirrups, shown in Fig. 3-19, and the method of determining their size and spacing are explained on page 128.

Review of a Rectangular Beam with Tension Reinforcement. A rectangular reinforced-concrete beam has a total cross section of 8 by 14 in. and a length of 20 ft 0 in. It is reinforced with four $\frac{1}{2}$-in. square bars in one row, the center of which is $1\frac{1}{2}$ in above the lower surface of the beam. Assuming a 2,000-lb concrete and structural-grade steel, what is the greatest concentrated load that can be placed on the beam at mid-span?

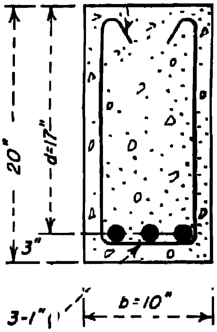


FIG. 3-19.

$$p = \frac{4 \times 0.25}{8 \times 12.5} = 0.0100$$

From Eqs. (4) and (6), $k = 0.418$, and $j = 0.861$.

$$M_c = \tfrac{1}{2} \times 900 \times 0.418 \times 0.861 \times 8 \times (12.5)^2 = 204{,}000 \text{ in.-lb}$$

$$M_s = 1.00 \times 18{,}000 \times 0.861 \times 12.5 = 194{,}000 \text{ in.-lb}$$

Hence, this beam is slightly weaker in tension than it is in compression, and the maximum bending moment cannot exceed 194,000 in.-lb. Owing to its own weight (117 lb per lin ft) the bending moment is $\frac{1}{8} \times 117 \times (20)^2 \times 12 = 70{,}200$ in.-lb. The maximum moment available for the concentrated load is then

$$194{,}000 - 70{,}200 = 123{,}800 \text{ in.-lb}$$

The maximum bending moment caused by the concentrated load is

$$\tfrac{1}{4}Pl = \tfrac{1}{4}P \times 20 \times 12 = 60P \text{ in.-lb}$$

hence $60P = 123{,}800$, and P (maximum allowable) $= 2{,}060$ lb.

Slabs Supported on Two Sides. A slab supported on only two sides is essentially a rectangular beam of comparatively large ratio of width to depth. Such a slab is analyzed by cutting out a 12-in. strip at right angles to the supporting beams and applying the same formulas as given for rectangular beams on page 124. Slabs are usually designed for uniformly distributed loads only, and the load per square foot on the slab becomes the load per linear foot on the imaginary 12-in. strip. Since all the load on the slab must be transmitted to the two supporting beams, it follows that all the reinforcing steel should be placed at right angles to these beams, with the exception of any bars that may be placed in the other direction to take care of shrinkage and temperature stresses. A slab that is so supported thus consists (in theory) of a series of rectangular beams 12 in. wide, placed side by side. The steel ratio p is equal to the area of one bar divided by the product of the effective depth d and the distance between bars, center to center. The areas furnished per foot of slab width by various sizes of bars and spacings are given in Table 3-9.

Slabs may be of one span only, resting freely on the supports, in which event they are designed for a moment of $\frac{1}{8}wl^2$. They may, however, be continuous over several supports, in which case moment coefficients for continuous or partly continuous beams may be used. Usually, for slabs of only two spans, a moment of $\frac{1}{8}wl^2$ is used, while for slabs of several spans, a moment of $\frac{1}{12}wl^2$ is used for the interior spans and $\frac{1}{10}wl^2$ for the end spans. Continuous slabs must have some steel near the upper surface over the supports to provide for the tension near this surface caused by the negative moment. This negative-moment steel is usually furnished by bending up some of the positive-moment steel at about the quarter points of the span and continuing this steel near the top surface to the corresponding points in the adjacent spans.

Table 3-9. Areas of Bars in Slabs, Sq In. per Ft

Spacing, in.	Size of round bar, in.							Size of square bar, in.			
	¼	⅜	½	⅝	¾	⅞	1	½	1	1⅛	1¼
3	0 20	0 44	0 78	1 23	1 77	2 40	3 14	1 00	4 00	5 06	6 25
3½	0 17	0 38	0 67	1 05	1 51	2 06	2 69	0 86	3 43	4 34	5 36
4	0 15	0 33	0 59	0 92	1 32	1 80	2 36	0 75	3 00	3 80	4 69
4½	0 13	0 29	0 52	0 82	1 18	1 60	2 09	0 67	2 67	3 37	4 17
5	0 12	0 26	0 47	0 74	1 06	1 44	1 88	0 60	2 40	3 04	3 75
5½	0 11	0 24	0 43	0 67	0 96	1 31	1 71	0 55	2 18	2 76	3 41
6	0 10	0 22	0 39	0 61	0 88	1 20	1 57	0 50	2 00	2 53	3 12
6½		0 20	0 36	0 57	0 82	1 11	1 45	0 46	1 85	2 34	2 89
7		0 19	0 34	0 53	0 76	1 03	1 35	0 43	1 71	2 17	2 68
7½		0 18	0 31	0 49	0 71	0 96	1 26	0 40	1 60	2 02	2 50
8		0 17	0 29	0 46	0 66	0 90	1 18	0 37	1 50	1 89	2 34
9		0 15	0 26	0 41	0 59	0 80	1 05	0 33	1 33	1 69	2 08
10		0 13	0 24	0 37	0 53	0 72	0 94	0 30	1 20	1 52	1 87
12		0 11	0 20	0 31	0 44	0 60	0 78	0 25	1 00	1 27	1 56

Design of One-way Slab. Design a fully continuous reinforced-concrete slab, supported on two sides only, to sustain a live load of 120 psf. Each span of the slab is 11 ft 0 in. Use allowable unit stresses of 800 and 18,000 for the concrete and steel, respectively, with $n = 15$.

Assume weight of slab as 62 psf (5-in. thickness). For a 12-in. strip at right angles to the supports,

$$M = \tfrac{1}{12} \times (120 + 62) \times (11)^2 \times 12 = 22{,}000 \text{ in.-lb}$$

From Eqs. (5) and (6), $k = 0\ 400$ and $j = 0\ 867$

$$22{,}000 = \tfrac{1}{2} \times 800 \times 0\ 400 \times 0\ 867 \times 12 \times d^2$$
$$d \text{ (required)} = 3.6 \text{ or } 4 \text{ in}$$

The total thickness, with 1-in. insulation, is 5 in., and the slab weight is 62 psf, as assumed.

$$72{,}000 = A_s \times 18{,}000 \times 0\ 867 \times 4$$
$$A_s \text{ (required)} = 0\ 353 \text{ sq in.}$$

Use ½-in. round bars on 6½-in. centers.

Slabs Supported on Four Sides. When a slab is square or nearly so and there are beams at the four edges of the panel, the slab should be reinforced in two directions so as to transmit the total load to all four beams. If the panel is square and if the construction is such that the same degree of restraint exists at each edge, one-half of the total load will be transmitted to each pair of beams. If the panel is longer in one direction than in the other, more than one-half of the load will be transmitted in the shorter direction, and the remainder will be transmitted in the longer direction.

A satisfactory assumption for distribution for ordinary panels is that recommended in the *Joint Building Code* (1928), where the part w_1 of the total load w, which is transferred in the short direction (see Fig. 3-20), is

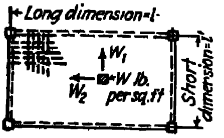


FIG. 3-20.

$$w_1 = \left(\frac{l}{l'} - \frac{1}{2}\right) w \tag{7}$$

in which l = the longer dimension and l' the shorter dimension, both in feet. Then, if w_2 is the part of the total load that is transferred in the longer direction,

$$w_2 = w - w_1 \tag{8}$$

Two strips, each with a width of 12 in., must be designed (or investigated), one strip parallel to the short dimension of the slab and one parallel to the long dimension. The values of d for each strip differ by one bar diameter. The shorter bars are usually placed beneath the longer bars, with the two layers in direct contact.

Design of Two-way Slab. A typical interior (fully continuous) floor panel is to be 9 ft 0 in. × 10 ft 0 in. in plan. The slab is to be reinforced in two directions and is to carry a uniform live load of 300 psf. Assume $f_c = 1{,}000$, $f_s = 18{,}000$, and $n = 12$. Design the slab.

Assume weigh of slab as 50 psf (4-in. slab); then $w = 300 + 50 = 350$ lb/ft. *For a 12-in. strip in the short direction,*

$$w_1 = ({}^{10}\!/_{9} - {}^{1}\!/_{2})350 = 215 \text{ lb/ft}$$
$$M = {}^{1}\!/_{12} \times 215 \times (9)^2 \times 12 = 17{,}500 \text{ in.-lb}$$

From Eqs. (5) and (6), $k = 0.400$ and $j = 0.867$.

$$17{,}500 = {}^{1}\!/_{2} \times 1{,}000 \times 0.40 \times 0.867 \times 12 \times d^2$$
$$d \text{ (required)} = 2.9 \text{ in.}$$

Use $d = 3$ in.; with 1-in. insulation, total thickness = 4 in., as assumed.

For a 12-in. strip in the long direction, $w_2 = 350 - 215 = 135$ lb/ft.

$$M = {}^{1}\!/_{12} \times 135 \times (10)^2 \times 12 = 13{,}500 \text{ in.-lb}$$
$$13{,}500 = {}^{1}\!/_{2} \times 1{,}000 \times 0.400 \times 0.867 \times 12 \times d^2$$
$$d \text{ (required)} = 2.55 \text{ in.}$$

With a 4-in. slab, the d furnished for the long bars (assumed as ½ in. in diameter) is only $4 - 1 - {}^{1}\!/_{2} = 2.50$ in. While sufficiently close for practical purposes, theoretically the slab should be made thicker to furnish at least 2.55 in. of effective depth for the long (upper) bars. A 4¼-in. slab will be used. Then, for the short bars, $d = 3.25$ in., and for the long bars, $d = 2.75$ in.

Short Bars:

$$17{,}500 = A_s \times 18{,}000 \times 0.867 \times 3.25$$
$$A_s = 0.345 \text{ sq in.}$$

From Table 3-9, ½-in. round bars, 6½ in. center to center are satisfactory for the short direction.

Long Bars:

$$13{,}500 = A_s \times 18{,}000 \times 0.867 \times 2.75$$
$$A_s = 0.314 \text{ sq in}$$

From Table 3-9, ½-in. round bars, 7½ in. center to center are satisfactory for the long direction.

Shear and Bond. In addition to the design for bending stresses as previously discussed, reinforced-concrete beams must be designed to resist shear and bond stresses. The latter tend to cause the steel to pull away from the concrete when under stress and thus destroy the unity of the beam.

The maximum unit shear v (pounds per square inch) at any cross section is

$$v = V/bjd \qquad \text{(rectangular beams)} \qquad (9)$$
$$v = V/b'jd \qquad \text{(T beams)} \qquad (10)$$

In any beam, the absolute maximum unit shear occurs at the section where the total shear V is a maximum. Allowable unit stresses are given on page 123.

The maximum unit bond stress u (pounds per square inch of contact surface between the longitudinal reinforcement and the concrete) occurs at the section where the shear is a maximum and is equal to

$$u = V/\Sigma_0 jd \qquad (11)$$

in which Σ_0 = sum of the perimeters of all the tension bars at the section. In all computations for shear and bond, a value of $j = {}^{7}\!/_{8}$ may be used.

Example: A rectangular beam with an over-all cross section of 12 by 22 in. supports live loads as shown in Fig 3-21. The beam is reinforced with four $\frac{3}{4}$-in round bars in one row, the center of which is 2 in. above the lower surface of the beam A 2,000-lb concrete is assumed Determine the maximum unit shearing stresses in sections at the left and right supports and in sections just to the left and right of the concentrated load Determine also the maximum unit bond stress

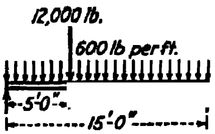


Fig. 3-21.

The dead weight of the beam is 275 lb per lin ft, hence the total uniform load is 875 lb per ft. At the left support,

$$V = (\tfrac{2}{3} \times 12{,}000) + (\tfrac{15}{2} \times 875) = 14{,}560 \text{ lb}$$

$$v = \frac{14{,}560}{12 \times \frac{7}{8} \times 20} = 70 \text{ psi}$$

At the right support,

$$V = (\tfrac{1}{3} \times 12{,}000) + (\tfrac{15}{2} \times 875) = 10{,}560 \text{ lb}$$

$$v = \frac{10{,}560}{12 \times \frac{7}{8} \times 20} = 50 \text{ psi}$$

Just to the left of the concentrated load,

$$V = 14{,}560 - (5 \times 875) = 10{,}180 \text{ lb}$$

$$v = \frac{10{,}180}{12 \times \frac{7}{8} \times 20} = 49 \text{ psi}$$

Just to the right of the concentrated load,

$$V = 14{,}560 - (5 \times 875) - 12{,}000 = -1{,}820 \text{ lb}$$

$$v = \frac{1{,}820}{12 \times \frac{7}{8} \times 20} = 9 \text{ psi}$$

The maximum unit bond stress occurs at the left support, since the shear is a maximum at this point

$$\frac{14{,}560}{4 \times 2.356 \times \frac{7}{8} \times 20} = 88 \text{ psi}$$

The above computations indicate that the beam is safe from shearing or bond failures provided adequate web reinforcement is used throughout those portions of the span where the unit shear is greater than $0.02 \times 2{,}000 = 40$ psi The design of the web reinforcement is given in the following paragraphs

Diagonal Tension and Web Reinforcement. Inclined tension stresses are produced by a combination of vertical and horizontal shearing forces and the flexural stresses in the tension fibers These stresses tend to cause diagonal cracks in regions of large shearing stresses, as shown in Fig 3-22 The magnitude of the diagonal-tension stress varies primarily with the amount of the shearing stress at the section By limiting the allowable unit shearing stress to a value which has been found by actual tests to be low enough to ensure against failure by diagonal tension, it may be considered that the danger of such failure has been eliminated.

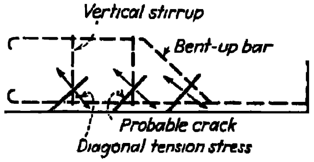


Fig. 3-22.

When the unit shearing stress at any section exceeds $0.02f_c'$ (or $0.03f_c'$ if the longitudinal bars are hooked at the ends), some form of web reinforcement must be used at that section in order to prevent the diagonal cracks from opening up Separate vertical stirrups may be used for web reinforcement, or some of the longitudinal bars may be bent up where they are no longer required for the tension fiber stresses so as to cross the planes of the probable cracks and thus prevent the formation of the cracks. Stirrup bars are usually bent in the form of the letter U and placed as shown in Fig. 3-19 The effective area A_v of the stirrup is then twice the area of the bar. For very wide beams, more legs than two might be used, in which case the area A_v would be equal to the number of vertical legs times the area of the bar.

The size and spacing of vertical stirrups (if required) are determined from Eq. (12). The spacing should not exceed about $\frac{1}{2}d$.

$$s = \frac{A_v f_v jd}{V - V_c} \tag{12}$$

In Eq. (12), V is the total shear at the section under consideration, and V_c is the amount of shear that can be taken by the concrete itself, i.e., $V_c = v_c bjd$ for rectangular beams, or $V_c = v_c b'jd$ for T beams, in which v_c is either $0.02f_c'$ or $0.03f_c'$.

EXAMPLE. Referring to the example on page 128, if the longitudinal bars are not hooked at the ends, the allowable unit shear that can be resisted by the concrete is $0.02 \times 2{,}000 = 40$ psi. Over what portions of the beam is web reinforcement required, and what arrangement of ¼-in. round U stirrups is necessary to provide for the diagonal-tension stresses?

A study of the unit-shear computations on page 128 shows that web reinforcement is required from the left support to the concentrated load and from the right support for a distance x to the section at which the unit shear is 40 lb per sq in. This distance is obtained from Eq. (9) as follows.

$$40 = \frac{10{,}560 - 875x}{12 \times \frac{7}{8} \times 20}$$

$$x = 2.47 \text{ ft}$$

The maximum spacing of stirrups at the left support, from Eq. (12), is

$$s = \frac{2 \times 0.0491 \times 16{,}000 \times \frac{7}{8} \times 20}{14{,}560 - (40 \times 12 \times \frac{7}{8} \times 20)} = 4.4 \text{ in.}$$

Just at the left of the concentrated load the total shear is 10,180 lb, and

$$s = \frac{2 \times 0.0491 \times 16{,}000 \times \frac{7}{8} \times 20}{10{,}180 - (40 \times 12 \times \frac{7}{8} \times 20)} = 15.5 \text{ in.}$$

Remembering that the spacing of vertical stirrups should not exceed $\frac{1}{2}d$, or 10 in. in this case, by simple proportion the following selection of spacings may be made for the region between the left support and the concentrated load, placing the first stirrup about two inches from the edge of the support: three at four in., two at six in., and four at ten in.

At the right support the total shear is 10,560 lb, and $s = 12.7$ in.; four stirrups will be used at this end, spaced 10 in. on centers.

T Beams. When a reinforced-concrete floor slab is constructed as a monolith with the supporting beam, part of the slab may be assumed to assist the upper part of the beam in resisting compressive stresses. These two acting together constitute what is known as a T beam (Fig. 3-23). The slab is called the "flange," and the portion of the beam below the slab is called the "web" or "stem." The exact width of slab that can be assumed effective in resisting compressive stresses depends upon the thickness of the slab, the span of the beam, and the spacing of beams. This effective width b is determined as follows.

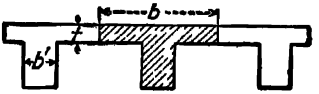


FIG. 3-23. T beam.

1. It shall not exceed one-fourth of the span length of the beam, except that, for beams with a flange on one side only, it shall not exceed one-tenth of the span.
2. Its overhanging width on either side of the stem shall not exceed eight times the thickness of the slab.
3. It shall not exceed the distance center to center of beams.

Owing to the relatively large width of flange, it is safe to say that the compressive strength of a T beam which is part of a floor system will seldom (if ever) govern the design. The effective cross section $b'd$ is determined from the shearing requirement, Eq. (10), with $v = 0.06f_c'$, web reinforcement being provided as necessary and in accordance with the methods outlined on page 128. In all computations involving shearing strength of T beams, the width of the stem b' is used, instead of the width of flange b, because of the fact that the unit shear involved is the unit shear along any horizontal plane between the tension steel and the neutral axis.

Formulas for the resisting moments, and for k and j, are derived in a manner similar to that used for similar functions of rectangular beams. The small amount of compression in the stem (between the neutral axis and the bottom of the slab) is disregarded, so that the total compression is considered to be a trapezoid with bases equal to f_c and $f_c[(kd - t)/kd]$, an altitude equal to t, and a length equal to the effective width of the flange b as shown in Fig. 3-24. Approximate resisting-moment formulas for use in design are obtained by assuming that $jd = d - \frac{1}{2}t$ and that the average compression stress in the trapezoid of depth t is $\frac{1}{2}f_c$. The essential equations for T-beam design and review are tabulated below:

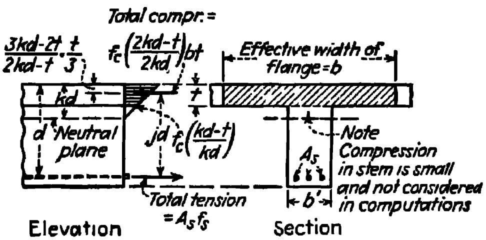

Total compr. = $f_c\left(\frac{2kd-t}{2kd}\right)bt$
$\frac{3kd-2t}{2kd-t}\cdot\frac{t}{3}$
Effective width of flange = b
kd
t
d
Neutral plane
jd
$f_c\left(\frac{kd-t}{kd}\right)$
A_s
Note Compression in stem is small and not considered in computations
Total tension $=A_sf_s$
b'
Elevation
Section

FIG. 3-24.

$$b = \tfrac{1}{4}\text{ span} \quad \text{or} \quad 16t + b' \quad \text{(maximum)} \tag{13}$$

$$p = A_s/bd \tag{14}$$

$$M_s = A_s f_s jd \tag{15}$$

$$M_s = A_s f_s(d - \tfrac{1}{2}t) \quad \text{(approximate, for design only)} \tag{16}$$

$$M_c = f_c\left(1 - \frac{t}{2kd}\right) bt \cdot jd \tag{17}$$

$$M_c = \tfrac{1}{2}f_c bt(d - \tfrac{1}{2}t) \quad \text{(approximate, for design only)} \tag{18}$$

$$b'd = V/vj \tag{19}$$

$$f_c = \frac{f_s k}{n(1 - k)} \tag{20}$$

$$k = \frac{np + \frac{1}{2}(t/d)^2}{np + (t/d)} \tag{21}$$

$$j = \frac{6 - 6(t/d) + 2(t/d)^2 + (t/d)^3(1/2pn)}{6 - 3(t/d)} \tag{22}$$

Method of Design. Determine $b'd$ [Eq. (19)] and select proper values for each (b' is usually from $\frac{1}{3}$ to $\frac{1}{2}d$); A_s [Eq. (16)]. If a more exact design is required, review the beam (see following paragraph) to determine f_s and f_c. If f_c is too great, the beam must be deepened; if f_s is too small, less steel may be used and the review repeated. In rare cases only will the cross section be governed by the bending stress in the concrete.

Method of Review. Determine k [Eq. (21)], j [Eq. (22)], M_c or f_c [Eq. (17)], M_s or f_s [Eq. (15)]. In all computations $p = A_s/bd$ [Eq. (14)], in which b is the width of the flange [Eq. (13)]. After the resisting moment has been determined, the safe live load may be computed as explained on page 125.

Design of a T Beam. A series of parallel continuous floor beams is spaced 11 ft 0 in. on centers, supporting a monolithic 5-in. slab which sustains a live load of 100 psf. The span of the beams is 23 ft 0 in. Design one of the beams, assuming $f_s = 20{,}000$ and $f_c' = 2{,}000$.

Weight of slab = 62 psf. Load on beam from slab = 11(100 + 62) = 1,780 lb per ft. Assume weight of stem = 200 lb per ft; then the total load on the beam is 1,780 + 200 = 1,980 lb per ft.

$$M = \tfrac{1}{12} \times 1{,}980 \times (23)^2 \times 12 = 1{,}045{,}000 \text{ in.-lb}$$

$$V = 1{,}980 \times \tfrac{23}{2} = 22{,}700 \text{ lb}$$

$$b'd = \frac{22{,}700}{\tfrac{7}{8} \times 120} = 215 \text{ sq in.}$$

Select $b' = 10$ in., and $d = 22$ in. Allowing for two rows of bars, the total height of the beam must be $22 + 3 = 25$ in. (see Fig. 3-25). The cross section below the slab is 10×20 in., and the weight of the stem is 210 lb per ft, approximately as assumed.

$$A_s = \frac{1{,}045{,}000}{20{,}000 \times (22 - 2.5)} = 2.68 \text{ sq in.}$$

Six $\tfrac{3}{4}$-in. round bars (area 2.65 sq in.) will be used. The required size and spacing of web reinforcement bars are obtained as explained on pages 128 and 129.

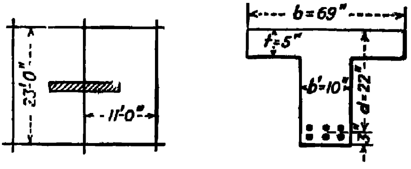


Fig. 3-25.

Beams Reinforced for Tension and Compression. *Design.* Quite frequently a beam is limited by certain structural conditions or architectural limitations to a size which is not adequate to furnish the required compression resistance. Additional compression resistance is furnished in such cases by placing reinforcing bars near the extreme compression surface of the beam, of sufficient area to provide for the moment in excess of the carrying capacity of the concrete Formulas for the design of a beam reinforced for tension and compression are derived by determining first the amount of moment (M_1) which the given concrete area can develop in compression [Eqs. (1), (5), and (6)] and computing the area of tension steel (A_{s1}) required to develop this moment [Eq. (23)]

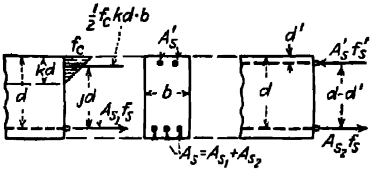


Fig. 3-26.

$$A_{s_1} = M_1/f_s jd \tag{23}$$

If the total moment (M) in the beam is greater than M_1, the amount (M_2) still to be taken care of by the compression steel (A_s') and additional steel (A_{s_2}) is $M - M_1$. The lever arm of this additional stress couple (Fig. 3-26) is $d - d'$. Hence

$$M_2 = A_{s_2} f_s(d - d')$$

from which

$$A_{s_2} = \frac{M_2}{f_s(d - d')} \tag{24}$$

The total tension steel area required to develop the moment M is, then,

$$A_s = A_{s_1} + A_{s_2} \tag{25}$$

The area of compression steel required is determined from the assumptions that, for equilibrium, the additional tension resistance must be equal to the added compression resistance, and that unit stresses vary as the distance from the neutral axis. Thus,

$$A_s' f_s' = A_{s_2} f_s$$

$$\frac{f_s}{f_s'} = \frac{d - kd}{kd - d'}$$

Hence

$$A_s' = A_{s_2} \times \frac{1 - k}{k - (d'/d)} \tag{26}$$

Example: A simply supported rectangular beam with a span of 20 ft 0 in. is limited in cross section to 8 by 18 in. The beam supports a live load of 580 lb per lin ft over its entire length. Using 2 in. of insulation and assuming $f_s = 18{,}000$, $f_c = 1{,}000$, and $n = 12$, determine the area of steel required for tension (A_s) and for compression (A_s').

The weight of the beam is 150 lb per ft; hence

$$M = \tfrac{1}{8}(580 + 150) \times (20)^2 \times 12 = 438{,}000 \text{ in.-lb}$$

From Eqs. (5) and (6), $k = 0.400$, and $j = 0.867$.

$$M_1 = \tfrac{1}{2} \times 1{,}000 \times 0.400 \times 0.867 \times 8 \times (16)^2 = 355{,}000 \text{ in.-lb}$$

$$M_2 = 438{,}000 - 355{,}000 = 83{,}000 \text{ in.-lb}$$

$$A_{s_1} = \frac{355{,}000}{18{,}000 \times 0.867 \times 16} = 1.42 \text{ sq in.}$$

$$A_{s_2} = \frac{83{,}000}{18{,}000(16 - 2)} = 0.33 \text{ sq in.}$$

$$A_s = 1.42 + 0.33 = 1.75 \text{ sq in.}$$

$$A_s' = 0.33 \times \frac{1 - 0.44}{0.400 - {}^2\!/_{16}} = 0.72 \text{ sq in.}$$

Four ¾-in. round bars in tension and two ¾-in. round bars in compression are selected; each set is placed in one row, the center of which is 2 in. from the bottom and top surfaces of the beam, respectively.

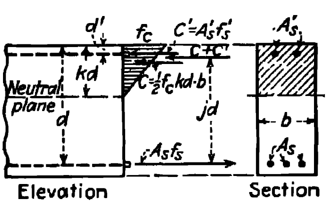


FIG. 3-27.

Review. The equations for k, j, and the resisting moments to be used in reviewing a doubly reinforced rectangular beam are derived in a manner similar to that used in deriving the corresponding equations for rectangular beams with tensile reinforcement only. The resulting equations are as follows (see Fig. 3-27):

$$p = A_s/bd \tag{27}$$

$$p' = A_s'/bd \tag{28}$$

$$M_s = A_s f_s jd \tag{29}$$

$$M_c = \frac{1}{2} f_c k \left(1 - \frac{k}{3}\right) bd^2 + f_s' p' bd(d - d') \tag{30}$$

$$f_c = \frac{f_s k}{n(1 - k)} \tag{31}$$

$$f_s' = \frac{f_s \left(k - \dfrac{d'}{d}\right)}{1 - k} \tag{32}$$

$$k = \sqrt{2n\left(p + p' \cdot \frac{d'}{d}\right) + n^2(p + p')^2} - n(p + p') \tag{33}$$

$$j = \frac{k^2 - \dfrac{1}{3}k^3 + 2p'n\left(k - \dfrac{d'}{d}\right)\left(1 - \dfrac{d'}{d}\right)}{k^2 + 2p'n\left(k - \dfrac{d'}{d}\right)} \tag{34}$$

If the unit stresses for a given loading are required, determine k [Eq. (33)], j [Eq. (34)], f_s [Eq. (29)], f_s' [Eq. (32)], f_c [Eq. (31)]. If the resisting moment of a given beam is required, determine k [Eq. (33)], j [Eq. (34)], M_s [Eq. (29)] and M_c [Eq. (30)]. The smaller of the two latter values is the resisting moment of the beam.

REINFORCED-CONCRETE COLUMNS

Types of Columns. The two general types of columns are (1) square or rectangular columns reinforced with longitudinal bars which are stayed laterally with ¼-or ⅜-in. ties spaced from 8 to 12 in. on centers (Fig. 3-28*a*); (2) round or octagonal columns with longitudinal bars wrapped with a continuous spiral wire ¼ to ⅝ in. in diameter, with a pitch (spacing) of from 1½ to 3 in. (Fig. 3-28*b*).

Columns with Longitudinal Reinforcement and Lateral Ties. Columns of this type which support concentric loads are designed by Eq. (35).

$$P = 0.18f_c'A_g + 0.8A_sf_s = 0.18f_c'A_g + 0.8pA_gf_s \qquad (35)$$

in which P = total load that the column can or must support
A_g = over-all cross-sectional area of the column
A_s = area of longitudinal steel
p = steel ratio = A_s/A_g
f_c' = ultimate strength of the concrete in compression
f_s = the working stress in the longitudinal reinforcement, at present specified as 16,000 psi for intermediate-grade steel

The longitudinal reinforcement should consist of at least four bars with a minimum diameter of $\frac{5}{8}$ in., and the value of the steel ratio $p = A_s/A_g$ should not be less than 0.01 or more than 0.04. The lateral ties should be at least $\frac{1}{4}$ in. in diameter, spaced not more than 48 tie diameters, or 16 bar diameters, apart, and so arranged as to support all the bars against lateral displacement when the concrete is placed in the form. In designing such a column, the value of p may be assumed (usually the minimum value of p will give the most economical, although the largest, column) and the required value of A_g can be obtained from the second form of Eq. (35). The size of the column can then be selected and the value of A_s obtained from the relation $A_s = pA_g$. The thickness of concrete outside the lateral ties should be at least $1\frac{1}{2}$ in.

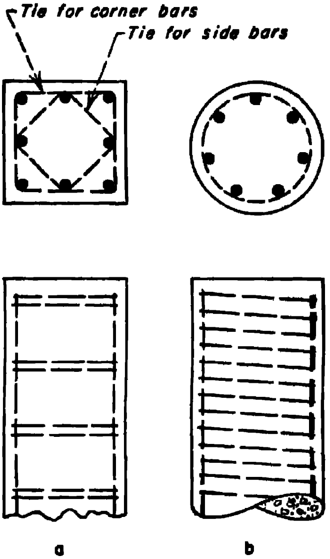


Fig. 3-28 Types of columns.

Example: Design a square column, using the minimum amount of reinforcement, to support an axial load of 240,000 lb, assuming f_c' = 3,000 psi and using intermediate-grade steel.

$$240{,}000 = 0.18 \times 3{,}000A_g + 0.8 \times 0.01A_g \times 16{,}000$$
$$A_g = 360 \text{ sq in.} \qquad \text{19-in. square column}$$
$$A_s = 0.01 \times 360 = 3.6 \text{ sq in.}$$
Use six $\frac{7}{8}$-in. round bars

Columns with Longitudinal Reinforcement and Spirals. Whenever a material is subjected to compression in one direction, there will be an expansion in the direction perpendicular to the compression axis. Spirals tend to resist this expansion and thus develop lateral compressive stresses which neutralize to some extent the effect of the axial stress. Such columns can, therefore, support greater loads than the same size of column without spirals. Equation (36) is used in the design of spirally reinforced columns which support concentric loads.

$$P = 0.225f_c'A_g + A_sf_s = 0.225f_c'A_g + pA_gf_s \qquad (36)$$

in which the terms have the same significance as in the preceding example. The design procedure is also the same.

At least six bars with a minimum diameter of $\frac{5}{8}$ in. should be used, and the value of p should be not less than 0.01 or more than 0.08. The size of spiral wire and the spacing or pitch of the spirals are obtained from the fact that the ratio p' of the volume of spiral reinforcement to the volume of the spiral core (out to out of spirals) should not be less than

$$p' = 0.45(R - 1)\frac{f_c'}{f_s'}$$

in which R = ratio of over-all area of column to core area, and f_s' = useful limit stress of spiral reinforcement, 40,000 psi for hot-rolled bars of intermediate-grade, 50,000 for hard-grade, and 60,000 for cold-drawn wire.

The center to center spacing of the spirals should not exceed one-sixth of the core diameter, and the clear distance between spirals should not be more than 3 in or less than $1\frac{3}{8}$ in The thickness of concrete outside the spirals should not be less than $1\frac{1}{2}$ in

Example Design a round column with spirals to support an axial load of 262 000 lb, assuming $f_c' = 3{,}000$ psi and using the minimum amount of intermediate-grade reinforcement

$$262\,000 = 0\ 225 \times 3\,000 \times A_g + 0\ 01 \times A_g \times 16\,000$$
$$A_g = 314 \text{ sq in} \qquad \text{Diameter of column} = 20 \text{ in}$$
$$A_s = 0\ 1 \times 314 = 3\ 14 \text{ sq in} \qquad \text{Use eight } \tfrac{3}{4} \text{ in round bars}$$

Hot-rolled intermediate-grade spirals $\frac{3}{8}$ in in diameter with a pitch of 2 in, are required to satisfy the above equation for p'

FLOOR SYSTEMS FOR CONCRETE BUILDINGS

Floor and Roof Loads. The minimum live loads for which the floors and the roof of any building must be designed are always specified in the building code that governs the site of the construction Typical minimum live-load values are given in Table 3-10 In all cases the dead weight of the floor and surfacing must be included in the total design load When plastered ceilings are specified, as is frequently the case where ribbed floor construction is used, an additional allowance of from 10 to 15 lb per sq ft is made for the lath and plaster In certain types of buildings particularly office buildings where all partitions are not definitely located on the plans, an extra allowance of from 10 to 20 lb per sq ft of floor area is frequently made for the weight of these partitions

Table 3-10. Average Minimum Live Loads for Buildings, Lb per Sq Ft

Apartments	50	Office buildings first floor	100
Assembly halls	100	Upper floors	75
Dwellings	50	Roofs slope under 20°	40
Hospitals	60	Schools	75
Hotels lobby floor	80	Stables garages	100
Upper floors	60	Stores and warehouses	200
Manufacturing buildings	150	Stairways	75 100

Types of Floor Systems. The different floor systems that are commonly used in reinforced-concrete buildings may be divided into five general classes as follows

1 Beam-and-girder floors
2 Beam-and-slab floors
3 Flat-slab floors
4 Ribbed floors, with clay-tile, gypsum-tile or steel-tile fillers
5 Steel-joist or precast-concrete-joist floors

Beam-and-girder Floors The beam-and girder floor consists of a series of parallel beams supported at their extremities by girders which in turn frame into concrete columns placed at more or less regular intervals over the entire floor area This framework is covered by a reinforced-concrete slab the load from which is transmitted first to the beams and thence to the girders and columns The beams are usually spaced so that they come at the mid-points, at the third points or at the quarter points of the girders As the slabs, beams, and girders are built monolithically, the beams and girders are designed as T beams, and advantage is taken of continuity

Beam-and-slab Floors Beam-and-slab floors are those in which a solid reinforced-concrete slab is supported on beams that transmit the loads directly to the columns The beams are framed into the columns, usually in both rectangular directions the slab is poured monolithically with the beams, with reinforcement parallel to each side of the floor panel Occasionally the beams in one direction are omitted, and the slab is supported on the beams between the columns in the other direction the main slab reinforcement is then all placed at right angles to the beams, and only shrinkage reinforcement is placed in the other direction Two-way slab construction is economical only for panels that are approximately square, with spans not exceeding 16 or 18 ft

One-way slab construction is suited to rectangular panels if the shorter span is not greater than 10 or 12 ft.

Flat-slab Floors. A flat-slab floor consists of a reinforced-concrete floor slab, built monolithically with the columns and supported directly by the columns without the aid of beams and girders. The slab may be of uniform thickness throughout the entire floor area; or part of it, symmetrical about the column, may be made somewhat thicker than the rest of the slab, the thickened portion of the slab thus formed constituting what is known as a dropped panel or "drop" (see Fig. 3-29). Dropped panels are usually square, the width being approximately equal to one-third (Joint Committee Report specifies 0.35) of the longer dimension of the panel. The thickness of the drop (below the slab) varies in practice from $0.25t_2$ to $0.50t_2$, in which t_2 is the thickness of the slab outside the drop.

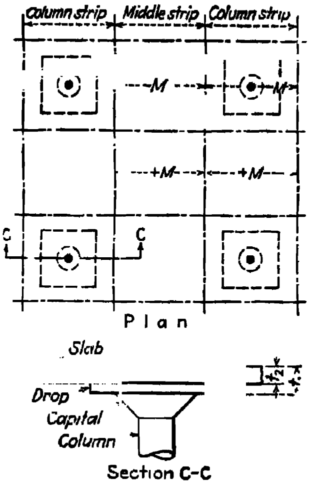


FIG. 3-29. Flat-slab floor.

The columns themselves in practically all cases flare out toward the top, forming a *capital* of a shape somewhat similar to an inverted truncated cone. The capital gives a wider support for the floor slab, reduces the shearing stresses in the slab, and tends to make a more rigid structure. In practice, the effective diameter c of the capital is usually equal to $0.225l$, in which l is the average span of the panel.

Flat-slab construction is the ideal type for industrial buildings in which the live load is 100 lb or more per sq ft and where the spans are 18 ft or greater. The panels should be as nearly square as possible. If rectangular panels are necessary, the longer span should not be greater than 1.33 times the shorter and preferably not greater than 1.1 times the shorter.

The two most common methods of reinforcing flat-slab floors are (1) the two-way system and (2) the four-way system. In the *two-way system* small bars are placed parallel to the lines of columns over the entire area of the floor at short intervals. The maximum spacing allowable varies in the different codes and specifications but is seldom greater than 1½ times the thickness of the slab, or 12 in. The *four-way system* consists of two main bands of steel running parallel to the lines of columns, each band centered about the column lines, and two diagonal bands of sufficient width to fill up the floor area left uncovered by the direct bands.

Ribbed Floors. For apartments, hotels, schools, hospitals, etc., where the live loads are relatively light and the spans relatively long, ribbed-concrete floor systems are usually the most economical. This type of construction consists of a series of small T beams spaced fairly closely, framing into beams or girders which in turn frame into the supporting columns. The T beams are formed by the use of fillers, which may be terra-cotta tiles, gypsum tiles, steel cores, or wood forms. When terra-cotta or gypsum tiles are used, the tile blocks are left in place, giving a flat ceiling which may be plastered underneath. Steel cores are either permanent (left in place) or removable; the latter type permits the reuse of the cores. If a flat ceiling is desired, metal lath is placed in contact with or hung below the bottom of the ribs or T beams when steel cores are used, and the lath is plastered.

The ribs are usually 4 or 5 in. wide, but for unusual loads they may be made an inch or two wider; ribs directly under partitions or concentrated loads are generally made from 1½ to 2 times the normal width, with a proportionate increase in reinforcing steel. As it is not easy to place stirrups in these narrow ribs or beams, the shear resistance of the rib will often govern the depth required. The cover slab forming the flange of the T is generally 2, 2½, or 3 in. in thickness. The floor construction is designated by the depth of the tile pieces plus the cover-slab thickness, *i.e.*, 6 + 2, 10 + 3, etc.

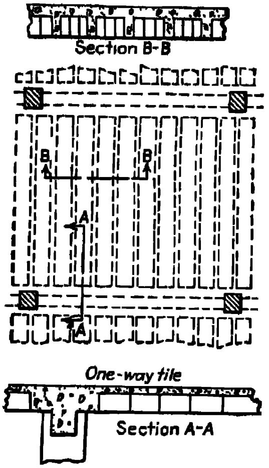
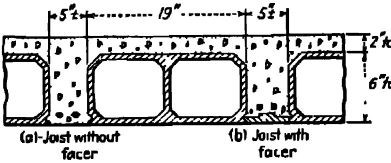


Fig. 3-30. One-way tile ribbed floor.

width equal to the distance center to center of the ribs.

Terra-cotta Tile Blocks. Terra-cotta tile blocks are usually 12 in. square and can be obtained in thicknesses varying from 4 to 12 in. The blocks are placed flatwise and end to end, in rows between, and at right angles to the girder forms, as shown in Fig. 3-30. They are supported either by a solid decking similar to that used for regular concrete slabs or on 2- by 8-in. planks called "soffit boards," which are centered under the space between the rows of tile. The decking or soffit boards are supported on light shoring from the floor below. The usual clear distance between rows is 4 in., so that the distance center to center of the rows is 16 in.

Gypsum-tile Fillers. Gypsum-tile fillers are made of dense, specially prepared gypsum, in two sizes, one 12 in. wide and 30 in. long, with thicknesses of 3, 4, or 5 in., and the other 19 in. wide and 18 in. long, with thicknesses of 6, 8, 10, or 12 in. The 19- by 18-in. fillers are cast with one integral end, which provides a seal for each row at the girders. The 12-in. tiles are usually placed 16 in. on centers (with 4-in. rib), while the 19-in. tiles (Fig. 3-31) are usually placed 24 in. on centers (with 5-in. rib). Rib widths can, however, be varied to suit any conditions. Open centering, consisting only of planks or soffit boards for the bottom forms of the ribs, is usually used.



(a)-Joist without facer (b) Joist with facer

Fig. 3-31. Ribbed floor with gypsum tile fillers.

When *special steel cores* are used, the formwork again consists of open centering, in which planks or soffit boards are placed only under the concrete ribs or joists. The steel cores are available usually in lengths of 30, 35, and 36 in. and in depths of 4, 6, 8, 10, 12, and 14 in. They are tapered in cross section, as shown in Fig 3-32; the most common width at the bottom is 20 in., although other widths between 10 and 31 in. are also obtainable. With permanent cores, if a plastered ceiling is desired, metal lath *f* (Fig. 3-32) is placed under the cores before they are put in position and anchored to the concrete ribs by suitable metal clips which project up into the ribs. The entire ceiling is plastered after the formwork is removed. With removable cores, the ceiling may be plastered by fastening metal lath to the concrete after the cores

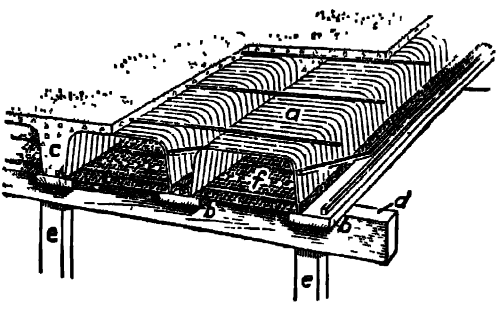


Fig. 3-32. Ribbed floor with steel cores.

are removed, or the ribs may be left exposed. The ribs are usually made 5 in. wide at the bottom, so that the distance center to center of ribs is ordinarily 25 in.

The girders which support the ribs are rectangular beams or T beams with a maximum flange thickness equal to the total floor thickness, as shown in Fig. 3-33. If the maximum flange thickness is to be used, the rows of tiles are stopped at the proper distance from the stem of the girder. If a flange thickness less than the maximum is desired, the end pieces of tile in each row are short sections which are dropped down on the forms so that the tops of the pieces are at the proper elevation to furnish the required slab thickness above them. End caps are used at each end of each row of tile to hold the girder concrete in position.

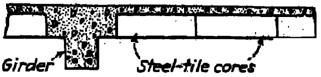


Fig. 3-33.

Steel-joist and Precast-concrete-joist Floors. A steel-joist floor consists of a series of closely spaced, parallel, shallow joists or trusses of the Pratt, Warren, or double-Warren type, supported at the ends on steel or concrete beams, or on masonry walls, and covered with a thin concrete slab. Two common types of steel joists are shown in Fig. 3-34. Ordinary steel joists are manufactured in depths of 8, 10, 12, 14, and 16 in. and are carried in stock in a range of depths and lengths to meet all usual conditions of load and span. The span limits vary from 4 to 32 ft. The joists are spaced from 12 to 30 in. on centers, depending on the load and span, and are covered with a 2- or 3-in. concrete slab poured on metal rib lath which rests on and is fastened to the upper surfaces of the joists. The joists are braced transversely by some form of rigid diagonal bridging placed between adjacent trusses, at intervals of from 5 to 7 ft. One type of bridging is shown in position in Fig. 3-35. The bridging members are light ¾-in. channels which are notched at the ends to rest against and be clipped to the chords of the trusses.

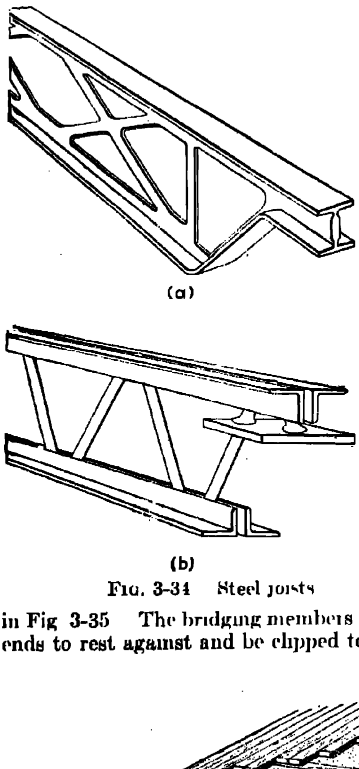


Fig. 3-34. Steel joists.

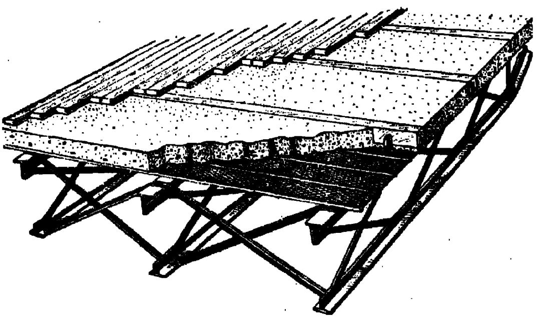


Fig. 3-35. Steel-joist construction.

Special long-span steel joists are also available, primarily for roofs or very light floors, with depths up to 32 in. for spans up to 64 ft. Roofs supported by these long-span joists have wood-plank, gypsum-plank, or precast-concrete-plank decks with built-up roofing or poured concrete decks. The spacing of the joists is limited by the safe span of the floor slab or roof deck. Long-span joists are bridged with cross bracing of not less than 1¼- by 1¼- by ⅛-in. angles, bolted together at the intersection and spaced not more than 10 ft apart.

Steel-joist floors are economical for light occupancies, but they are unsuitable for heavy or vibrating loads. They are easily installed, and the cost of construction is low because no forms are required for the concrete slab. Ceiling lath for plaster is attached directly to the lower chords of the trusses. Since the webs of the joists are open, all plumbing and service pipes may be concealed in the construction without interfering with the structural members. Joists are selected from the manufacturers' catalogues, which give the total load which may be supported safely by joists of various depths, spans, and spacings.

Precast-concrete joists may be used in much the same way as steel joists. Concrete joists are made of lightweight aggregates in depths varying from 6 to 14 in. and in lengths up to 36 ft. They are of I cross section, reinforced top and bottom. A 2- or 2½-in. slab is poured over the joists after they are set in position, as shown in Fig. 3-36. Concrete joists are selected from the manufacturers' catalogues.

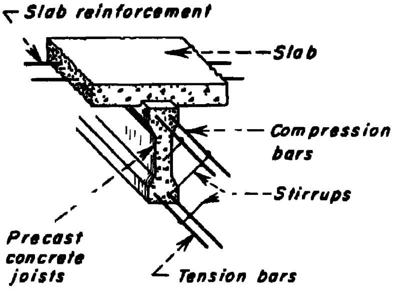
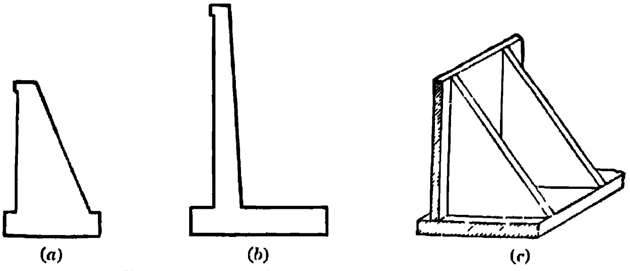


FIG. 3-36. Precast-concrete-joist construction.

RETAINING WALLS

Types. Concrete retaining walls are of three classes: (1) gravity (Fig. 3-37*a*), (2) cantilever (Fig. 3-37*b*), and (3) counterfort (Fig 3-37*c*). The latter two are reinforced. The gravity type is commonly used for low walls, the cantilever type for walls of medium height, and the counterfort type for the higher walls No marked line of separation exists, in general, counterfort construction is not economical for walls less than 18 ft in height.



(*a*) (*b*) (*c*)

FIG. 3-37. (*a*) Gravity wall. (*b*) Cantilever wall. (*c*) Counterfort wall.

The width of the base of the wall will, in general, average about 0.45 of the total height of the equivalent earth column at the heel of the wall, measured from the base of the wall. Greater economy is obtained in the reinforced walls by placing the vertical arm at some distance back of the toe

Conditions of Loading. Four general cases are possible: (1) walls with no surcharge, the top surface of the fill being horizontal and level with the top of the wall; (2) walls with an inclined surcharge, the top surface of the fill extending upward and back from the top of the wall (the angle of inclination of the sloping surcharge is usually taken as the angle of repose of the material in the fill); (3) walls with a horizontal surcharge extending some distance above the top of the wall, and (4) loadings in which the actual surface of earth does not extend above the top of the wall but supports an external load such as a building, railroad tracks, etc. Such loads are converted into an equivalent height of earth above the top of the wall by dividing the weight of the additional load per square foot by the weight of the earth per cubic foot. The pressure against the wall is then computed for this equivalent height of earth surcharge, as in case 3.

Formulas for Earth Thrust. Rankine's development of the theory of earth pressure leads to the general equation,

$$P = C \times (wh_1^2/2)$$

where P is the total thrust per linear foot of wall upon the back of a retaining wall, w the weight of the earth per unit volume, h_1 the height of the earth column, and C a constant depending upon the angle of inclination of the back of the wall, the conditions of loading, and the physical properties of the earth fill. Equations for P are given in Fig. 3-38. In the equations, h is the height of the wall in feet, h' the height of surcharge in feet, and ϕ the angle of repose of the retained fill. The location of the line of action of P is also given in Fig. 3-38 for each of the cases shown. The value of ϕ is usually taken as 33°42′ (slope $1\frac{1}{2}$ to 1) for ordinary earth fills. The amount of pressure on any given horizontal strip 1 ft in height at a distance x ft below the surface of the earth is

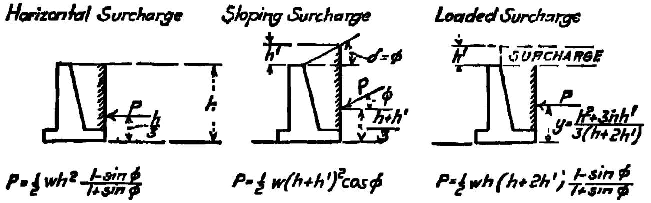


FIG. 3-38. Earth pressures on retaining walls.

$$P_1 = Cwx$$

Overturning and Crushing. In order to prevent overturning of the wall, the resisting moment of the weight of the wall and the earth above the base of the wall about the toe must be greater than the overturning moment of the earth pressure about the same point. A factor of safety of 2 is generally required, *i.e.*, in Fig. 3-39, $W_1x_1 + W_2x_2 + W_3x_3$ must be greater than $2Py$.

In order to prevent crushing of the soil at the toe of the wall, the soil pressure at that point should not be greater than the allowable pressure for the soil under the wall. Formulas for the toe and heel pressures are given in Fig. 3-40,

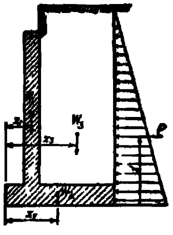


FIG. 3-39.

where E = resultant of the total earth pressure and the resisting weight (concrete and earth above base of wall) on a 1-ft strip of wall, lb

F = vertical component of E

l = length of base, ft

a = distance, ft, from toe to intersection of E with the base

Figure 3-40 illustrates the desirability of having the resultant pressure E intersect the base of the wall within the middle third in order that the base will bear on the soil for its full length.

Sliding. In order to prevent sliding, the frictional resistance of the base against the foundation material must be greater than the horizontal component of the earth pressure P. A safety factor of $1\frac{1}{2}$ is usually specified, *i.e.*, $f(\Sigma W)$ must be greater than $1\frac{1}{2}P_H$,

where f = coefficient of friction, concrete on earth

ΣW = total vertical load on base of wall = F in Fig. 3-40

P_H = horizontal component of earth pressure P

Values of f may be assumed as follows: on dry clay, 0.5; on wet clay, 0.33; on sand, 0.4; on gravel, 0.6. In case an adverse condition of sliding exists, the base may be widened, thus increasing the weight of the wall; narrow shallow trenches may be dug in the foundation, forming projections which will materially increase the resistance to sliding; or the base may be inclined upward toward the toe.

Design of Gravity Walls. Assume a cross section (see Fig. 3-37a), and investigate for sliding, overturning, and crushing of the soil as previously outlined, basing all computations on a 1-ft length of wall. If any of these factors is unsatisfactory, revise the cross section and repeat.

Design of Cantilever Walls. Assume a cross section (see Fig. 3-37b), and investigate for sliding, overturning, and crushing of the soil as for a gravity wall. The vertical slab is then designed as a cantilever (using a wall length of 1 ft) with a horizontal load equal to the horizontal component of the total earth pressure on the vertical slab. The horizontal slab is designed as a cantilever which supports an upward load equal to the reaction from the soil under the cantilever and a downward load equal to the weight of the slab and earth above it.

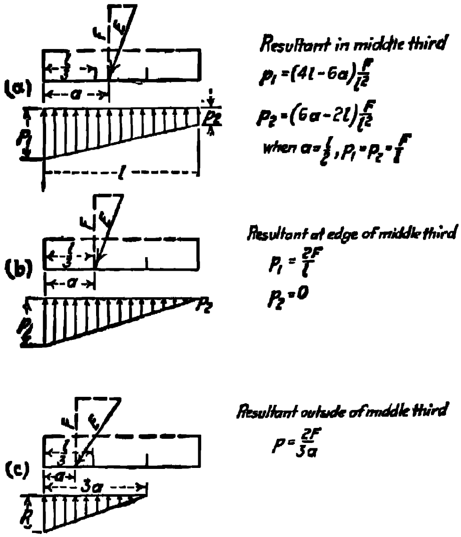

(a)
a
l
p_1
p_2
Resultant in middle third
$p_1 = (4l - 6a)\frac{F}{l^2}$
$p_2 = (6a - 2l)\frac{F}{l^2}$
when $a = \frac{l}{2}$, $p_1 = p_2 = \frac{F}{l}$
(b)
a
p_1
p_2
Resultant at edge of middle third
$p_1 = \frac{2F}{l}$
$p_2 = 0$
(c)
a
3a
p
Resultant outside of middle third
$p = \frac{2F}{3a}$

FIG. 3-40. Foundation pressures.

Design of Counterfort Walls. Assume a cross section (see Fig. 3-37c), and investigate for sliding, overturning, and crushing, basing all computations on a length of wall equal to the distance center to center of counterforts. A spacing of about ten feet for the counterforts for walls of medium height will usually prove economical. The thickness of counterforts varies from 12 to 18 in. The face wall is built with a vertical back; the front might be battered slightly for appearance. This wall is designed as a simple or partly continuous slab, supported by the counterforts with a horizontal load equal to the horizontal component of the earth pressure. The base slab is designed as a simple slab supported by the counterforts; sometimes as a partly or fully continuous slab. The steel is placed parallel to the wall and may be in the top or in the bottom of the slab, depending upon whether the resultant moment causes tension in the upper or lower portions of the slab. The resultant moment is the algebraic sum of the moments caused by the upward earth pressure and the downward load of the filling.

The counterforts are designed as wedge-shaped, vertical cantilever beams supporting a horizontal load equal to the total horizontal component of the earth pressure on the face wall between two counterforts. The effective depth of the counterfort is the perpendicular distance to the steel near the rear edge of the counterfort from the point of intersection of the counterfort, the front face of the vertical wall, and the upper face of the base slab. The counterforts must be tied to the vertical wall and to the base slab with horizontal and vertical bars, respectively. The main steel in the counterforts is placed along the back or sloping edge.

Details of Construction. The front face of a retaining wall is usually built with a batter of from ½ to 1 in. in 12 in. A coping, projecting a short distance beyond the wall, adds to the architectural appearance and, to a certain extent, protects the masonry in the body of the wall from dripping water. A top width of less than 10 in. is generally undesirable; heavy gravity walls should preferably have a top width not less than 18 to 24 in. The base of the foundation should be a sufficient distance below the surface of the ground to prevent damage by frost, a minimum of about 3½ ft being satisfactory in ordinary climates. Expansion joints should be provided at intervals along the wall, preferably not farther apart than 30 ft. In the reinforced walls where cracks would not only be unsightly but also detrimental to the integrity of the wall, additional steel should be placed at right angles to the main reinforcement to provide for temperature and shrinkage stresses An amount varying from 0.10 to 0.33 per cent of the cross section is usually specified. Proper drainage of the fill behind the wall may be effected by inserting 4-in. drain tiles through the wall near the bottom, at intervals of 10 to 15 ft, and piling crushed stone, gravel, or other coarse material around the holes. At least one drain should be provided for each pocket formed by counterforts.

SECTION 4

MATERIALS OF CONSTRUCTION

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BY RUSSELL D. ROGERS

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MODERN MATERIALS OF CONSTRUCTION

BY RUSSELL D. ROGERS, B.S.

Senior Architectural and Civil Engineer, E. I. du Pont de Nemours & Co.

Numerous excellent texts are available on analytical design of structures, strength of materials, and methods of erection of simple and composite structures of wood, metal, and masonry. The plant engineer can find volumes of these data in any public library, but he will have considerable difficulty in locating a text that will assist him in his maintenance and repairs.

Usually the plant engineer inherits his job after the plant has been designed and erected, and it is his duty to keep the plant operating with a minimum of interference with operation and with a minimum of expense.

This section will therefore consist of information relative to proper selection of materials for construction and repair with a description of the advantages and disadvantages of these materials for certain uses. For useful coverage, the discussion must be limited to floors, walls, and roofing.

FLOORS

Probably 90 per cent of all recently constructed large industrial plants have concrete floors, and the remaining 10 per cent have wooden floors

The types of finishes for concrete floors, each of which is suitable for a specific use, are listed below and discussed in the paragraphs following.

1. Cement
2. Mastic
3. Wood block
4. Wood, laminated
5. Brick
6. Rubber
7. Plastic, resin type
8. Linoleum
9. Mastic tile
10. Mastic block
11. Cork
12. Magnesite

Cement Finish. A cement finish should not be used when the surface is continuously subjected to extremely high temperatures, *i.e.*, above 250°F. The effect of heat on the strength of concrete, as reported by Portland Cement Association's *Bulletin* ST32, is shown in Table 4-1.

Table 4-1

Temp., deg F	*Reduction in compressive strength, per cent*
200	18
500	40
800	52
1100	85
1400	96
1700	97

While the above statistics apply specifically to monolithic concrete, the effects are relative in so far as the cement finish is concerned. The finish will disintegrate owing to a combination of temperature change and impact from floor loads.

Where heat resistance is necessary, the use of a special cement such as Atlas Lumnite is recommended.

Cement finishes are susceptible to attack from

Coal-tar distillates—phenol, cresol, lysol, creosote, carbolinium

Acids—acetic, nitric, hydrochloric, hydrofluoric, lactic, carbonic, carbolic, humic, phosphoric, sulfuric, sulfurous, and tannic

Salts and alkalies—carbonates of ammonium, potassium, and sodium

Chlorides —calcium, potassium, sodium, strontium —especially when in presence of moisture, also chlorides of ammonia, copper, iron, magnesium, mercury, zinc

Hydroxides— Ammonia, potassium, sodium

Nitrates—Ammonia, potassium, sodium

Sulfates—Practically all

Vegetable oils—Almond, castor, cocoa nut, olive, peanut, poppy seed, rape seed, and walnut

Additional valuable information on this subject may be secured from Portland Cement Association's *Bulletin* ST4.

Preparation of a proper base is necessary to maintain a proper finish. The following precautions will assist in obtaining maximum service with minimum maintenance costs. A sound coarse and fine aggregate meeting ASTM Specification C33 should be used.

Mix. A laboratory-designed mix to obtain a 28-day compression strength of *at least* 2,500 psi should be prepared An arbitrary but safe mix would be 1 part portland cement; 2½ parts fine aggregate, well graded from ¼ in. down; and 3½ parts coarse aggregate, well graded from 1½ down to ¼ in. Not more than 6 gal of water should be used per bag of cement.

The base should be poured in alternate areas bounded by the column center lines, preferably not exceeding 1,000-sq ft areas. Tests and experience have indicated that the initial shrinkage of concrete is greater than any subsequent expansion due to either moisture or temperature changes. By pouring in alternate areas, this initial shrinkage will have taken place before the adjacent area is poured.

If a separate finish is used, it is the general opinion of experienced contractors and engineers that it is preferable in the average industrial plant to permit the finish to crack on a straight line directly over the construction joint rather than deliberately to construct a tooled groove. If a scored finish is desired and not objectionable to traffic and operations, it should always be placed on the center line of columns or at the corner of large foundations projecting above the floor.

Where the unit pressure on the floor is greater than that on the footings, especially in heavily loaded one-story structures, an expansion joint should be placed at all walls and around all column and machinery foundations to permit differential settlement The joint should be free to move vertically without resting on its edge, thus causing "dishing." A typical expansion joint is shown in Fig. 4-1.

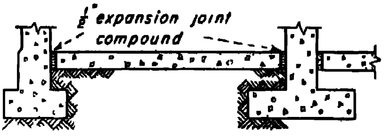

Fig 4-1. Typical expansion joint for floating floor.

The top surface should be screeded to the proper elevation and roughened if a separate topping is to be provided The topping should consist of a mix of 1 part portland cement, 1 part fine aggregate, and 1¾ parts coarse aggregate. The fine aggregate should range from No. 100 mesh to No. 4 sieve and the coarse aggregate from ⅛ to ½ in.

For severe service a metallic aggregate of from 50 to 150 lb per square, *i.e.*, 0.5 to 1.5 lb per sq ft, may be used. This type of topping and all heavy-duty toppings require experience and skill beyond the average for proper installation. Specialists such as the Kalman Floor Co. or Master Builders Co. should be consulted for assistance.

Patching. In spite of advertising to the contrary by manufacturers of patching compounds, no patch should be "featheredged." The sides should be chipped to secure a vertical surface and thoroughly soaked until they cannot absorb further moisture Then a grout of pure cement and water of the consistency of paint should be broomed into the area. While still damp, the patching material should be applied and thoroughly tamped into place. This material should consist of a mix as nearly like the original material as possible. The mix should be so dry that it will just retain its shape when squeezed in the hand. It should then be troweled to match the adjacent surface.

As an additional precaution against shrinkage, pulverized pure grey iron such as Master Builders Embeco may be added in the proportion of 25 lb of iron to each bag of cement. An excellent proprietary patching compound identified as Hydroment, manufactured by Upco Co., of Cleveland, Ohio, is also available for emergency use.

Where a wearing surface is to be added for trucking aisles, the method shown in Fig. 4-2 should be used.

Proprietary compounds are available to assist in securing a good bond between new and old concrete surfaces. Examples of such compounds are Bondsit, made by the A. C. Horn Co., Hydroment, by The Upco Co., or Bond-tite, by the American Fluresit Co. A prime coat of cement and iron with an oxidizing agent can also be secured from L. Sonneborn Sons, Master Builders, A. C. Horn, and others.

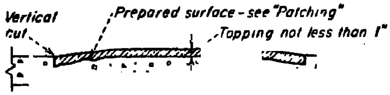

FIG. 4-2. Trucking-aisle patch to eliminate "featheredge."

Mastic Topping. Mastic topping consists of the best grade of asphalt, especially Trinidad Lake natural asphalt, silica filler, silica sand and pebbles, carbon, and other similar fillers.

This topping is satisfactory where it is free from heavy or concentrated traffic, especially heavy wheel loads. It should not be used where constant temperatures exceed 100°F or in the presence of oils unless special oil-resistant mastic (similar to Ralph V. Rulon Co. mastic) is used.

If properly installed, mastic topping is excellent protection in extremely wet and wet-acid areas, is odorless, dustless, slipproof, and easily repaired. It is also satisfactory as a tank or pit lining.

If installed by a specialist in this line of work, no waterproof membrane is required for protection against acid leaks and their resultant attack on the concrete base. If membrane is used, the use of Fiberglas (Owens-Corning) cloth instead of cotton fabric is recommended. Graded carbon is added where resistance to hydrofluoric or hydrofluosilic acid is required.

A method proved satisfactory is to prime the concrete surface, apply approximately 1/8 in. of asphalt matrix consisting of a pure grade of low-melting-point asphalt similar to the Trinidad Lake matrix of the Ralph V. Rulon Co., Atlas Mineral Products Co. No. 31, or Johns-Manville No. 60 flux. Over this a 5/8-in layer of acid-resisting mastic is applied. The aggregates used should be free from limestone or materials that are easily attacked by acid. The material should be maintained at a temperature of between 350 and 450°F and combined to produce a plastic mixture. A second layer of 5/8 in. thickness should then be applied with all joints broken. Hot mastic should overlap the cold about four inches to make a soft edge. The top layer should be proportioned for its intended loading. It should be remembered that the softer and more elastic material is more acid-resisting but also unsatisfactory for either static or moving loads.

The finished surface should be given a sand- or silica-dust sprinkle while hot and rubbed to a smooth finish with wood floats.

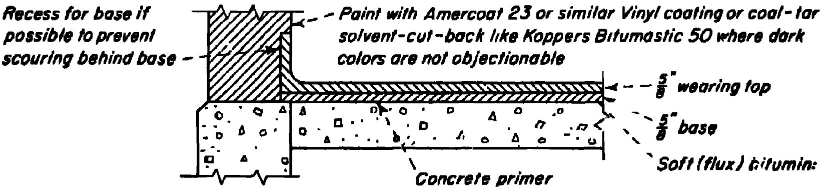
Recess for base if possible to prevent scouring behind base
Paint with Amercoat 23 or similar Vinyl coating or coal-tar solvent-cut-back like Koppers Bitumastic 50 where dark colors are not objectionable
5/8" wearing top
5/8" base
Soft (flux) bitumin:
Concrete primer

FIG. 4-3. Acid-proof mastic floor.

Fillets and coves are as necessary as the floor and should be used throughout to prevent the accumulation of salts and dirt in sharp corners (see Fig. 4-3).

The maintenance on this type of floor is very low if it is installed by a reputable, experienced contractor. It can also be extremely high if so poorly constructed that frequent repairs are necessary to floor and damaged equipment. Premiums paid for expert application in this work will result in a long-time economy.

Wood Block. Wood block is available in treated and untreated end-grain blocks. The advantages of this type of floor finish are that it is comfortable for footwork; dustless; has a low noise factor; is odorless if necessary; cushions the fall of tools, castings, etc., in machine-shop service; insulates against ground coldness; is repellent to insect attack; sparkproof; skidproof; has a low maintenance cost; does not sliver; is easy to repair; and has excellent resistance to mechanical abuse. It is also medium-priced for its service.

Its disadvantages are that it is dark-colored except in redwood and untreated block and cannot withstand frequent wetting without a drying period, although the treated type is immune to wet rot. Also careful workmanship is required to secure true and level surfaces, or the surface should be sanded and refilled.

Typical Available Types. 1. Kreolite (Jennison-Wright Co., Toledo, Ohio), single end-grain yellow-pine lug blocks, 2 to 3 in. in thickness, 3 to 4 in. in width, 4½ to 8 in. in length. This block is treated and provided with lugs to secure proper spacing. It is installed on a concrete base with a coat of primer, ⅛ in. bitumen. The joints are filled with pitch. The surface may be buffed or varnished after a top coating of coal-tar-pitch emulsion. It is also available in flexible strips with metal splines, 2 to 8 ft long to secure a true and level surface. Dark or light blocks are available. It may be waxed.

2. Worthwood (Worth Lumber Co., Seattle, Washington), end-grain Douglas fir, with the wood splined in lengths of 2 to 8 ft, 1⅜ by 3½ in. minimum, 1⅞ by 4½ in. maximum face, thickness of 2 or 2½ in., width of 3½ to 4½ in. It has a grooved base, shaved edges, and sanded face to secure uniform thickness. There are two grades, select and clear. The strips are set in mastic over a concrete primer. Subfloors on ground or basement should have a three-ply waterproof membrane under the blocks.

3. The Koppers Co., Century Wood Preserving Division, has a wood block which is the same as the Jennison-Wright without lugs. It can be secured with V-cut faces to assist the placing of bitumen.

4. Palco Redwood (Pacific Lumber Co., Chicago, Illinois) consists of 3⅝- by 5⅝-in. redwood vertical-grain blocks, 2, 2½, 3, and 4 in. in thickness. There are also fancy finished floors for parquetry, herringbone, etc., in patterns 2 in. thick. The weight of 2 in is 4.3 lb. It has a compressive strength of 1,000 psi. It is set on a concrete subbase with special mastic.

Precautions. In truckways and wherever possible, the loose blocks should be set with their length at right angles to the line of traffic. The joints should be staggered, and expansion joints should be provided around the room and adjacent to columns and other vertical obstructions. Machinery should not be bolted to blocks. A three-ply membrane should be used over subfloor when at or below grade.

Pipes extending through floors should be encased in sleeves firmly set in subfloor. Concrete curbs should be provided at all openings, stairs, elevators, edges of platforms, etc., to act as stopping points for blocks. Ramps should be provided with curbs at both top and bottom.

Wood Floors- -Industrial-laminated Type. The advantages of laminated floors are that they are resilient; easily maintained; easily obtainable; flexible for moving machinery; economical; long lived owing to freedom from termites and dry rot or dampness; will stand medium to heavy loads with good foundation; and expert skill is not required for installation.

The disadvantages are that multiple joints catch dirt; they are hard to keep clean if the wood is not properly dried or oil and dust or grit are present in operation; and they are not suited to extremely wet operation continuously.

Advantageous use of this type of construction naturally cannot be obtained without proper construction. A type of construction proved satisfactory and which should be used as a basic type for any necessary variation is the Tar-Rok construction as recommended by the Barrett Division of the Allied Chemical & Dye Corp., illustrated in Figs. 4-4 and 4-5.

The construction of subfloors over earth is shown in Figs. 4-4 and 4-5. The foundation course consists of 4 in. of dry crushed stone or screened gravel thoroughly mixed with No. 5 Barrett tar. The proper amount to use is

6 gal per cu yd of 1- to 2½-in. crushed stone
9 gal per cu yd of ¼- to 2½-in. crushed stone

7 gal per cu yd of coarse screened gravel
10 gal per cu yd of fine screened gravel

The tar should be heated to not more than 200°F. After it is uniformly spread, the foundation courses should be rolled and compacted with at least a 300-lb roller (per foot of length). Over this foundation course the final dampproof course should be installed.

The dampproof course is made by thoroughly mixing in a suitable mechanical mixer Barrett tar No. 7 with clean, dry sand which has been preheated. Thirty gallons of tar are required per cubic yard of sand. The temperature of the tar should not exceed 225°F. If a thick, white smoke, indicating overheating, rises from the mix, 5 gal of tar should be added for each cubic yard of sand. The dampproof course should be spread evenly to a depth of 1¼ to 1½ in. (so it will compact to 1 in.) over the concrete, leveled with a straightedge, and followed closely with the plank, which is set and hammered level in the soft mix and firmly toenailed in position.

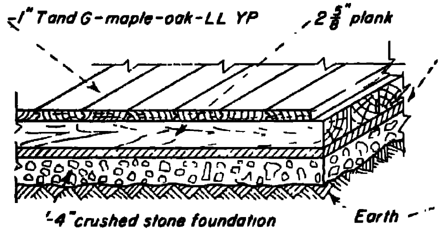

FIG. 4-4.

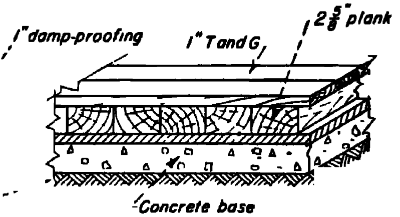

FIG. 4-5. Industrial-laminated-type wood floor.

For intermittent wet use, or for local areas subjected to wet processes, it is most advisable to use lumber that is treated with 1 lb of chromated zinc chloride salts per cubic foot of wood or at least 8 lb of pressure-treated creosote in accordance with the American Wood Preserving Association specification.

Brick. Brick floors are so seldom used for other than acidproof areas that their installation will not be described; and all details and discussions will be confined to acidproof floors.

Brick floors are immune to attack of acids or alkalies; are capable of withstanding high temperatures and extremely heavy static or moving loads; can be made attractive and sanitary; are easily incorporated with construction of equipment foundations, curbs, sumps, trenches, etc.; are easily patched; and have low maintenance cost if properly constructed.

The disadvantage of brick floors is their high first cost. The bricks used should be a dense deaired acidproof shale, having an absorption of approximately 1 per cent or less, equal in quality to Claycraft, Hanley, Belden, Atlas, Knight, Duro, Summitville, General Refractories, etc. The most commonly used sizes are (nominal sizes) 1⅛ by 4 by 8 in.; 1⅜ by 4 by 8 in.; 2¼ by 4 by 8 in.; 1½ by 6 by 12 in.; 3¾ by 4½ by 8 in.; 2½ by 4½ by 9½ in ; 2½ by 4 by 8½ in. Special shapes, such as bullnose, internal and external corners, cove bases, jamb tiles, etc , are generally available, especially for the 1⅜- by 4- by 8-in. size, which is an excellent general-utility size for all but extremely heavy moving loads. Abrasive-surfaced or nonskid-type bricks should be used in wet areas.

Acidproof Mortars. There are many reliable sources of supply for a variety of mortars, each of which is designed for a specific purpose. It is evident that this discussion can indicate only a very small portion of the technical details and precautions essential for proper installation.

If a considerable amount of this type of construction is necessary, more economy will be realized if the engineer will present his entire problem to one of the many reliable manufacturers and profit by their extensive field and laboratory tests on similar problems.

An abbreviated list of products and their manufacturers is provided for guidance. It is not intended to be all-inclusive, as each manufacturer has available special modifications of these cements for specific uses.

The type of material used, width of joint, moisture, temperature, mixing, and placing are all extremely important for proper workmanship. If work of this type is contracted, it should be kept in mind that the lowest bidder may not necessarily be the most economical, unless he has had considerable experience in this type of work.

There are three major types of hard-setting cements available, sulphur, resin, and silicate. The proper joint thickness and method of mixing are given in Table 4-2.

Table 4-2

Sulfur type	Resin type	Sodium silicate type
1. Tegul Vitrobond, Tegul Tileset	8. Korez	20. Penchlor
2. Carbo Vitrobond	9. Carbo Korez	21. Vitrex
3. Basolit	10. Alkor	22. Sauereisen No. 31
4. Carbon Basolit	11. Durisite	23. Glaseal
5. Plasul Basolit	12. Alk-Li-Pruf	24. Duro
6. Sauereisen No. 48	13. Pennsalt PRF	25. Quigley Black
7. Weldtite	14. Pennsalt HF and HFK	26. Knight No. 2
	15. Causplit	
	16. Permanite	
	17. Asplit	
	18. Nukem-ALKA	
	19. Nukem-Carbon Resinous	

Approximate average characteristics

	Sulfur type	Resin type	Sodium silicate type
Weight, standard types, lb/cu ft	137	120	120
Carbon types, lb/cu ft	137	90	
Tensile strength, psi	650–900	650–1 500	375
Compressive strength, psi	6,200–8 000	10 000–12,000	2,800
Absorption, per cent	1–2	1–2	10–15
Setting time @ 70°F	5 min	1 to 6 hr	25 min to 1 hr
Elasticity, in deflection, bar 24 × 1 × ½ in thick	0.20–0.25	1.3	0.17
Bond strength psi	500	600	185
Max service temp, deg F	200°F	350°F	1600–2000°F
Coef of expansion	8.3×10^{-6}*	6.5–14×10^{-6}	6.2×10^{-6}
Ready for service	5 min	16–24 hr	2–4 days
Joint thickness	¼–½ in max	1/16 to ⅛ in.	(20) ¼–⅛ in. (21) ¼–⅜ in. (22) 1/16–5/16 in. (23) ⅛–⅛ in.
Application	Melting and pouring at 280°–300°F	Mix liquid and powder to form plastic mix per mfr's directions, apply with trowel	Mix and apply with trowel

* Tegul Tileset = 5×10^{-6}.

NOTE. The values given are averages for the group. Actual values for the specific material should be secured from the manufacturer before installation of work. The width of joints must obviously be increased to at least ¼ in. for repointing.

It is important to note that sulphur cements should not be used where operating temperatures exceed 200°F, resin cements at 350°F, and silicate cements at 2000°F.

In all cases, bricks and concrete base must be dry; the temperature of the brick should be between 65 and 80°F. Resin and silicate are extremely sensitive to changes in temperature.

Resins and silicates must not come in contact with portland cement or any material having an alkali content. The surfaces must be neutralized with dilute hydrochloric acid or the surface primed with an acid-resisting bitumen approved by the manufacturer. An acid-soaked surface should be neutralized with a 5 per cent caustic soda solution, water-washed and dried.

All sodium silicate joints should be washed down with a 5 to 10 per cent solution of hydrochloric acid before use. In a food industry, the hardened joint should be treated with a 10 per cent solution of calcium chloride and then washed with water.

Resin and sulfur cements do not require any acid wash or curing. They are ready for service when hard.

Since the jointing material comprises a major portion of the cost of the floor, it is essential to select the lowest priced cement that will fulfill the requirements. Silicate mortar and sulfur-base mortars can be purchased for approximately one-fourth the cost of a resin mortar. However, the resin and silicate mortars are mixed cold, and the elimination of the heating process with its inconvenience reduces the labor cost for these mortars to a competitive basis.

When using either resin or sulfur mortars it is necessary to provide good ventilation and goggles. Some resin cements may cause skin irritation to those allergic to the cement, so the material should not be left on the skin for long periods of time. Yellow vaseline may be used as a good protection against skin irritation. Sand soap and plenty of water will remove the cement from the hands. Solvents should never be used on the skin. It is always advisable to consult a physician in case of irritations. Tools should be cleaned with 5 per cent caustic soda.

Resin solutions that have solidified are useless. Resin powder will not deteriorate in dry storage.

Resistance Data. SULFUR—STANDARD TYPES (1, 3, 5, 6, 7, Table 4-2). These cements are resistant to abrasion, mechanical abuse, and sudden changes in temperature. They are inert to such acids as sulfuric, hydrochloric, phosphoric, acetic, lactic, nitric, and corrosive salts of these acids. They are attacked by strong alkalies, chromic acid, hydrofluoric acid, and some organic solvents.

SULFUR—CARBON TYPES (2, 4, Table 4-2). These cements will resist attack of hydrofluoric acid; otherwise they have the same characteristics as the standard types.

RESIN TYPES (8 to 19, inclusive, Table 4-2). There are two types of resin cements in general use—phenol-formaldehyde resin and furane resin.

The phenol-formaldehyde type is subdivided into two groups, one without carbon (8, 15, 17, Table 4-2) and one with carbon (9, 13, 14, 19, Table 4-2). The plain phenol-formaldehyde resin is resistant to most acids, including hydrochloric, phosphoric, sulfurous, acetic, chloracetic, chlorasulfuric, oxalic, lactic, sulfuric up to 50 per cent concentration; neutral salt solutions, water, steam, and oil, to 350°F; also *mild* alkalies, ammonia, benzine, methylene chlorite, chlorobenzine, oleic acid, naphtha, ether alcohol, and acetone. Some of the resins require heating to 300°F to resist alcohol and acetone. They are *not* resistant to nitric acid, hydrofluoric acid, chromic acid, caustic soda, sodium hypochlorite, or for aniline or other organic bases.

The carbon-type phenol-formaldehyde resins have all the characteristics of the plain resin and in addition are resistant to hydrofluoric acid, sulfuric acid of any concentration up to 150°F, sodium carbonate, calcium hydroxide, ammonia, fats, oils, most organic solvents, and corrosive salts except chromates. They are attacked by nitric acid, chromic acid, and sodium hypochlorite.

The furane resins (10, 11, 12, 16, 18, Table 4-2) have more favorable characteristics than the phenol type. They are resistant to strong alkalis or alternate exposure to alkalis and acids. They are also resistant to inorganic acids such as sulfuric, hydrochloric, phosphoric, and hydrofluoric, as well as most solvents, oils, and greases.

The furane resins are *not* resistant to highly oxidizing acids such as nitric and chromic, phosphorous bromides, nitroglycerine, nitrotoluene, iodine, iodoform, and sulfur oxychloride.

While it is advisable to use the furane resins (10, 11, 12, 16, 18, Table 4-2) where it is known that alkalies are continuously present, the sulfur and phenol types will stand up if the solution is alkaline for short periods followed immediately by acid solution.

Sodium Silicate Type (20 to 26, Table 4-2). Silicate cements are resistant to all inorganic acids except hydrofluoric and to almost all organic acids, solvents, oils, paraffine wax, hot and cold water, and steam They are attacked by hydrofluoric acid, alkaline solutions, and acetone oil

The sodium silicate types are *not* recommended for floors owing to their high porosity The salts recrystallize and expand, thus destroying the joints. Neither should they be used in trenches where the liquid level varies sufficiently to permit drying of the liquid with resultant crystallizing

They are much more stable in strong acid solutions and where always saturated with the acid liquid

Installation Details Figures 4-6 to 4-11, inclusive, show typical acidproof floor and trench protection Note particularly the limitation on joint thickness Refer to the schedule for type of material and temperature limitations. Figures 4-8 and 4-9 have more resistance than Fig 4-7, and Fig 4-7 has more than Fig 4-6 The cost is in direct proportion to the quality Figure 4-8 is ideal for use in dairies and food plants

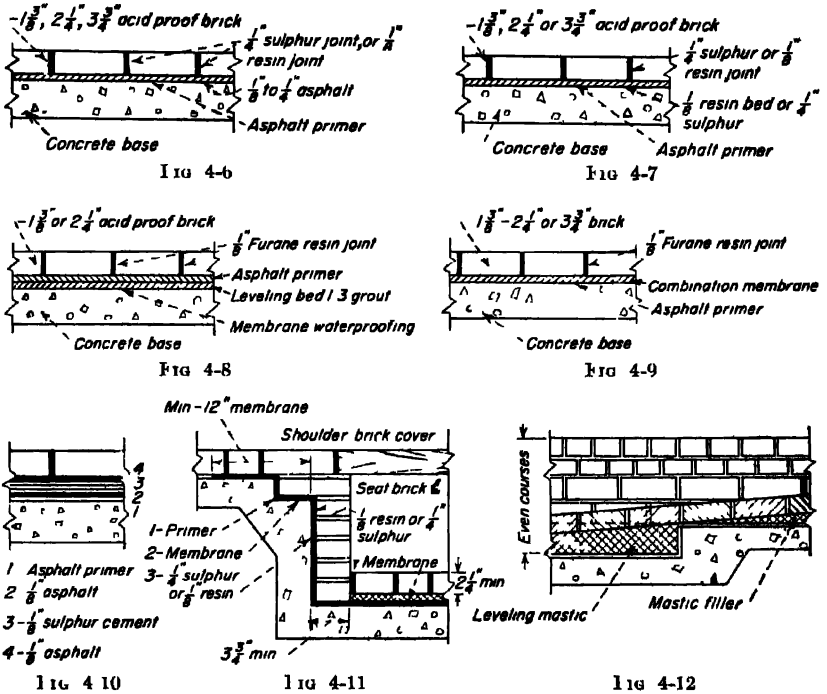

Fig 4-6 Fig 4-7 Fig 4-8 Fig 4-9 Fig 4-10 Fig 4-11 Fig 4-12

Figs 4-6 to 4-12. Acidproof brick floors

The membrane waterproofing should be constructed of alternate layers of impregnated cotton fabric and acid-resisting asphalt Glass cloth (Fiberglas by Owens-Corning) may be used with considerable advantage instead of cotton fabric, as poor workmanship often permits exposure of the cotton to the acid-bearing liquid with consequent deterioration

A typical section through an acid-bearing trench is shown in Fig 4-11 Note particularly that the shoulder and seat bricks are turned at right angles to the trench for rigidity The membrane, preferably of two-ply Fiberglas, is carried at least 12 in past the shoulder brick

The leveling acidproof mastic is used to permit the use of whole bricks for the side wall of the trench and at the same time to secure a uniform slope for the bottom course of brick. The method of stepping the depth at intervals so that only full bricks are used is shown in Fig. 4-12. Cutting of bricks is one of the most expensive labor items involved and should be eliminated. This is especially so when bricks are cut to a bevel. It is unnecessary, uneconomical, and results in a poorly constructed trench.

Table 4-3. Manufacturers' Index

1, 2, 8, 9, 10, 21	Atlas Mineral Products Co., Mertztown, Pa.
3, 4, 5, 18, 19	Nukem Products Co., Buffalo, N. Y.
6, 22	Sauereisen Cements Co., Pittsburgh, Pa.
7, 23	Installed by Ralph V. Rulon Co., Philadelphia, Pa.
11	U.S. Stoneware Co., Talmadge, Ohio
12, 24	Electro-Chemical Supply & Eng. Co., Paoli, Pa.
13, 14, 15, 17, 20	Pennsylvania Salt Mfg. Co., Philadelphia, Pa.
25	Quigley Co., New York, N.Y.
16, 26	Maurice A. Knight, Akron, Ohio

Similar materials are available from Haveg Corp., Newark, Del., La Clede Christy Clay Products Co., St. Louis, Mo., The Robinson Clay Products Co., Akron, Ohio.

Rubber. Rubber is a "dress-up" finish suitable for offices, laboratories (especially neoprene base), locker rooms, dining rooms, cafeterias (greaseproof type only), stores, public buildings, churches, hospitals, etc., where the surface is subjected to extensive foot traffic. It is only occasionally used in industrial work. It is a very comfortable floor finish and has good resistance to abrasion.

Rubber should not be used on ground floors or for exterior use. If it is desired on ground floor, membrane waterproofing or mastic underlayment should be provided to keep out moisture, and a waterproof type of adhesive should be used.

Rubber is supplied in thicknesses of ⅛ to 3/16 to ¼ in. Tile sizes are 4 by 4 in., 6 by 6 in., 9 by 9 in., 12 by 12 in., 24 by 24 in., and 25-yd rolls 36 in. wide. Some producing firms are Goodyear, U.S. Rubber, Ohio Rubber (Willoughby, Ohio), Armstrong Cork, American Tile & Rubber Co. (Trenton, New Jersey), Akron Rubber Co. (New York), B. F. Goodrich, and Selby-Battersby. (See Table 4-4.)

Plastic—Resin Type. Resin-type plastic is manufactured for both "dress-up" and industrial uses. It has characteristics similar to asphalt tile but has greater flexural strength and impact resistance. Hence it will stand industrial uses to a much greater extent than asphalt tile, cork, rubber, linoleum, or mastic. Plastic may be secured made of highly fire-resistant ingredients and also resistant to oils, gasoline, greases, acids, alkalies, and moisture. Installation is similar to that of other tiles, but it may be put in use immediately.

It comes in the same sizes and thickness as asphalt tile, although some manufacturers have sizes up to 18 by 24 in.

Since each manufacturer has his own specific mixture of plastic binders, plasticizers, asbestos fiber, pigment, etc., the chemical and physical resistances vary considerably more than any type of tile and should be thoroughly checked before purchasing.

Resin-type plastic is produced by Tile-Tex, Bird & Son, Armstrong Cork, Thomas Moulding, David E. Kennedy, Selby-Battersby, etc. (See Table 4-4.)

Linoleum. Linoleum is strictly an interior, above-grade floor finish. It will not stand subfloor dampness and should not be cemented directly to wood or metal, but installed over felt or mastic underlayment. Linoleum is supplied in rolls ⅛- to 3/16- to ¼-in. and 6-mm (0.236 in.) thickness, as well as medium gauge (0.095 in.) and standard gauge (0.079 in.), both plain and marbleized.

Linoleum is not suitable for rolling loads. In Bureau of Standards Tests, *Report* BMS80, however, marbleized linoleum withstood abrasion tests very satisfactorily.

It is not suitable where exposed to acids, alkalies, oils, gasoline, greases, or moisture. It is ideal for light foot traffic owing to resilience and resistance to abrasion.

Linoleum is manufactured by Armstrong Cork Co., Congoleum-Nairn, Sloan-Blabon, etc. An economical industrial grade ⅛ in. thick, identified as Mastipave, is also manufactured by Paraffine Industries. (See Table 4-4.)

Mastic Tile. Mastic tile is a general-purpose floor finish for both public buildings and industrial uses. It is economical in cost and maintenance if properly selected and installed. It is one of the few types of finishes that can be used on floors below grade and subjected to moisture.

Mastic tile should not be used for exterior exposure. The binder, filler, asbestos fiber, etc., may be changed to secure a considerable range of resilience, hardness, toughness, brittleness, colors, etc. It can also be secured with oil, chemical, and greaseproof resistance.

Although it is available in $\frac{1}{8}$-in. thickness, lower maintenance and longer life makes the $\frac{3}{16}$-in. thickness much preferred. Four price groups, A, B, C, and D are usually available, Group A, plain colors, being the lowest priced. Available sizes are 6 by 6 in., 9 by 9 in., 6 by 12 in., 12 by 12 in., 9 by 18 in., 18 by 24 in., and 12 by 24 in. Octagonal and hexagonal shapes may be secured on special order.

When placed on wood floors, an underlayment of at least $\frac{5}{16}$ in. should be used. Where installed below grade, it is advisable to use a mastic underlayment to reduce maintenance. Aluminum or brass edging strips should be used on stair treads or other termination edges.

Some producers are Tile-Tex Co., Johns-Manville, Armstrong Cork Products, Thomas Moulding, David E. Kennedy, Duraflex Co., Uvalde Rock Asphalt Co., Selby-Battersby Co., Mastic Tile Corp. of America (Newburgh, New York), etc. (See Table 4-4.)

Mastic Block. Both mastic block (bricks) and planks are available for interior or exterior use as

Industrial floors—Factories; airport hangars, runways, etc.
Piers and docks—Bridges and approaches
Roof decks—Parking space, miscellaneous traffic, storage
Warehouses—Packing and canning factories
Driveways—Courts, terraces

Blocks are obtainable in such sizes as 4 by 8 in., 4 by 12 in., 4 by 16 in., 12 by 24 in., and hexagonal (Hastings). Planks are obtainable in 12- by 24-in. and 12- by 36-in. sizes as standard.

The thicknesses of blocks generally available are $1\frac{1}{4}$ to $1\frac{1}{2}$, 2 to $2\frac{1}{2}$, and 3 in.

Planks are available from $\frac{1}{2}$ to 2 in. in thickness, and special thicknesses may be ordered.

The weight of the blocks usually averages 9.5 lb per sq ft per 1 in. thick; weight of the planks is somewhat lighter, usually 8.0 lb per sq ft per 1 in. Material consists of asphalt, asbestos or similar fiber, mineral filler, silica flour or crushed limestone, heated and pressed.

The advantages of mastic block are that it is self-healing, water-resistant, resistant to wear, acid, and trucking abuse It is resilient and free from decay and dry or wet rot. It is quiet, slipproof, odorless, and easily repaired. Special nonslip surfaces are available, usually identified as "mineralized plank" or "mineral surfaced." Johns-Manville, Carey Co , Asphalt Block Pavement Co., Servicised Products, and Hastings Co., are typical of the more reliable producers.

The usual thickness for heavy-duty floors where blocks are necessary is 2-in. For light-duty traffic the surface may be as thin as $\frac{1}{2}$ in., but in all cases the installation should be discussed with the manufacturer's engineers.

Blocks are usually bedded in a 1:3 portland-cement base $\frac{1}{2}$ in. in thickness. Blocks are set tight with staggered joints. Vertical joints in acid-resisting work should not be filled, but it is necessary to set the blocks or planks in asphalt or over a two- or three-ply waterproofing membrane. For nonacid work, the joints may be filled with fine sand or a cement grout. (See Table 4-4.)

Cork. Cork is a special-purpose floor finish usually selected for elevator cars, working space in high-occupancy offices, bathrooms, and stair treads and landings. It is exceptionally quiet and comfortable, but its resistance to abrasion is low. The resistance may be increased by applying lacquer.

Cork is obtainable in $\frac{1}{4}$- to $\frac{5}{16}$- to $\frac{1}{2}$-in.-thick tiles; 6- to 9- and 12-in. squares, which are cemented to floors with a special adhesive. A special cork-cellulose binder tile is made in ⅛-in. thickness by Bird & Son.

Rectangular sizes such as 6 by 12 in., 3 by 36 in., 12 by 18 in., 6 by 18 in., 6 by 36 in., 3 by 12 in., up to 18 by 36 in., etc., are also available

Cork is not recommended for below-grade installation or where the surface is subjected to continuous moisture or heavy moving loads.

Cork is produced by Bird & Son, David E. Kennedy, Armstrong Cork, Goodyear, Cork Insulation Co , Selby-Battersby, etc. (See Table 4-4.)

Magnesite (Magnesium Oxychloride). Magnesite has a high compressive strength varying from 8,000 to 12,000 psi. It is resilient, excellent for standing or walking operators; attractive, as it may be secured in many colors; and sanitary.

Table 4-4. Characteristics of Floor Finishes

Characteristic	Cement	Mastic	Wood block	Wood, laminated	Brick	Rubber	Plastic tile	Linoleum	Mastic tile	Mastic block	Cork	Magnesite
Cost index	3[a]	7[b]	[c]	7[d]	8[e]	6	5	1	2	1	3	4
Ease of repair	3	1	1	1	4	1	1	2	1	1	1	1
Fast installation—days	5[f]	2	1	1	1	1	1	2	2	1	1	2–5
Compressive strength	2	4[g]	5	6	1	8	3[h]	8	4	3	7	2
Tensile strength	2	4	8	5	8	1	1	7	3	8	6	1
Toughness	2	3[i]	1	3	7[k]	5	1	6	3	2	4	3
Resistance to denting	2	4	4	5	1	2	3[g]	5	4	3	5	2
Abrasion resistance (wear), in	0.025[l]	0.007–0.032	[p]	0.002[m]	[q]	0.006	0.002	0.002	0.007	0.007–0.032	0.011[n]	0.006
Sparkproof[s]	N[x]	N[x]	Y	Y[r]	N	Y	Y	Y	Y	N[x]	Y	Y
Self-healing	N	1	N	N	N	N	N	N	N	2	N	N
Resistance to moisture[t]	3	2	N	N	1	4	2 4	N	4	3	N	N
Easy wheeling	1	3	3	3	2	[u]	2	[u]	[u]	3	[u]	1
Nonskid—walking[y]	3	2	2	3	2 4	3	3	2[w]	2	2	1	2
Resilient, comfortable	11	9	6	4	12	1	7	3	10	8	2	5
Variety of color	4	N	N	N	N	1	1	1	1	N	2	3
Quiet	11	9	5	4	12	2	6	3	10	7	1	8
Odorless (after 7 days)	4	3	3	2	2	2	1	1	1	1	1	1
Acid- and alkali-proof	9	2	7	8	1	6	3	12	5	4	10	11
Fire resistance	1	6	7	11	1	9	3	10	4	5	8	2
Nominal thickness, in	4–6	1½	2–3	2½ 4½	1⅜ 4½	⅛ ¼	⅛ ¼	⅛ ¼	⅛ 3/16	½ 3	⅛ ½	⅝ ¾
Weight, lb/sq ft*	5 60	15–20	2¼ 9	3 12	2¼ 25–35	3/16 0.8	3/16 1.5	3/16 0.9	3/16 2.0	2 16–20	¼ 1.58	½ 3–6

NOTE: Lowest number indicates highest quality, in cost index, the lowest cost. Y = Yes N = No or None

[a] Separate topping, metallic hardener

[b] With acidproof membrane under mastic

[c] Creosote-impregnated end-grain lug block (Kreolite)

[d] Tar-Rok

[e] Acidproof 2¼ in brick

[f] 24 hr for Lumnite cement

[g] Can be hardened if necessary, but increases tendency to crack or chip

[h] For hard type

[i] Soft base, tough top

[k] Hard to classify for toughness Will stand considerable mechanical abuse.

[l] Based on 1.2.4 mix. Regular 1:1.2 mix with fluosilicate wash should be approx. 0.012

[m] Based on maple finish

[n] High density, 5/16 in dark = 0.026 in 5/16 in light = 0.011 in., ½ in. light = 0.019 in.

[p] No available tests—probably about 0.035 in

[q] No available tests, probably about 0.005 to 0.010 in for shale

[r] Concealed or copper nails

[s] Cement, mastic, rubber, linoleum, mastic tile, and magnesite may be purchased with conductive quality to prohibit sparking from static discharges

[t] Moisture to extent that material cannot dry between periods

[u] Not recommended

[w] Without wax or lacquer

[x] Can be obtained on special order

[y] Dry surfaces, designed for traffic

* For thickness indicated.

When copper salts are included, growth of bacteria is discouraged, hence magnesite is excellent for locker rooms (Hubbellite, H. H. Robertson Co. patent). It is also avoided by insects. It is *not suitable* for exterior use owing to absorption of moisture and disintegration. It will stand flushing with water if drained or mopped off so that the material can thoroughly dry between wettings. It can be supplied with conductive quality (Kompolite Co., Thomas Moulding Co., etc.) to eliminate static discharge when desired for use around explosive vapors or solvents.

Magnesite is suitable for heavy loads such as trucking aisles—has a fast-trucking finish.

Thickness over concrete should preferably be ⅜ in. or over. When installed on wood, it should be approximately ½ to ¾ in. thick to permit the installation of a leveling underlayment over chicken wire tacked over building paper. Use a metal molding to separate plaster walls from a magnesite base.

Fillers such as wood flour, silica flour, etc., are often used to build up thickness.

Probably no type of floor finish has caused so much grief from improper use or more satisfaction from proper use and installation as the magnesium oxychloride type. It is essential that an experienced contractor install the floor and that the ingredients be properly blended for the intended use.

Other typical types are made by Selby-Battersby, Federal Flooring Corp. (New York), Marbleloid Inc. (See Table 4-4.)

ROOFING

Next to floors, there is probably no other part of a building, especially an industrial building, that causes as many maintenance headaches as roof covering, especially on large buildings.

In spite of this, the rules governing proper construction and repairs are well known by all the reputable roofing manufacturers and are available to all interested parties if they request assistance from the manufacturers' technical division and *not* every Tom, Dick and Harry who has applied a piece of felt at some time in his life. If your local roofer is capable of advising you, the manufacturer will so advise you.

Volumes could be prepared on proper installation, selection of proper bitumens, flashing, repairs, etc., so it will be obvious that the following data can describe only a few of the pertinent points to be observed for minimum maintenance.

Basic Application Requirements. The use of asphalt or asphalt-saturated felt on a roof of extensive area having a slope of ½ in. per ft or less should *not* be permitted. For slopes over ½ in. such material may be used if desired. No slag or gravel surface is required.

Coal-tar-pitch-saturated felts and coal-tar pitch with slag or gravel finish should be used for all flat roofs, especially those having slope of ½ in. per ft or less. A 20-year bonded roof can be secured with this type up to slopes of 5 in. in 12 in.

The mixing of asphalt and tar should never be permitted. If it is necessary to repair an old smooth-surfaced asphalt roof and convert it to a coal-tar-pitch roof, a 15-lb unsaturated felt or a 15-lb asphalt-saturated asbestos felt should be employed as the separating medium. In this case the felt may be cemented to the roof with asphalt, but the top mopping should be pitch. Asbestos felt is preferred because the amount of saturant is less than that of rag, and it is less liable to cause trouble owing to the presence of the asphalt undercoat and pitch overcoat.

No more pitch should be poured at one time than can be immediately covered with slag or gravel. It is essential that the pitch be hot to hold the slag properly.

The temperature of pitch should not exceed 400°F in the kettle. The proper temperature for placing is from 325 to 375°F. If the white vapor coming from the kettle turns yellow, the fire should be reduced and cold pitch added.

The temperature of asphalt should not exceed 450°F in the kettle. Pouring temperatures should be from 350 to 400°F.

The top coat of asphalt on smooth-surfaced roofs should not exceed 25 lb. Too heavy an application will result in checking or "alligatoring."

All masonry should first be primed before applying asphalt. Polished concrete surfaces should be primed with a creosote primer before applying pitch.

Manufacturers will not issue a 20-year flashing bond where 8-in. thick parapet walls exceed 12 in. in height owing to possible leakage through the walls.

Parapet walls over 12 in. high should be waterproofed with a transparent solution similar to Sonneborn & Sons Hydrocide Colorless if the flashing is carried up to the coping. Trouble is certain without it.

The flashing is the most difficult part of the roof to construct and the first to become defective. Each manufacturer has his own pet method of installation which he is willing to insure with a guarantee bond.

If a high parapet is necessary and a watertight installation is desired, a through-wall flashing should be used immediately above the counterflashing so that all water penetrating the wall will spill out over the roof and not under the flashing.

Don't put a new roof over a wood deck having warped boards or open joints. A reputable roofer will not guarantee the installation, and you are inviting perpetual trouble if you insist.

Don't waste your money on insulation by leaving out a moisture barrier under the insulation. If you use 3 in. or more, ensure the job by installing a barrier of at least two-ply saturated felt.

Considerable research has been conducted by the various roofing manufacturers during the past 5 years on the elimination of "blisters." One of the contributing elements is "green" concrete, especially where there is a "pressure head" within the building due to air conditioning or a fairly high temperature below the roof. The moisture is thus driven out of the slab, and the vapor makes its way upward and through the layers of roofing material, forming "blisters." It has also been discovered that, where roof insulation is blanketed off into *squares*, the tendency to form blisters is greater because the gas is sealed in within the "square." Roofing specifications should indicate the placing of the moisture seals in *parallel* strips about 20 ft 0 in. apart. In other words, the cutoffs should all run in the same direction. A cutoff is also recommended around large openings, not more than 18 in. away from the face of the opening, curb, or wall. The simplest method of making the cutoff is to divide a 15-lb sheet into three parts (about 11 in. each) for 1-in. insulation and two parts (±16 in. each) for 2 in. or over and install it as shown by Fig. 4-13.

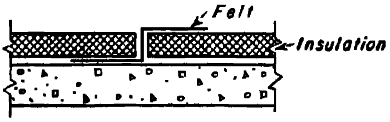

FIG. 4-13. Roof insulation cutoff.

In areas where wind is a factor and where you desire the most satisfactory roofing over wood, insist on the first felt being securely nailed with either "simplex" nails or nails with disks at least $\frac{7}{8}$ in. in diameter. The first felt should be 30 lb or two layers of 15 lb. Over concrete, the first layer may be 15 lb, but it should be thoroughly mopped under its entire area.

Where a slag or gravel roof is used, be certain that slag stops are used at the eaves and at the drain head. The roof drain, if properly selected, will have suitable slag and flashing rings. If the slag or gravel is properly installed, you will not have loose material to clog up your gutters or drains.

Where a metal deck is used, at least $\frac{1}{2}$ in. of roof insulation must be used in order to secure a guarantee-bonded roof. All reputable manufacturers have the same requirement.

Felts. With the modern methods used by manufacturers today, there is little difference in the efficiency or ease of saturation required for either rag or asbestos felts. In the case of rag felts, the fibers as well as the interstices are filled with pitch or asphalt, while in the case of asbestos the fibers are coated with pitch or asphalt and the remaining saturant is in the interstices. While this difference in impregnation may give the impression that the rag felt is superior because the fibers are filled with saturant, it should also be kept in mind that there is nothing in the chemical make-up of asbestos which would permit it to oxidize or disintegrate even if not saturated with bitumen, and its life will depend upon the tensile strength or quality of the fibers. Its waterproofing quality will depend only on the interstices being completely filled with a waterproofing medium. The degree of saturation and weights of the two types are as follows:

Asphalt-saturated asbestos felts—ASTM D250-39T; desaturated weight 9 lb min, saturant to be 50 per cent dry weight of felt, min

Asphalt-saturated rag felt—ASTM D226-38T, desaturated weight 5.2 lb min saturant to be 140 per cent dry weight of felt, min

Weights are for 100 sq ft of fabric

Rag felts have a higher tensile strength than asbestos felt, and their proponents contend that the field application of the hot asphalt or pitch adheres and spreads easier over rag than asbestos felts. In a long U- or L-shaped building, the rag-felt roofs withstand the strain from expansion and contraction at the corners better than asbestos felts. A saturated cotton fabric is stronger for this purpose, usually having a tensile strength of between 50 to 60 lb per in both ways.

Asbestos felts *are* more fire-resistive than rag felts, even though the asphalt or pitch may be inflammable in both cases.

Asbestos felts withstand direct exposure to the sun and heat much better than rag felts, especially when the roof surface is alternately wet and dry. Rag felts rot out rapidly if used on a flat or semiflat surface if alternately wet and dry, especially if not protected by sufficient bitumen, slag, or gravel.

Asphalt versus Pitch. There is no doubt that a good grade of either asphalt or pitch will prove absolutely satisfactory if used properly.

Even a medium grade of asphalt or pitch will last many years if protected from the sun by a slag or gravel finish, but if it is not protected, the oils will burn out and the surface will crack or "alligator."

Pitch should *never* be used on a steep sloping surface because it will run in hot weather. "Steep" pitch should be specified when used on a surface sloped more than 2 in. in 12 in.

Asphalt, on the other hand, should be used on steep slopes and may be used on any surface where water will *drain entirely off* the roof areas and not stand in puddles.

Pitch has a self-healing quality and the ability to withstand exposure to water and sun. The property of being self-healing is advantageous when water manages to get under a layer of felt through a leak or crack, as it does not prevent the layers from sticking together later after the water has evaporated and does not affect the waterproofing qualities of the pitch.

Material Specifications. Where reference to standard material is required, it is recommended that the following be used as a guide:

Table 4-5

Tarred felts, ASTM spec	D 227
Coal tar pitch for roofing	
U S Govt master spec	R P-381, Type 1
ASTM spec	D-450
Creosote primer, ASTM spec	D 43
Plastic roof cement Federal spec	SS-C 153-Types 1 and 2
Asphalt primer, ASTM spec	D 41
Asphalt felts (rag), ASTM spec	D-226
Asphalt for roofing, ASTM spec	D-312
Asbestos felts, asphalt-saturated with min 85 per cent asbestos fiber ASTM spec	D-250
*Cotton fabric—asphalt-saturated, ASTM spec	D-173
Cotton fabric—coal-tar-pitch saturated ASTM spec	D-173

* See below for more definite specification, which is preferred

The use of an impregnated cotton fabric for repairing flashing, especially eaves, corners, or long parapet walls where expansion and contraction occur is excellent because of the inherent tensile strength of the fabric. Lead flashing has proved unsatisfactory on long runs because of buckling from expansion.

An excellent specification for the fabric, where used as roof flashing, waterproofing membrane, or spandrel flashing, is as follows:

Asphalt- or Coal-tar-saturated Cotton Fabric. An asphalt- or coal-tar-saturated all-rag woven fabric shall have a tensile strength, both directions (grab method), of not less than 50 lb per lin in. and not less than 10 per cent elongation. The fabric shall weigh before saturation not less than 4 oz per sq yd and after saturation not less than 12 oz per sq yd and shall have a thread count between 18 by 18 in. and 36 by 36 in. In all other respects fabric shall conform to ASTM D173.

WALLS

Brick is one of the oldest of our building materials and is still one of the most extensively used. Our modern methods and constructions are the result of long usage instead of theoretical analyses, which is so essential with most building materials.

The masonry of this period has deviated somewhat from that of our ancestors in that we have combined brick with numerous shapes and types of units such as tile, concrete, terra cotta, etc. The greatest improvement, however, has been made by the development of textures and colors, especially in the ceramic field, and improvement in deairing and baking.

For industrial use, it is essential to select the most economical wall both in first cost and maintenance and one that is suited for special conditions such as extremes in weather, moisture, acid-fume resistance, and often for sanitary reasons such as in plants processing foods or medicines.

Structural requirements with hints on special precautions to be followed in construction will be discussed followed by comparisons of various types of walls.

Structural Requirements. Usually local building codes limit the thickness, height, laterally unsupported lengths, etc., of all types of walls. However, for quick reference a few limitations are presented.

1. *Height of Walls.* Exterior nonbearing walls are usually limited to a height of twenty times their thickness for solid masonry, fourteen for cavity walls, and eighteen for hollow walls where the maximum horizontal distance between supports is not over 20 ft 0 in.

2. *Interior Partitions.* Considerable variation exists in the requirements for interior nonbearing partitions, but thirty times the thickness is usually considered maximum height where loads are not bracketed or supported by the wall, especially for industrial buildings where vibration is present. A few of the building codes are more lenient where an interior partition is built solidly against the ceiling.

3. *Masonry Walls.* For general code limitations on masonry walls an excellent bulletin, CP33, may be secured from the Portland Cement Association.

4. *Weather Tightness.* With the finest material and workmanship, an 8-in. wall *may* be made watertight, but the chances are about 10 to 1 that it will leak. A questionnaire to over one hundred of the largest and most reputable contractors in the entire United States failed to reveal a single contractor who would guarantee a tight wall under 12 in. thick. Experience has indicated that it is extremely difficult to make a tight vertical through joint.

For maximum impermeability, solidly filled mortar joints, low-absorption brick or tile, preferably shale, a minimum of 12 in. in thickness should be used, and a mortar having a high-plasticity lime, high water retentivity, and low shrinkage mix, as described below should be used. If an 8-in. wall is desired, besides the above requirements, a good stearate waterproofing admixture should be added to the mortar and then an extra 3 cents per sq ft spent in applying a good transparent waterproof wash to the exterior, similar to Sonneborn Sons' Hydrocide Colorless. Joints in all cases should be tooled to secure a dense concave joint.

For a good treatise on permeability, refer to Bureau of Standards *Report* BMS-82, and for strength, absorption, and resistance to freezing and thawing of bricks throughout the United States, BMS-60.

Brick and Tile. For maximum resistance to freezing and spalling, moisture penetration, strength and volume change, a dense deaired unit, preferably of shale, having a 5-hr boiling absorption of less than 8 per cent, should be used.

All brick having an absorption of over 20 g per min when the flat side of the brick (32 sq in.) is immersed in $\frac{1}{8}$ in. of water should be wetted sufficiently to decrease the absorption to that ratio when laid.

As a general rule, all-brick walls offer the greatest opposition to moisture penetration, brick and tile next, and tile last. Quoting from Bureau of Standards *Report* BMS-82:

> Twelve-inch walls faced with brick and backed with hollow units were more permeable than similar all-brick walls. The kind of hollow unit used in the backing had no consistent or important effect on permeability, but the walls containing end-bearing units appeared to be slightly less permeable than those containing side-bearing hollow units.

Bricks exposed to weather should conform to ASTM C62 grade MW; structural clay tile, to ASTM C34, grade LB for bearing walls or piers, grade LBX when exposed to weather; ceramic glazed tile, ASTM C126, nonload-bearing tile, ASTM C56; concrete hollow load-bearing masonry units, ASTM C90; concrete solid load-bearing units, ASTM C145.

Tile for exterior walls and interior partitions may be classified as (*a*) plain dense deaired red shale (often designated as red industrial tile); (*b*) buff, iron-spot (ferro glaze), or manganese-spot unglazed facing tile; (*c*) ceramic glazed tile of many colors, the most economical color of which is "clear glaze" or "lustre glaze," which is a buff or light ivory.

Salt-glazed tile is not due any consideration as the price differential does not justify its use over the "clear-glazed" unit. The salt-glazed finish is *not* an impervious finish but merely provides a "sheen." If it is examined under a magnifying glass, its numerous cracks will become evident. The clear-glazed tile has an impervious surface, is easily cleaned and pleasant in appearance. Manufacturers will provide the salt-glazed finish upon request, but they much prefer to supply the clear glaze because greater satisfaction will ensue.

Sizes. The standard brick is $2\frac{1}{4}$ by $3\frac{7}{8}$ by 8 in. for smooth-surfaced; $2\frac{1}{4}$ by $3\frac{3}{4}$ by 8 in. for common and rough-faced brick as established by ASTM. Space does not permit discussion of the many methods of manufacturing or classifying the types and surface finishes.

The Glazed Brick and Tile Institute has classified tile sizes as shown in Table 4-6.

Uses. For excellent appearance, low maintenance, good housekeeping with minimum effort, dustless, sanitary construction for food or pharmaceutical processing, ceramic or clear glaze should be used. A disadvantage is its high first cost.

Table 4-6

Face size	Series	Thickness
5 × 8 in	N or D Series	$1\frac{3}{4}$-$3\frac{7}{8}$-$5\frac{7}{8}$-8 in
5 × 12 in	T Series	$1\frac{3}{4}$-$3\frac{7}{8}$-$5\frac{7}{8}$-8 in
$2\frac{1}{4}$ × 8 in	S Series	$3\frac{7}{8}$

For modification of the above, as required for manufacture of dustfree products, the buff unglazed facing tile may be used to within about 5 ft 0 in. of the floor, the wainscot being either clear-glazed buff, iron spot (ferro glaze), or manganese spot. This provides a clean light-reflecting surface above and a wainscot that will not show elbow or hand smudges down the trucking corridors and will eliminate the usual 3- to 5-year periodic maintenance painting. It is sufficiently hard to withstand mechanical abrasion. A disadvantage is that it is medium high priced.

For good appearance, impervious exterior and interior, low maintenance, low cost, and good housekeeping, deaired red shale facing tile (industrial red tile) may be used. Walls have the color of normal red brick but have fewer mortar joints owing to larger size and a smoother and more dense face. It has been demonstrated that a mason can lay up 10 sq ft of this type of unit in the 5- by 12-in. face in the same time it takes him to lay up 7 sq ft of brick wall. Immunity from attack by acids may be afforded by pointing the mortar joints with a phenol- or furane-resin mortar (see Brick for floors, pp. 148–149).

For maximum strength, maximum impermeability for the same thickness, maximum mechanical abuse, good to fair appearance, good resistance to freezing, thawing, and acid atmospheres, the dense shale brick is hard to beat. Some disadvantages are that housekeeping is more difficult, and there is no decorative appearance without maintenance painting. Also more numerous mortar joints are exposed to attack from acid vapors and liquors, but this may be remedied as noted above for tile.

Mortar. Next to workmanship, the quality of the cementing medium is most important. The conclusions expressed herein are the result of 30 years of intimate observation. Strange as it may appear, substantiation appeared in an excellent article by Watstein and Seese in the August, 1947, issue of *ASTM Bulletin.*

There are four important types of masonry mortar, which for the purpose of this article will be designated as types I, II, III, IV.

Type I—1:0.25:3 mix by volume of portland cement, lime, sand

Type II—1:1:3 mix by volume of portland cement, lime, sand

Type III—1:1:6 mix by volume of portland cement, lime, sand

Type IV—Commercial masonry mortars meeting Federal Specification SSC-181B-type 2

Type I has the highest compressive strength—approximately 6,000 psi, with types II, III, IV decreasing in order to about 1,000 psi. Bond strength of type I is approximately 65 psi, decreasing in order to about 40 psi for IV. Water retentivity is approximately 70, 75, 80, and 80 per cent in order. (Type I is frequently varied by some authors to 1:0.1:3)

For all types of tile, with the possible exception of glazed furring tile, whether horizontal or vertical cell, type II is emphatically recommended. Where the fineness of the sand is suitable, a 1:1:4 is also satisfactory.

Type III is recommended for all brick walls except where walls are over twenty feet high, run over one hundred feet, or where high winds are frequent. In such cases a 1:1:5 or even a 1:1:3 should be used to secure the necessary compressive and bond strength with reduced volume change. Watstein and Seese, in the *ASTM Bulletin,* have expressed the following conclusions:

> There was no consistent relation between the shrinkage of the various mortars and any of their other properties. However, for mortars of similar composition, such as the portland cement-lime mortars, the shrinkage increased roughly with the *amount of mixing water* required. Of these mortars, the minimum shrinkage was shown by the group 1PL (type I) in which the average for five mortars, after 6 months' drying, was about 0.12 per cent.

(The average shrinkage of the 1:1:6 mix was 0.148 in ; 1:2:9 mix was 0.157 in., and type IV was 0.143 in. On the basis of proportions, the type II should therefore have a shrinkage value of between 0.13 and 0.14 in.)

For ceramic glazed furring tile, type I would be preferred if the mason could be influenced to mix an additional type.

Mortars are no different from concrete. The more water put in the mix, the greater the amount of water that must be eliminated, with consequently greater volume change. Volume change induces *cracks.*

Lime. Lime should conform to the National Lime Association specification for mason's hydrated lime type S, having a plasticity of more than 200 after wetting (not soaking) and a water retentivity of more than 85 per cent when tested in a mortar made from the dry hydrate. The ordinary run of commercial hydrated lime does not conform to the above requirements, however. If it must be used, insist on a soaking period of not less than 24 hr before use.

Typical examples of several very satisfactory commercial limes are Miracle by G. & W. H. Corson, Plymouth Meeting, Pennsylvania; Rockland-Rockport Lime Co., Rockland, Maine; Flamingo Hydraulic Lime by Riverton Lime & Stone Co., Riverton, Pennsylvania; Super Limoid by Warner Co.

Economic Comparisons of Walls. Fourteen of the most common combinations of brick and tile are presented on p. 162, and their cost factors are indicated.

Cost factors are based upon brick and tile prices for the Philadelphia area and are listed below. Labor rates are also indicated. Conversions are therefore readily made for any local costs. One hundred square feet of plain wall is used as the basis. Mortar quantities are minimum with no waste allowance. No profit or overhead is included. Therefore, cost factors are to be considered as *purely relative costs.*

One-quarter-inch mortar joints are included in all types except 1.0, 4.0, 4.3, 8.0, 8.1, 8.2, 11.0, 11.1, 12.0, 12.1, 13.0, 13.1, 14.0, 14.1, and 14.2 (see Figs. 4-14 to 4-22, inclusive, and Table 4-8), for which $\frac{3}{8}$-in. joints are necessary. Mortar costs are included with labor costs. Labor costs include tooled joints, both faces.

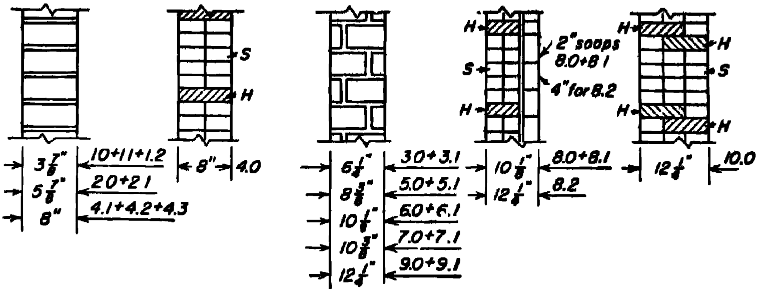

FIG. 4-14. FIG. 4-15. FIG. 4-16. FIG. 4-17. FIG. 4-18.

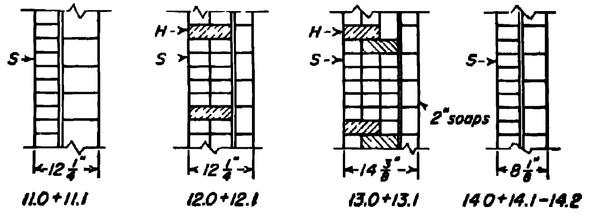

FIG. 4-19. FIG. 4-20. FIG. 4-21. FIG. 4-22.

FIGS. 4-14 to 4-22. Brick and tile walls. Proper bonding for plain and composite construction.

Basis of costs for tabulation:

Table 4-7

Type	Buff facing tile, /M	Ivory luster glaze—clear, /M	Iron spot-ferro glaze—facing tile, /M	Red shale facing tile, /M
TA—2 × 5 × 12-S1S	$ 90	$150	$ 92	$ 53
TAK—4 × 5 × 12-S1S	124	284		
T—4 × 5 × 12-S1S	116	197	119	81
TD- 4 × 5 × 12-S2S	130	292	132	88
T60 6 × 5 × 12-S1S	145	277	164	139
T60D -6 × 5 × 12-S2S	172	360	179	147
T80—8 × 5 × 12-S1S	195	333	210	151
T80D—8 × 5 × 12-S2S	220	449	336	162

Brick—red shale industrial, wire cut, smooth face, $30/M

Labor rates - mason at $2 25/hr and helper at $1 30/hr

Number of units per square foot of wall—4 in. brick wall = 6.25, 8 in. = 12.50; 12 in. = 18.5; 5 by 12 in. = 2.2.

Where 4- and 2-in tiles are used on same wall, it is assumed that 4-in. Kerfed tile (TAK) will be used for the 2-in. tile. Where 2-in. tiles are used with units other than 4-in , the 2- by 5- by 12-in. unit (TA) is used. Modular units are not considered, but their use will not alter the relative unit costs. No allowance is made for breakage or defects.

KEY: FT = buff facing tile, unglazed; GT = glazed (clear); FGT = ferro glazed facing tile; RFT = red shale industrial facing tile; H = header course; S = stretcher course.

The use of 4 + 2 or 6 + 4, etc., self-bonded walls with reversed or staggered joints results in a superior wall both structurally and physically as no through vertical joints exist, and both exterior faces may be easily lined up to secure a smooth, properly aligned face. Variation in size of tile is compensated for by variation in vertical mortar joint. Unbonded walls should be tied together with metal ties, preferably with bronze or monel wire ties, every 24 in. horizontally and every 15 to 18 in. vertically.

A saving in material price may be obtained by using a 4-in. Kerfed unit where possible, especially when both 4- and 2-in. units are used, as the 4-in. type may be split in half (tile symbol TAK).

Types T60D and T80D are not supplied by all manufacturers. Wall types 2.0, 2.1, 4.1, and 4.2 are not so satisfactory as other types of the same thickness because of the lack of uniform thickness with the resultant uneven appearance of the finished surfaces. They are inferior structurally and are more permeable than the staggered construction of types 3.0 and 5.0. Types tabulated are based upon having the same surface on both sides. For variation, the quantities tabulated with the corresponding price lists may be easily adjusted (see Table 4-8).

Five- by eight-inches D series tile is not tabulated, since the 5- by 12-in. T series has a lower labor cost. The D series, however, generally presents a better appearance owing to easier alignment and less variation in dimensions. There are 2.2 units of 5- by 12-in. tile per sq ft of wall and 3.3 units of 5- by 8-in. tile per sq ft. A ¼-in. mortar joint may be advantageously used with the 5- by 8-in., tile, but masons usually prefer the ⅜-in. joint for the 5- by 12-in. tile, although ¼ in. is preferable for both maintenance and appearance.

Table 4-8

Type	Wall, in.	Thickness, in.	Masonry units, 100 sq ft	Mortar, cu ft	Mortar cost	Labor, 100 sq ft	Total sq ft	Cost factor
1.0	4B	3⅞	18.75	7.0	3.29	42.00	0.61	1
1.1	4FT	3⅞	28.60	3.4	1.60	50.00	0.79	2
1.2	4GT	3⅞	64.24	3.4	1.60	50.00	1.14	5
2.0	6FT	5⅞	37.84	5.0	2.35	72.00	1.10	4
2.1	6GT	5⅞	79.20	5.0	2.35	72.00	1.51	12
3.0	4 + 2FT	6¼	39.16	10.0	4.70	85.00	1.24	7
3.1	4 + 2GT	6¼	71.58	10.0	4.70	85.00	1.60	13
4.0	8B	8	37.50	18.5	8.70	78.00	1.16	6
4.1	8FT	8	48.40	7.0	3.29	84.00	1.32	10
4.2	8GT	8	98.78	7.0	3.29	84.00	1.83	19
4.3	8RFT	8	35.64	7.0	3.29	72.00	1.08	3
5.0	6 + 2FT	8⅜	45.54	11.4	5.36	116.00	1.62	14
5.1	6 + 2GT	8⅜	92.18	11.4	5.36	116.00	2.08	24
6.0	6 + 4FT	10⅛	57.42	11.8	5.50	116.00	1.73	16
6.1	6 + 4GT	10⅛	104.28	11.8	5.50	116.00	2.20	27
7.0	8 + 2FT	10⅜	56.54	12.5	5.88	128.00	1.85	20
7.1	8 + 2GT	10⅜	104.50	12.5	5.88	128.00	2.33	28
8.0	8B + 2FT	10⅛	47.40	21.0	9.87	128.00	1.75	17
8.1	8B + 2GT	10⅛	54.00	21.0	9.87	128.00	1.82	18
8.2	8B + 4GT	12¼	80.84	23.5	11.00	128.00	2.09	25
9.0	8 + 4FT	12¼	68.42	13.7	6.44	128.00	1.96	22
9.1	8 + 4GT	12¼	116.60	13.7	6.44	128.00	2.45	29
10.0	12B	12¼	55.50	28.5	13.40	114.00	1.70	15
11.0	4B + 8FT	12¼	61.65	17.7	8.32	126.00	1.88	21
11.1	4B + 8GT	12¼	92.01	17.7	8.32	126.00	2.18	26
12.0	8B + 4FT	12¼	63.02	28.5	13.40	122.00	1.85	20
12.1	8B + 4GT	12¼	80.84	28.5	13.40	122.00	2.03	23
13.0	12B + 2FT	14⅜	75.30	32.0	15.00	158.00	2.33	28
13.1	12B + 2GT	14⅜	88.50	32.0	15.00	158.00	2.47	30
14.0	4B + 4FGT	8⅛	44.93	13.5	6.35	85.00	1.30	9
14.1	4B + 4FT	8⅛	44.27	13.5	6.35	85.00	1.29	8
14.2	4B + 4GT	8⅛	62.09	13.5	6.35	85.00	1.47	11

NATURAL AND SYNTHETIC RUBBER*

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Owing to its unique properties, natural rubber has developed into an important material of construction. In the last 15 years a large number of rubberlike materials called "synthetic rubbers" have been developed. In a number of cases these are replacing natural rubber, but in a great many cases their unique properties are opening up new fields of application. These fields are so complex that it is impossible to outline any definite rules as to which material should be used for a given application. The following discussion will indicate the general types of materials available and their physical characteristics, but anyone interested in a specific application should consult an expert rubber technologist.

NATURAL RUBBER

Natural rubber has been employed as a material of construction for many years. It has basic properties which the technical men have been able to exploit in a great many ways. Elastically it is unique in allowing essentially reversible elongations up to 600 to 1,000 per cent. It can be compounded to give high tensile strength, *i.e.*, 40,000 psi of cross section at break. It is very water-resistant and has a very low permeability to water; it likewise has low permeability to a number of the more common gases; it is a good electrical insulator; it can be compounded to give material of outstanding abrasion resistance; and if desired, it can be converted into a hard material of low-percentage elongation of break.

While it is not possible to obtain all the properties listed above in the same sample, it is generally possible to obtain a combination of several of them. For example, in automobile tires and belting, it is possible to combine a high degree of flexibility or elasticity with superlative abrasion resistance. In inner tubes, balloons, and rubber gloves, it is possible to combine flexibility and high-percentage elongation with the impermeability to gases and water. For vibration damping it is possible to combine high strength with elasticity. Over a period of three to four decades rubber technologists have learned to exploit the properties of natural rubber to obtain the best combination of properties for a given use. Owing to the versatility developed in the products by this research work, rubber has become one of the important materials of construction.

In the last 20 years natural rubber has been challenged by a number of synthetic materials. This competition has been the result of two main factors—the high cost of natural rubber and the fact that in certain applications elastic or rubberlike materials were desired, but the conditions were such that natural rubber did not give satisfactory performance.

Natural rubber has varied widely in price. During the last 35 years the actual wholesale price has been as low as $3\frac{1}{2}$ cents and as high as 82 cents per lb. It is in the periods of high prices that the greatest impetus has been given to the development of rubber substitutes. Originally, the work of developing such substances was carried out along the lines of attempting to synthesize a material having the same chemical and physical constitution as natural rubber. Such materials were therefore called "synthetic rubbers." To date it has been impossible to synthesize a material identical with natural rubber, and a number of years ago the emphasis was shifted from trying to develop the same chemical and physical properties to synthesizing a material that could be used in a manner similar to natural rubber even though it did not have the same chemical and physical properties. The generic term "synthetic rubber" has been applied to this class of materials also.

*Superior numbers refer to specific references listed at the end of this subsection.

In the crude state natural rubber can be easily worked, molded, and shaped and then can be changed chemically into a strong elastic material that will resist deformation. As a result of intensive study, chemists have concluded that those materials having elastic properties of this type are composed of very long, thin molecules. The ratio of length to diameter of a fully extended molecule of this type may be as great as several thousand to one. Such molecules in the unvulcanized or uncured form of rubber are presumably greatly contracted in length and interwoven with one another. When a mass of such material is molded or shaped by being subjected to pressure or stretching, its individual molecules are displaced relative to one another. If, then, by chemical processes these long-chain molecules can be tied together, further displacement relative to one another becomes much more difficult, and the material acquires the property of resisting forces that would otherwise cause permanent deformation. However, owing to the contracted condition of the molecules, the material can still be extended. In this process the molecules are simply pulled out somewhat like a spring without causing rearrangement. In natural rubber the vulcanizing of the rubber structure is generally carried out by the addition of sulfur, although other materials can be used. The mechanism of this process is not definitely known, but it is believed that it is related to a special chemical characteristic of the natural-rubber molecule, which is termed "unsaturation." The sulfur reacts with the unsaturated portion of a molecule and forms some type of bridge with it to another molecule.

DEVELOPMENT OF SYNTHETIC RUBBER

The research on synthetic rubber has, therefore, been directed to the production of long-chain molecules which can be locked together by vulcanization or an equivalent process. In general, chemical substances that are easily available or can be prepared easily and economically are low in molecular weight. Only those capable of combining to form long chains are useful in the production of rubberlike substances. There are a large number of such substances, and in a great many cases they will combine with each other in a similar manner. Thus, it is possible to produce an almost infinite number of synthetic rubberlike materials. In general, none of these materials is identical with natural rubber, although in any one physical characteristic it is usually possible to develop a synthetic material that is superior to natural rubber. However, it is not always possible to produce a synthetic rubber that will combine several characteristics into a given product as well as it has been possible to do with natural rubber. This deficiency may be due to the rubberlike material itself, or it may be due to the fact that the rubber compounders have not had time to develop the products based on the new material as well as they have those of natural rubber. In a great many properties the synthetics are so superior to natural rubber that there is no real competition. For example, in resistance to solvents such as gasoline, resistance to acids and alkalies, permeability to gases, dielectric properties, heat resistance, vibration damping, etc., there are synthetics that are superior to the materials which can be made from natural rubber. In other cases the balance is the other way. For example, in heavy-duty truck tires, natural rubber combines the properties of elasticity, high resilience, high cut and tear resistance, and high abrasion resistance in a manner superior to synthetic materials.

PLASTICS

The foregoing discussion related to materials that have a high degree of extensibility similar to natural rubber. However, there are a great many cases in which rubber is employed that do not require this extremely high degree of elongation, and in this field a large group of materials called "plastics" can be similarly used. In general, the plastics are divided into two main classes—the thermoplastics and the thermal-setting plastics. In the thermoplastic class the materials are very similar to the synthetic rubberlike materials except that they are not in general vulcanized. Their use depends upon the fact that at high temperatures they soften sufficiently to be molded and shaped, but at lower temperatures they are able to hold this shape. Such materials vary widely in properties. They can be hard, such as the polystyrenes, polythenes, polyacrylates, and the various polyvinyl compounds. These materials can be heat-softened and then cast or molded. At ordinary temperatures they are relatively hard and can be used to replace hard rubber or ebonite in most of its uses, and they have the advantage of better physical properties in a great many cases. For example, some of them have excellent electrical properties, and some of them are transparent and can, therefore, be utilized where hard rubber is completely unsuitable. They can likewise be compounded into almost any desired color, which makes them much more suitable as materials of construction where appearance is important. While their flexibility is in general much less than that of the vulcanizable rubberlike materials, they can be compounded to give satisfactory characteristics in this respect. For example, they are flexible enough to make suitable insulation for electric wires and cables, and they can be formed in sheets which are as flexible as most woven fabrics.

The thermal-setting resins are a different class in that, after they are molded or formed, a chemical reaction is carried out which ties the whole structure together so firmly that it becomes an essentially rigid structure. The phenol-formaldehyde resins of the bakelite type are typical of this category. These resins can be molded or cast into almost any shape and have largely replaced hard rubber for this purpose. They combine high physical strength with good electrical properties and the possibilities of making products of almost any desired color. They are particularly suited for molded articles, such as electrical parts, radio cabinets, etc. They have gained wide acceptance in places where rubber was never particularly satisfactory. For example, they constitute a major portion of the film-forming materials in a wide variety of the modern paints and lacquers. They are being widely used as the bonding agent in various plywoods, and for these uses they are superior to the natural glues.

PRINCIPAL TYPES OF SYNTHETIC RUBBERS

While it is not possible to cover the extremely large number of rubberlike materials which have been produced, it is possible to review the main types and to compare their physical properties, uses, and costs with those of natural rubber. In the synthetic rubberlike materials, there are three types that are produced on a large scale, *viz.*, the buna, or diolefin, type of polymers, the neoprene type, and the butyl type. In addition, there are miscellaneous polymers such as thiokol and silicone rubber. Owing to the importance of these materials, the method of preparation and characteristics will be outlined.

Buna S (GR-S). The chief polymers of the diolefin type are those produced by the polymerization of butadiene either by itself or in conjunction with a limited amount of some other constituent. The group has been given the name "buna" rubber after the German terminology. The original buna type was made from essentially pure butadiene and was produced in Germany on a modest scale and in Russia on a very large scale. Very little of this type of material has been or is being used in the United States. The chief butadiene polymer utilized in this country is of the buna-S type, *i.e.*, it is a copolymer of butadiene and styrene. Buna S is again the German nomenclature, and the term is widely used in the United States to signify a butadiene-styrene copolymer. This type of rubber constituted the major portion of the United States government's synthetic-rubber program during the Second World War. It was given the official designation GR-S indicating government rubber of the styrene type. There are a number of different copolymers of butadiene and styrene, and the government program differentiated among them by using GR-S to indicate the whole group and adding a number to indicate the exact material, *i.e.*, GR-S-10. GR-S production capacity of over 700,000 tons (long tons) was built in the United States by the government, and an additional 30,000 tons of capacity was built by the Canadian government in Canada. The actual capacity of these plants is in excess of a million long tons per year. This capacity is larger than the annual consumption of rubber in the United States for any year prior to the Second World War. GR-S is a general-purpose rubber, *i.e.*, it can be used to replace natural rubber for most uses, and it was the chief general-purpose material used to replace natural rubber in tires and other rubber articles in the United States during the Second World War.

In the manufacture of buna-S-type rubbers (GR-S), it is necessary to produce the butadiene and the styrene. In the United States the large quantities of butadiene required during the war were produced chiefly from either alcohol or petroleum raw materials. In peacetime the cost of alcohol is such that it does not compete economically with the petroleum methods. Styrene is produced from coal tar and petroleum raw materials. Butadiene and styrene are mixed and emulsified with water by the use of soap as an emulsifying agent. They are then polymerized in emulsified form by adding a catalyst to the solution. In this manner a milky emulsion is obtained from which the unreacted butadiene and styrene can be removed. This emulsion is called the "latex" and in appearance is similar to the natural-rubber latex obtained by tapping the rubber trees.

The latex can be used directly for preparing dipped articles or for coating materials, such as cords and fabrics, or the emulsion can be broken by the addition of acids or other materials to obtain the solid rubber which can be dried and baled. Since the process involves a number of ingredients and a number of steps, it is possible by various modifications to produce polymers of different characteristics. Thus, the ratio of butadiene to styrene can be varied over a wide range, and this gives a variety of products varying from the rubberlike polybutadiene to the thermoplastic polystyrene. The general-purpose type of buna S generally contains approximately three parts by weight of butadiene to one part of styrene, although polymers containing much lower and much higher percentages of styrene are produced. Besides varying the ratio of the major constituents, it is possible to vary the type and amount of the emulsifying agent. These changes again make major differences in the property of the polymer. The operating conditions and the degree of polymerization can also be varied. Thus in the buna-S type it is possible to produce a wide variety of polymers of various characteristics, some of which are particularly suited to special uses. As a result of these variations, several hundred GR-S-type polymers have been produced in the government plants, and a number of these are produced as standard commercial grades. Several of these are available in latex form, but most of them are sold as dry rubber.

The cost of producing these materials is such that for large-scale production they can be sold at a profit for 15 to 17 cents per lb, which makes them definitely competitive with natural rubber.

Buna N. Butadiene can also be polymerized with a number of other materials besides styrene. The most widely used alternative ingredient is acrylonitrile, and the copolymer is usually classified by the German name, buna N. It is produced in a manner very similar to the buna-S polymers except for the fact that the acrylonitrile replaces the styrene. It was made to a limited extent by the United States government during the Second World War and was sold under the trade name of GR-A. However, the main production is by private plants, and in this case the material has been sold under a variety of trade names such as Perbunan, Hycar-OR, Chemigum, Butaprene, etc. Again a wide variety of different polymers can be produced, depending upon the relative quantity of the various ingredients and the operating conditions. Buna N is not employed as a general-purpose rubber, and it sells at a considerably higher price than does buna S. Its outstanding characteristic is its excellent resistance to solvents of the gasoline type. It is widely used in gasoline hoses and other elastic products that must come in contact with petroleum or other organic solvents.

Neoprene. Neoprene-type rubbers were developed by E. I. du Pont de Nemours & Company and have been produced in the United States for about fifteen years. Instead of employing butadiene, it employs a substituted diolefin called "chloroprene." The method polymerization is essentially the same as that employed for buna rubber, and the product is available in either the latex or the dry-rubber form. It is also copolymerized with other ingredients to obtain varied physical properties. Thus there are a number of types of neoprene each of which is specially suited for a different purpose. Its outstanding characteristics are resistance to solvents, resistance to aging, and resistance to burning. Because of these qualities it has been used in products requiring its solvent-resistant properties and for insulating electric cable. The cost of neoprene per pound is about twice that of the buna-S type. In addition, frequently more pounds of neoprene are required to produce a given article than with the other types of materials because of its high density.

Butyl Rubber. Butyl-type rubbers differ from the buna type and the neoprene type in that the main raw material is of the olefin type rather than the diolefin type, and this results in a product with significantly different properties. Isobutylene is the main ingredient, but in order to obtain the desired chemical reactivity, a small amount of diolefin is added. The polymerization is carried out in a nonaqueous medium at a temperature of around −100°F. The product is not obtained as an aqueous latex but is easily available as a dry rubber. It is an extremely elastic material and appears very much like natural rubber. However, owing to its structure, it has extremely low permeability to gases, and thus it has been widely used in automobile-tire inner tubes. Tubes made of butyl rubber require much less frequent addition of air than do natural-rubber inner tubes. It is likewise very resistant to chemical attack, but it has poor resistance to petroleum hydrocarbons, and its resilience is low. The low resilience is advantageous for certain damping uses, but in other cases it is undesirable. For example, it gives a high heat build-up within the rubber, and in automobile tires, for example, the high temperatures generated are detrimental. The cost of butyl rubber is roughly comparable to that of the buna-S type. It is available in large quantities as a result of the government program during the Second World War.

Thiokol. Thiokol is a trade name given to a series of organic polysulfides. The thiokol rubbers are not general-purpose rubbers but are special polymers which have solvent-resistant properties which make them extremely valuable for certain uses. Thus they are extremely resistant to solvents of the gasoline type, and they are, therefore, desirable for packings and gaskets that must operate in contact with such solvents.

Silicone. All the other synthetic rubbers are largely organic in nature, *i.e.*, the main backbone of their structure is a carbon-to-carbon chain to which hydrogen and certain other elements are attached. Silicone rubber is unique in that its basic structure is a silicon oxygen chain, and as would be expected. it has properties significantly different from the other synthetic rubbers. In general, its rubberlike characteristics, such as tensile strength, per cent elongation at break, tear resistance, abrasion resistance, etc., are decidely inferior to most synthetic rubbers, but it is able to withstand much higher temperatures than the organic materials. It has sufficient rubberlike properties to be used in gaskets, packings, and tubings, and it extends the upper operating temperature for rubberlike material by at least 100°F.

PHYSICAL PROPERTIES OF THE VARIOUS SYNTHETIC RUBBERS

The physical properties of any given synthetic rubber product depend not only on the rubberlike material itself but also upon the compounding and vulcanizing procedure which is employed. It is therefore possible to produce a fairly wide variation in properties with a given polymer, but in a great many cases the difference in the physical properties characteristic of the rubbers is so large that it is possible to give a relative comparison of their properties, and such comparisons aid in the selection of the proper material for a given use.

Tensile Strength. One of the most important properties of any rubberlike compound is its tensile strength. Such information is useful even when the product is not being used in tension, since it is an indication of the quality of the material.

Natural rubber has a high tensile strength, and in elastic products the value is of the order of 4,000 to 5,000 psi, based on the original cross section before elongation. This would correspond to 35,000 to 45,000 psi based on the cross section at break. These high tensile strengths can be obtained with natural rubber both in "pure-gum" stocks, *i.e.*, those with very little loading, and with loaded stocks containing carbon black, zinc oxide, and other materials. Butyl rubber and neoprene in general have tensile strengths slightly lower but approaching those of natural rubber. In the case of butyl rubber, these high tensile strengths can also be obtained both in pure-gum and loaded stocks. In the case of buna S, the tensile strengths are in general significantly lower than for natural rubber. Pure-gum stocks normally have tensile strengths of the order of 1,000 psi or less (based on original cross section). Loading the rubber compound with certain materials gives a marked increase in the tensile strength of buna-S compounds. With 100 parts of zinc oxide per 100 parts of rubber, it is possible to obtain tensile strength of the order of 2,000 psi, and with 50 parts of carbon black per 100 parts of rubber it is possible to obtain tensile strengths with buna S from 2,500 to 3,500 psi. Thus, in general, it is necessary to load a buna-S-type polymer with a relatively large amount of carbon black in order to obtain a product that has satisfactory physical properties. It is not a satisfactory material for producing pure-gum compounds. Buna N has tensile-strength properties similar to buna S. Thiokol and the silicone-type rubbers have low tensile strength and are not used in cases where high tensile strengths are required. In general, all the materials have satisfactory compressive strength.

Elongation at Break. In a great many cases the percentage of elongation of the rubber is an important factor in its mechanical use. This physical characteristic is very dependent upon the compound employed and the vulcanizing conditions, but in general natural rubber and butyl rubber give the highest percentage elongation at break, varying from 800 to 1,200 per cent. Buna S compounded with carbon black gives elongations at break of the order of 600 to 800 per cent, but as a pure-gum stock, this value is reduced to about 200 per cent. Thiokol and silicone rubbers have relatively low percentages of elongation in breaks.

Effect of Temperature on Tensile Strength and Elongation at Break. In general, increasing the temperature above room temperature decreases the tensile strength of all the rubber compounds, but the percentage effect varies greatly. For example, increasing the temperature from 70 to 212°F lowers the tensile strength of natural-rubber compounds from approximately 4,000 to 2,500 psi. The same temperature increase for a carbon-black-loaded buna S lowers the tensile strength from approximately 3,000 to less than 1,000 psi. Thus, while rubber at 212°F is weaker than at room temperature, it still has sufficient tensile strength to make it a desirable material of construction. The physical properties of buna S at the high temperature are becoming so poor that its desirability has greatly decreased under these conditions. With the exception of silicone, natural rubber and the other synthetic rubbers are not satisfactory for uses at temperatures above 250°F. If temperatures of this order are required for any extended period of time, silicone-type polymers have the most desirable characteristics. They will stand temperatures of 300°F for long periods of time without serious deterioration, and temperatures of 400 to 500°F can be withstood for a few hours. As indicated above, the silicone rubbers have low tensile strength, but they can be satisfactorily used as gasket material under compression at these elevated temperatures. The other rubberlike material decomposes very rapidly in this temperature region.

Density of Rubberlike Material. The density of a number of the most common rubberlike materials is given in Table 4-9. These densities are for the pure-rubber material and will be modified by the other ingredients in the compound. It will be noted that the rubbers based on hydrogen and carbon, *i.e.*, natural rubber, buna S, and butyl rubber, all have densities of between 0.9 and 0.95 g per cu cm at 25°C. Neoprene and thiokol have significantly higher densities, and this results in more pounds of these polymers being required to produce a given article than with the hydrocarbon rubbers.

Table 4-9

Material	*Density, g/cu cm at 25°C.*
Natural rubber	0.91
GR-S	0.93
Butyl rubber	0.92
Neoprene GN, CG, and E	1.23
Neoprene FR	1.14
Buna N and Perbunan	0.97-1.01
Thiokol RD	1.06
Thiokol A	1.6
Thiokol FA	1.33

Electrical Properties. The electrical properties are very dependent upon the compounding ingredients employed. The hydrocarbon-type rubbers as pure-gum material have high specific resistance (ohm per cubic centimeter), being of the order of 10^{12} to 10^{16} at room temperature. However, the addition of compounding ingredients usually lowers this resistance. The data of Yerzley[4] for a neoprene-type rubber are given in Table 4-10. Thus with acetylene-type carbon black it is possible to obtain relatively low resistance, *i.e.*, the material becomes a conducting rubber. Habgood and Waring[1] studied acetylene black in both natural rubber and neoprenes, and their results are given in Table 4-11.

Table 4-10

Type of carbon black	Parts/100 parts polychloroprene	Specific resistance, ohms/cu cm
Acetylene black	50	2.7×10^2
Acetylene black	100	1.2×10^2
Soft black	50	1.6×10^5
Soft black	100	2.0×10^7
Semireinforcing black	50	1.0×10^8
Semireinforcing black	100	6.7×10^4
Channel black	50	8.5×10^7
Channel black	100	1.7×10^6

Table 4-11

Material	Parts of acetylene black/100 parts rubber	Resistivity $\times 10^3$, ohms/cu cm	
		Through sheets	Along sheets
Natural rubber	100	0.31	0.01
Natural rubber	70	0.54	0.03
Neoprene E	75	1.08	0.13
Neoprene E	55	1.72	0.28

The dielectric constant, resistivity, and the electrical losses for several of the materials as essentially pure-gum vulcanizates are given in Table 4-12. In general, these factors can be widely modified by the particular compound employed. For example, with buna S with carbon-black loading, it is possible to increase the dielectric constant to as high as 40.

Table 4-12

Material	Dielectric constant	Resistivity, ohms/cu cm	Power factor
Natural rubber	2.7	10^{14}	
Butyl	2.5	10^{15}	2×10^{-4} (1,000 cycles/sec)
Neoprene	7.5	10^{12}	300×10^{-4} (1,000 cycles/sec)
Buna N	9.0	10^{9}	
Buna S	2.9	10^{14}	7×10^{-4} (50 cycles/sec)
Thiokol	4.6		

Low Temperature Properties. In general, all the rubberlike materials harden as the temperature is lowered. The resilience also decreases rapidly as the temperature is reduced. Natural rubber at room temperature has a very high resilience of the order of 80 to 85 per cent. Neoprene and buna S under comparable conditions have values of about sixty-five to seventy per cent, while thiokol and butyl have very low values. All the polymers show a decrease in resilience as the temperature is lowered. For example, at 40°F approximate values are natural rubber, 75 per cent; buna S, 50 per cent; buna N, 20 per cent; and butyl and thiokol have almost negligible values. At still lower temperatures the materials become very brittle and will break instead of bending or stretching. The temperature at which this phenomenon occurs is called the "freezing point" or the "brittle point." It is not a sharply defined temperature, and it varies materially with the length of time the polymers have been at the low temperature. The brittle-point temperature can also be modified by the use of plasticizers. However, general comparisons can be made between the different polymers, and Table 4-13, based on the data of Selker, Winspeare, and Kemp,[3] compares a number of the different polymers.

Silicone rubber is outstanding in its small variation in flexibility with temperature. Marsden[2] has reported that it is almost as flexible at −55° as at room temperature but that it hardens up in the region between −55 and −80°F.

Table 4-13

Compound	*Brittle point, deg C ± 0.5°*
Natural rubber:	
Gum stock	−59.5
20 parts channel black	−57.5
40 parts channel black	−56.5
100 parts zinc oxide	−60.0
Tire reclaim (vulcanized with 5.25 parts S)	−50.5
Tire-carcass reclaim (vulcanized with 5.25 parts S)	−46.5
Shoe-stock reclaim (vulcanized with 5.25 parts S)	−42.5
Polychloroprene (neoprene type):	
Milled 5 min	−36.8
Gum stock	−38.5
Gum stock + 3 parts soft black + 8 parts oil	−40.6
Gum stock + 5 parts channel black + 20 parts oil	−42.5
Gum stock + 100 parts soft black + 8 parts oil	−36.4
Gum stock + 200 parts soft black + 10 parts oil	−32.0
Butadiene polymer:	
Milled 5 min	−67.8
Gum stock	−65.5
Tire-tread stock	−71 to −69
Butadiene-styrene copolymer (buna-S type):	
Milled 2 min	−65.5
Gum stock	−65.5
Tire-tread stock	−66 to −70
Butadiene-acrylonitrile copolymer (buna-N type):	
American	−49 to −52
German	−45.5
Gum stock	−42 to −45
Tire-tread stock	−46.5
Organic polysulfide (thiokol type):	
Type I	+ 7.0
Type I, carbon-black stock	− 3.8
Type II, carbon-black stock	−22.5
Type III	−35.3
Type III, carbon-black stock	−34.0
Type IV	−35.5
Type IV, carbon-black stock	−39.5

Solvent Resistance. One of the important characteristics of a rubber product is often solvent and chemical resistance, and Table 4-14 gives some approximate values for the swelling of vulcanized pure-gum samples in various solvents. While swelling of itself might not be too objectionable, in general, the other physical properties, such as tensile strength, elongation at break, cut and tear resistance, abrasion resistance, etc., decrease rapidly when the material swells any significant amount. It will be noticed that natural rubber, buna S, and butyl rubber all swell excessively in light hydrocarbons of the gasoline type. Thiokol and buna N show very low swelling, and the value for neoprene is also low. Thus, these three polymers are good for products such as hoses, gaskets, etc., that come in contact with paraffin-type hydrocarbons. It will be noted that none of the materials is very satisfactory for resistance to aromatic materials similar to benzene. Thiokol A appears to be the best in this case. In the case of aniline, natural rubber, buna S, and butyl appear to be the most satisfactory. Thus the relative solvent resistance depends largely on the solvent encountered, and for each solvent it is necessary to have actual data in order to be certain which material is most suitable.

Table 4-14. Swelling of Vulcanized Elastomers in Various Liquids
(Volume per cent increase)

Liquid	Natural rubber	Neoprene	Buna S	Butyl	Buna N	Thiokol
Light gasoline	160	...	60	. .	4	0
Gasoline	230	8	80	200	5	0
Kerosene	...	60	...	...	...	0
Diesel oil	120	...	...	...	15	
Fuel oil	...	...	...	...	...	0
Lubricating oil	...	40	..	...	0	0
Paraffin oil	140	..	...	...	3	
Transformer oil	150	...	...	...	5	
Acetone	..	25	...	...	110	
Benzene	370	160	180	200	210	7
Carbon tetrachloride	670	160	320	...	120	0
Ethyl ether	130	50	50	. .	50	
Linseed oil	100	. .	. .	...	20	
Turpentine	300	90	..	.	50	0
Aniline	30	100	20	3	300	

Summarized Properties. As a general basis of comparison, the properties of the different polymers are summarized in Table 4-15.

Table 4-15

Property	Natural rubber	Buna S	Neoprene	Butyl	Buna N	Thiokol	Silicone
Tensile strength	E	G	E	E	G	F	P
Elongation at break	E	G	E	E	G	F	P
Abrasion resistance	E	E	E	E	E	G	P
Tear resistance	E	F	G	E	G	P	P
Electrical properties:							
Conductivity	F	F	F	G	F	F	F
Resistance to corona cracking	P	P	E	E	P	E	E
Dielectric strength	E	E	F	E	G	F	E
Chemical resistance:							
Oxidizing solutions	P	P	P	G	P	P	G
Ozone	P	F	E	E	F	E	E
Solutions of salts, alkalies, acids	G	G	G	E	G	G	P
Adhesion to metals	E	E	E	G	E	F	
Resistance to cutting	G	F	G	E	F	P	
Rebound	E	G	G	P	G	P	
Resistance to diffusion of gases	F	F	G	E	F	E	
Resistance to flex cracking	G	G	G	E	G	F	
Resistance to flow:							
Cold	E	E	G	F	E	P	
Hot	E	E	F	F	E	P	
Low heat generation by hysteresis	E	F	F	P	F	F	
Resistance to swelling:							
Aromatic solvents	P	P	P	P	P	G	
Lacquer solvents	P	P	P	P	P	G	
Gasoline	P	P	G	P	E	E	
Water	F	F	G	G	E	F	
Resistance to deterioration in mineral oil	P	P	E	P	E	E	
Specific gravity	0.91	0.93	1.20	0.92	0.98	1.2–1.6	
Resistance to deterioration by sunlight	F	G	E	E	G	E	

Key: E = excellent; G = good; F = fair, P = poor.

References

1. Habgood and Waring, *Trans. Inst. Rubber Ind.*, vol. 17, p. 50, 1941.
2. Marsden, *Rubber Age*, vol. 59, p. 691, 1946.
3. Selker, Winspeare, and Kemp, *Ind. Eng. Chem.*, vol. 34, p. 157, 1942.
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SECTION 5

INDUSTRIAL-PLANT FIRE PREVENTION AND PROTECTION: COMBUROLOGY

BY GILBERT E. STECHER

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INDUSTRIAL-PLANT FIRE PREVENTION AND PROTECTION: COMBUROLOGY

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An industrial plant may consist of a single building ranging from one story to several stories in height, or it may consist of a group of buildings of varying heights and area and include a diversification of construction and the necessary outdoor storage tanks and mechanical equipment. Regardless of its size or the product it manufactures, the establishment represents the investment of intelligent planning, time, and money, and as such it becomes part of the economic life of the nation.

The owners of the plant have created it for the purpose of profit, for a reasonable return on their investment, and because of a desire to be of service in the nation's economy. In so doing, they have assumed responsibilities to the city or town in which they have elected to do business. Taxes constitute one of these obligations. The payment for services such as water and electrical power, the employment of labor, the safeguarding of the lives of employees, and the maintenance of the plant and its premises to ensure the continuity of these obligations are other responsibilities. Within their own organization the owners may be responsible to invested capital, stockholders and bondholders, and to many liabilities which depend upon the successful and continuous operation of the plant.

The capital investment, profits, and pay rolls are subject to no greater jeopardy than that of fire which may occur from inherent possibilities in the process, lack of maintenance, or sheer carelessness.

COST ACCOUNTING

Destruction by fire becomes a factor in the economics of manufacture, in no less degree than any other item, because it involves the entire project and assumes many phases. As an item in plant management, the subject of fire should be set up as a major one, for under this title will come a factor in cost accounting that must be considered part of the price of the finished product. This will include the fire insurance and the attending coverages, such as windstorm, hail, explosion, riot, civil commotion, smoke, and automobile damage, business interruption, or use and occupancy, rent and leasehold, all of which are based as to cost upon the rate charged for the fire insurance.

The fire-insurance rate is based primarily on the burning rate of an individual classification, such as woodworking, metalworking, or chemical manufacture. This basic charge is part of the cost accounting of the insurance business and affects the cost to industrial plants only as the burning rate of the classification as a whole. If the fire loss in a classification is lowered or raised over a given period, this base charge is altered accordingly. The fire-insurance rate is set up by applying to the plant a scientific schedule made up of charges and credits plus the basic charge and attempting to measure thereby the inherent and maintenance hazards of the processes and equipment and the defects of construction as applied to the spread and control of fire. These charges and credits are of vital interest to plant management, not only as to fire-prevention measures but as a direct cost against the manufacture of the product.

Fire-insurance schedules apply basic principles in different ways to secure an end result. In order to secure the facts applying to the particular risk or plant in a particular section, it is necessary to apply to the rating organization having jurisdiction.

This is generally done by the insurance agent or broker who handles the insurance placements of the company. The schedules applying to the several jurisdictions are in the East, the Universal schedule; in the Middle West, the Analytic system; and in the Far West, the Factoral system.

The fire-insurance rate is an overhead charge against the product and is competitive in that a low rate, aside from being indicative of low hazard, will reflect a lower overhead charge against the product. This may appear small percentagewise in a moderate-sized plant, but where large values are involved, the fire-insurance rate will be a factor.

Two plants, making a similar product, with values of buildings, equipment, and stock amounting to $500,000, started in business about the same time. One plant, having built all buildings of brick, or occupying brick or fire-resistive buildings, with no exposed steelwork and all horizontal and vertical communications conforming to standard practice, will have a rate of insurance ranging from 10 to 75 cents per $100. The other plant, occupying a conglomerate group of ordinary brick and frame buildings, with steelwork unprotected and communications disregarded, will have an insurance rate ranging from $1 to $2 per $100. This will mean a maximum difference in overhead cost of between $3,750 and $10,000.

Other factors, which apply to both plants, are the location and safeguarding of processes of a hazardous nature. The effect on cost accounting of these features is the hidden charge for a nonstandard arrangement of the process and protective features.

Plant management must be on the alert to avoid placing hazardous processes where they will jeopardize large values, as this affects the insurance rates adversely.

In considering plant extensions, new construction and occupancy should not be placed where they will be exposed to hazardous processes, for the insurance rate will reflect this exposure.

Cost-accounting methods may differ in the application of the insurance rate or premium to manufacturing costs. These methods should be analyzed to ascertain whether the unit or over-all basis is used. Insurance rates are made on a specific basis for each building and its contents and include a charge for exposure of one building to another, this being computed from buildings in the same group or plant and also from buildings unrelated to the plant. This exposure is measured by type of construction and distance removed, plus a percentage reduction for the greater distance.

In considering any additions or new construction or the installation or shifting of processes or hazardous units, therefore, the insurance rate of the building and contents and the exposure charge should be analyzed as they may have an effect upon the cost accounting for that unit.

A spray booth, located in a building where the operation is nominal and the contents value is low, may be shifted to a building where the operation is on a larger scale and the values high. The insurance rate in the latter case is increased on the high values, and the exposure charge affects all surrounding buildings.

Also, in building new additions, the uninterrupted area may be increased and the insurance rate increased since there is a charge for area over a standard minimum. The erection of standard fire walls may be required to avoid this charge. New buildings, regardless of the occupancy, if placed within a given distance of another building, will be subject to an exposure charge as well as creating one for another building. Fifty feet is the minimum distance to avoid this charge, and height limits range from one to four stories; ground-floor areas range from 5,000 to 10,000 sq ft minimum. Exact details cannot be given as these factors vary in different schedules and jurisdictions.

The above details will indicate the principle involved and the general application of the insurance schedule which affects this feature of plant management.

For cost accounting, the following factors should be taken into consideration:

1. Raw stock—Valuable and hard-to-procure raw stocks in fire-resistive segregated warehouses reduce the possibility of replacement at higher costs and also plant shutdown.

2. Finished stock—Similarly treated finished stock reduces the possibility of loss of market, overtime production, and loss of profits.

3. Processes—Processes that are safeguarded in a standard manner against possible shutdown or disruption reduce the cost of the product, as there is no lost time, and guarantee deliveries of finished product.

4. Housing—All housing in fire-resistive buildings reduces maintenance, insurance costs, and rebuilding costs.

Fire-insurance Rates. The fire-insurance rate is a reflection of the features that are involved in plant construction and occupancy and also includes the elements that are responsible for the causes and spread of fire. The following details, taken from the schedule as applied to an eastern area because of the greater familiarity with that area, will cover the necessary factors to be considered by plant management.

In some of the schedules used in other jurisdictions, the charges and credits are by percentages, and the approach to the theory of rating is focused upon the occupancy. The standards used for building construction, however, are generally based on the National Board of Fire Underwriters Building Code, and all other features are based on the Standards of the National Fire Protection Association, Mutual Laboratories of Boston, and Underwriters' Laboratories of Chicago. All these standards are available to the plant manager, free or for a nominal fee. The application of the schedule at the onset assumes the following conditions to be present:

1. Building is to be well built, in good repair, and clean.
2. Exposures must not affect the plant.
3. Environment must be good, *i.e.*, free from all objectionable features, conflagration, and railroad trackage.
4. Public fire protection must be class A. This is the classification given under the application of the National Board of Fire Underwriters Grading Schedules for Towns and Cities.
5. Occupancy must be of single type—no other tenants.

The standards shown in Table 5-1 and in the material following are required minimums under the schedule, and charges are made for deflection from them.*

Table 5-1

Height of building stories	Minimum thickness for bearing walls of brick and stone or reinforced concrete, in.															
	Stories															
	1		2		3		4		5		6		7		8	
	Bk.	Re-conc	Bk	Re-conc	Bk	Re-conc	Bk	Re-con	Bk	Re-conc	Bk.	Re-conc	Bk	Re-conc	Bk	Re-conc
One	12	9														
Two	12	10½	12	9												
Three	16	10½	12	10½	12	9										
Four	16	12	16	10½	12	10½	12	9								
Five	20	12	16	12	16	10½	12	10½	12	9						
Six	20	13½	20	12	16	12	16	10½	12	10½	12	9				
Seven	24	13½	20	13½	20	12	16	12	16	10½	12	10½	12	9		
Eight	24	15	24	13½	20	13½	20	12	16	12	16	10½	12	10½	12	9

Standard Requirements for Fire-insurance Rate

(Standards prescribed by the National Board of Fire Underwriters are to be used.)

Walls. *Bearing Walls.* To be brick, stone, or reinforced concrete.

Nonbearing Walls. May be 4 in. less in thickness than bearing walls, but no such wall shall be less than 12 in. in thickness for the uppermost four stories in height. Nonbearing walls which are supported by girders, at each story, and pilastered walls of one-story buildings may be 8 in. thick.

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* Data courtesy of Fire Insurance Rating Organization, State of New Jersey

Party or Fire Walls. To be same thickness as required for bearing walls.

Parapet Walls. Where required, shall be not less than 12 in. thick and carry not less than 3 ft above the roof.

Height. Not over four stories and basement. Basement 3 ft above ground level to be considered a story.

Metal Columns. Iron and steel columns, when placed within exterior walls of a building, shall be protected by approved masonry not less than 4 in. thick, but the interior faces may be protected as provided for interior iron or steel columns as follows: 3¾ in. brickwork; 2 in. concrete, solid gypsum block, or hollow clay tile (if single shell, it shall be plastered with gypsum or cement mortar at least ½ in. thick); 3 in. hollow gypsum block.

Metal Beams and Girders. Iron or steel beams and girders, supporting walls, including lintels (except spans 6 ft or less having masonry or other suitable arch support above the lintel), shall be protected by not less than 2 in. of approved masonry. When not supporting walls, iron or steel beams and girders shall be protected by not less than 1½ in. of brickwork, concrete, hollow clay tile, gypsum, cement, or cement mortar on expanded metal lath. Roof trusses, unless protected as above, may be protected on the sides and bottom by not less than 1 in. of cement or gypsum mortar on expanded metal lath.

Floors. Ordinary wood joist with double floor boards in good condition.

Interior Finish. Open, walls and ceilings.

Roof. Flat, covered with metal, slate, or approved composition.

Skylight. Metal frames, ¼-in. wired glass, or heavy glass not less than ¼ in. in thickness, protected with wire screens of not exceeding 1-in. mesh and not less than No. 12 gauge.

Exterior Attachments. Cornices, porches, bridges, monitors, and saw-tooth-type roof structures and similar attachments shall be constructed of incombustible material.

Floor Openings. *Stairways.* Enclosed with 2-in. tongued and grooved planking, with automatic closing door of same thickness at openings, or stairway enclosed in single board if covered with lock-jointed tin on the side toward the room, with automatic closing door of same construction at openings, or open stairs with automatic closing traps at floor openings of same material and thickness as floor, covered on lower side and lapping edges with lock-jointed tin.

Elevators. In brick shaft with standard cutoff or open with automatic closing traps at every floor opening, of same material and thickness as floor, covered on lower side and edges with lock-jointed tin.

Area. Ground area to be not over 5,000 sq ft in brick and 2,500 sq ft in frame buildings.

Heating (except for mechanical purposes). Steam or hot water safely arranged and well set with steampipes on metal support, free from woodwork.

Table 5-2

	Class of fire			Protection from freezing required
	A	B	C	
Water pails, 12-qt type	5			Yes
Sand pails		5		No
2½-gal soda-acid extinguishers	1			Yes
1¼- or 1½-gal soda-acid extinguishers	2			Yes
1¼- or 1½-gal foam extinguishers	2	2		Yes
2½- or 5-gal foam extinguishers	1	1		Yes
2½-gal calcium chloride	1			No
2½-gal pump tank	2			Yes
5-gal pump tank	1			Yes
22-gal bucket tank with 5 standard pails	1			Yes
50-gal cask with 3 standard pails	1			Yes
1-gal loaded stream	2			No
1¾- or 2½-gal loaded stream	1	2		No
1-, 1½- or 2-qt carbon tetrachloride extinguisher		2	2	No
1-gal carbon tetrachloride extinguisher		2	2	No
2-or 3-gal carbon tetrachloride extinguisher		2	1	No
4-lb carbon dioxide cylinder		2	2	No
7½- or 10-lb carbon dioxide cylinder		2	1	No
15- or 20-lb carbon dioxide cylinder		1	1	No
2½-gal plain water extinguisher	1			Yes

Chimneys. Brick, built up solid from foundation extending at least 18 in. above the roof and at least 8 in. in thickness or 4 in. in thickness with terra-cotta flue lining.

Lighting. Incandescent electric wiring in accordance with National Electric Code or gas from public plant, well arranged.

Power. Steam boiler in outside building with standard cutoff.

Inside Standpipe and Hose Protection. To be in accordance with the general standards.

Inside First-aid Fire Appliances. For all practical purposes there are three general classes of incipient fires:

Class A—Incipient fires on which quenching and cooling effect of water is of first importance, as ordinary combustible materials, such as wood or excelsior.

Class B—Incipient fires on which the blanketing or smothering effect of the extinguishing agent is of first importance: *i.e.*, gasoline, oils, greases, etc.

Class C—Fires in electrical equipment, where use of nonconducting extinguishing liquid is of great importance.

A unit has been adopted for convenience in measuring the fire protection afforded by first-aid fire appliances. This unit is composed of from one to five first-aid fire appliances, depending upon the type used. One unit is required for each 2,500 sq ft of floor area or not over 50 ft of travel from any point to reach the nearest unit.

The classes of fire, the suitable appliances, and the number required for each 2,500 sq ft of floor area are set forth in Table 5-2 (see list of Underwriters' Laboratories, Inc., for approved devices).

Table 5-3 will indicate the penalty for infraction of the standard from which computations can be made for differences in cost of new installation or for existing violations.

Table 5-3

	Cents
Walls	
a. Brick, stone, or reinforced concrete, deficient in thickness, each side and end wall	0.01
b. For cement block, terra-cotta, tile, corrugated iron on steel frame, or other noncombustible material not specifically mentioned, each side and end wall	0.04
Note 1. *If satisfactory evidence is submitted that the construction is of cinder or concrete block, listed by Underwriters' Laboratories, make one-half charge.*	
Note 2. *Nonbearing outside curtain walls, make one-half charge.*	
c. For absence of parapet on each party or exposed masonry wall	0.03
d. Parapets not in accordance with standard, each party or exposed wall	0.02
Note 1. *Charges under c or d do not apply if wall and roof constructed of incombustible material, or where adjoining or exposing building is higher than risk being rated.*	
Note 2. *Reduce charge under c and d one-fourth for each 5 ft distance removed from exposing building.*	
e. Frame, brick-veneer, corrugated iron on wood sheathing or skeleton wood frame, or other combustible material. If main building, additions, or extensions constructed as above, charge:	
If not more than 25 per cent of wall area	0.08
26 to 50 %	0.16
51 to 75 %	0.24
Over 75 %	0.32
Height	
a. Buildings of other than *e* (above) construction, each story over four	0.05
b. Buildings 50 % or over of *e* (above) construction, each story over two	0.05
Columns, Girders, and Beams	
a. For unprotected metal columns, each floor	0.03
b. For unprotected metal girders or beams, each floor	0.01
Note. *No charge to be made when building is 50 per cent or over of e (above) construction.*	
Floors	
a. If not according to standard, each floor	0.02
Note. *If no basement, make no charge for first floor.*	
Interior Finish (any finish forming concealed spaces, each floor)	
a. Walls	0.01
b. Ceilings	0.01
Note. *Make no charge if building is 50 per cent or over of e (above) construction.*	
Roof	
a. Wood shingle or board	0.10
b. Combustible mansard, monitor, or saw-tooth-type roofs (cumulative with Skylights below)	0.05
c. Roof space or blind attic	0.03
Note. *No charge under item b if building is 50 per cent or over of e (above) construction.*	
Skylights (if not in accordance with standard)	
a. For 16 sq ft or less	0.05
b. If over 16 sq ft (not cumulative with *a*)	0.10

	Cents
Note 1. *Item a or b cumulative with b under Roofs.*	
Note 2. *If wood frame protected with standard screens, reduce charge one-half.*	
Exterior Attachments	
a. Wood-boxed cornice, porches, awnings, bridges, or similar combustible attachments	0.01
Note. *No charge if building is 50 per cent or over of e (above) construction.*	
Floor Openings (any kind)	
a. Each open floor opening, each floor pierced	0.03
b. Each floor opening enclosed, but not standard, each floor pierced	0.02
Note 1. *Where charge is made for multiple floor openings, use full amount of highest charge, one-half of next lower, and one-quarter of each additional.*	
Note 2. *Floor openings within 15 ft of other floor openings, take one (highest) charge.*	
Area	
a. Each 1,000 sq ft (or fraction thereof in excess of 500 sq ft) in excess of 5,000 sq ft of ground area in building of other than *b* or *e* (under Walls) construction	0.02
b. Each 1,000 sq ft (or fraction thereof in excess of 250 sq ft) in excess of 3,750 sq ft of ground area in building of *b* (under Walls) construction	0.03
c. Each 1,000 sq ft (or fraction thereof in excess of 250 sq ft) in excess of 2,500 sq ft of ground area in building of *e* (under Walls) construction	0.04
Note. *For buildings of mixed construction, charge for area on the basis of the direct percentage that each type of construction bears to the total floor area.*	
d. If area subdivided on all floors by division or cross walls of brick, stone, or concrete, built from ground, strengthening the building (even if with openings), deduct 10 per cent of area charge for each wall, not to exceed 40 per cent in all.	
Note. *Total charge not to exceed* 0.40.	
Heating (except for mechanical purposes)	
a. High-pressure steam boilers, well arranged	0.10
Note. *Not cumulative with charge for high-pressure steam boiler (see Power below).*	
b. Stoves or other unapproved unit heaters	0.05
c. Gasoline stoves.	
Chimneys (not in accordance with standard)	0.05
Lighting	
a. Not in accordance with standard	0.05
b. Acetylene and gasoline systems.	
Power (for manufacturing purposes)	
a. High-pressure steam boilers, well arranged	0.10
b. Other approved methods, well arranged	0.05
c. Metal boiler stack, through roof, properly arranged	0.10

Table 5-4, Credits for Construction, which is better than anticipated under the schedule, will indicate the possible saving in costs, if at present existing or contemplated. These are granted on rate at this point.

Table 5-4. Credits for Construction

	Credits, per cent
a. Better than standard, other than mentioned below	3
b. Fire resistive throughout, except roof	40
Note 1. *Not to apply to buildings one story in height without basement. If with basement, reduce credit one-half.*	
Note 2. *For floors of mixed construction prorate the credit on the basis of the direct percentage that the floor area of fire-resistive construction (disregarding the first floor of buildings without basement, and any floor, the total area of which is 25 per cent, or less, of fire-resistive construction) bears to the total floor area of the building.*	
c. Standard mill throughout, including roof:	
Height, stories	
4 (or higher)	25
3	20
2	15
1	10
d. Semimill throughout:	
Height, stories	
4 (or higher)	10
3	8
2	6
1	4
e. Height, for each story less than four in masonry buildings	2
Note. *Not to apply to boilerhouses.*	

Installation of protective features such as alarm systems, watchmen, standpipe systems, automatic sprinklers in basements, large-unit fire extinguishers, private fire brigade, and smokeproof tower (aid to fire department and protection to life) will indicate savings for present installations or future possible savings, by the credits allowed in Table 5-5.

Table 5-5

	Credits, per cent
a. Automatic fire alarm, central station (deduction not to exceed 30 cents)	25
b. Automatic fire alarm, auxiliary (deduction not to exceed 25 cents)	22
c. Manual fire alarm, central station (deduction not to exceed 16 cents)	9
d. Manual fire alarm, auxiliary (deduction not to exceed 15 cents)	8
e. Watchman and clock (deduction not to exceed 10 cents)	5
f. Watchman, central station ordinary supervision (deduction not to exceed 10 cents)	6
g. Watchman, central station compulsory tour supervision (deduction not to exceed 20 cents)	11
NOTE 1. *Reduce credit under c or d 50 per cent if no approved watchman service unless risk is under 24-hr operation daily.*	
NOTE 2. *Reduce credit under d to 6 per cent if system not approved by Underwriters' Laboratories.*	
NOTE 3. *Reduce credit under a, b, c, d, f, and g 50 per cent if risk is not located within 20 miles of headquarters furnishing approved continuous supervisory service.*	
NOTE 4. *All systems should be approved by Underwriters' Laboratories and installed, maintained, and supervised in accordance with the rules and requirements of the National Board of Fire Underwriters and Rating office.*	
NOTE 5. *When credit is allowed under a or b, one-half of credits under e, f, or g may be cumulative.*	
NOTE 6. *When credit is allowed under e, f, or g, full credit under c, or d may be cumulative.*	
h. Standpipe and hose in accordance with standard requirements:	
Class A	2½
Class B	5
Class C	7½
Class D	10
i. Automotive sprinkler system in basement (approved)	5
j. For approved 40-gal portable chemical extinguisher, installed in accordance with the requirements (deduction not to exceed 5 cents)	2½
k. For approved private fire department with standard outside fire hydrant and hose and drilled fire brigade (deduction not to exceed 10 cents)	5
NOTE. *Credit under k to apply only in class H or K localities, which are without public fire protection.*	
l. Smokeproof tower to aid fire department	1

The following charges are for multiple occupancy or additional tenants who may share a building with the original tenant:

Table 5-6

	Charge, cents
a. For each additional tenant or occupant, in buildings over 50 per cent of other than *e* (under Walls) construction:	
Manufacturing	0.07
Mercantile, storage, or vacant	0.03
b. In buildings 50 per cent or over of *e* (under Walls) construction:	
Manufacturing	0.12
Mercantile, storage or vacant	0.10
NOTE 1. *Make full charge as provided above for first two tenants or occupants and reduce charge 1 cent for each additional tenant or occupant over two.*	
NOTE 2. *For each additional family over 10 in building being rated, 0.003. Total charge not to exceed 25 cents.*	
NOTE 3. *Make no charge for barbershops, beauty parlors, etc., above the grade floor.*	

For grades of classification of public fire protection lower than class A, a charge is made generally (see Table 5-7).

A further credit is allowed, under this heading, for fireboat and standard h-p systems as shown in Table 5-8.

Charges are made, as in Table 5-9, for unsafe and unusual conditions which may exist or are permitted to come into existence and are really penalties for neglect or disregard of causes that may result in a fire. They are corrective and removable and can be complied with, thereby securing a saving and eliminating a fire hazard.

Credit for coinsurance will vary in different jurisdictions, and Table 5-10 is given as an indication of such credit on a particular schedule applied to this particular subject (see Table 5-7).

Table 5-7. Public Fire Protection

Town class	*Charge, cents*
A	0.00
B	0.01
C	0.02
D	0.03
E	0.04
F	0.08
G	0.20
H	0.45
K	0.50

Table 5-8. Deductions for Superior Public Fire Protection

	Deductions, per cent
a. Fireboat. Risk so located that city-owned fireboat can be of active and beneficial service NOTE. *Deduction not to exceed* 15 *cents.*	5
b. If risk is within 500 ft (over public thoroughfares) of hydrant on standard h-p system	10
c. As in *b*, but substandard system	5

Table 5-9

For conditions of building and occupancy, add as follows:

	Charges
Stoves or Heaters	
a. Poorly arranged	0.10
b. Absence of metal under	0.10
c. Hazardous arrangement	0.25
Stovepipes and Flues	
a. Combustible material less than 18 in. away, or less than 9 in. away with standard protection	0.10
b. Hazardous installation	0.25
c. Entering bottom of chimney vertically	0.15
d. Entering chimney in attic or any unused room	0.25
Terra-cotta chimneys	0.15
Defective chimneys	0.15
Defective steam-boiler installation	0.25
Lighting	
a. Swinging and otherwise unsafe gas brackets or lamps	0.25
b. Large center-draft oil lamps	0.10
Defective electrical installation	0.25
Empty boxes, rubbish, and odd accumulations adjacent to building	0.25
Hazardous packing materials not properly disposed of	0.25
Odd accumulations, crowded and dirty conditions	0.25
Crowded condition of merchandise preventing proper access to building	0.15
Obstructions to building or roof that may interfere with fire department	0.10
Paper, burlap, or miscellaneous combustible materials on walls or ceiling	0.10
Building not in good state of repair	0.10
Building dilapidated or in poor state of repair	1.00
Gasoline, Naphtha, Etc.	
a. Storage and use of gasoline and gasoline devices, as per general rules.	
b. Absence of safety cans	0.10
Hazardous manufacturing appliances or devices not properly installed or maintained	0.25
Absence of approved metal ash, rubbish, or waste cans	0.10
Absence of No Smoking signs	0.10
Absence of adequate first-aid fire appliances	0.10
Condition not anticipated under schedule	0.25

In class H and K towns, when coinsurance credit is not given for superior construction or approved private fire protection, reduce rates without coinsurance 5 per cent (not to exceed 15 cents) for use of 80 percent coinsurance clause. Coinsurance deserves careful consideration, for herein lies the difference in insurance costs between good and bad construction and proper safeguarding of the plant premises.

The application of coinsurance to the total amount of capital assets in the insurance program will affect the amount of recovery from any fire loss.

The insurance rate is based upon the assumption that the values subject to destruction by fire will be insured to a given percentage of those values. This percentage is placed most generally at 80 per cent of the values. As the coinsurance clause in the insurance

Table 5-10. Deductions for Coinsurance Clause for Specific Form of Insurance

	Deductions, per cent
The final rates for specific form of insurance are subject to following reductions for use of 80 per cent coinsurance clause, but not to exceed 50 cents, except in class A and B towns:*	
In all towns, except H and K*	25
The following deductions apply, regardless of location:	
a. When credit is given for fire-resistive construction throughout, except roof	40
Note. *When credit is given under b in Table 5-4, prorate the coinsurance reduction on the basis of the same percentage as is allowed at b, disregarding any percentage less than 20 per cent.*	
b. When credit is given for approved private fire protection	15
c. Buildings not over one story (no basement) with incombustible walls, partitions, staging, floors, and roof, contents incombustible, no environment and no exposures, shall be subject to reduction of 75 per cent for attachment of the 80 per cent coinsurance clause.	

* The reference to class A and class B, H, and K towns, is the rating given for the character of fire protection or the lack of it entirely.

policy becomes part of the contractural agreement, if the agreement to carry 80 per cent of the value is carried out, the entire amount of the loss is paid whether it is partial or total.

If the percentage of value insured, however, is less than the agreed-upon 80 per cent, the settlement of the loss is made on the basis of the ratio the amount of insurance carried bears to the required 80 per cent of the total value.

$500,000 = capital asset building, equipment, and stock
400,000 = required 80 per cent, to be carried under the coinsurance agreement
300,000 = amount of insurance carried, with the coinsurance clause as part of the contract of insurance

$$\text{Amount recoverable} = \text{fire loss} = \$50{,}000$$

$$\frac{300{,}000}{400{,}000} = 75 \text{ per cent} = \$37{,}500$$

Contents Rate. The contents rate is made by including charges for types and classes of manufacture or nonmanufacture and for hazardous processes or operations which are subject to scientific and standard application treatment. A list of these standards is appended, and they can be secured from the National Fire Protection Association, the National Board of Fire Underwriters, or Underwriters' Laboratories.

A credit is also allowed in the contents item if merchandise is on skids or platforms 6 in. in height. This credit applies only to stocks particularly damageable by water, contained in other than barrels, bottles, and waterproof cases. An additional credit is allowed if all shelving is offset from wall 1 in. and backed up with approved wire screening. These credits will generally amount to 2 per cent of the flat building rate, plus the charge for susceptibility at this point on the schedule. The above charges are numerous and changeable and so must be secured from the rating bureau having jurisdiction.

The credits allowed for any and all fire-fighting equipment —first-aid equipment, standpipe systems and yard systems, automatic sprinkler systems, and alarm systems—although part of the insurance rate, must be sought for this purpose from a cost-accounting standpoint, the same as all other credits. Those fire-fighting units coming under the heading of fire defense will be treated fully under Fire Protection.

Plant management, cooperating with the cost-accounting department and considering the details of construction, hazardous processes, and fire defense, will secure a low insurance rate which will reflect in the cost of the product and also reduce the possibility of destruction by fire of the capital assets of the plant.

Fire-resistive-construction Rates. In some jurisdictions fire-resistive-construction rates are made on special schedules. Table 5-11 gives the acceptable standards in an eastern area. The charges on the schedule are made for deviation from these standards and include the usual deductions for fire protection, alarm systems, and the charges for faults of management (see Table 5-9). Coinsurance credits, for use of (90/80 per cent) coinsurance clause, will produce reductions up to 75 per cent for superior types of construction.

Standard Requirements for Fire-resistive Construction[*]

Bearing Walls. *Brick.* Shall be not less than 12 in. in thickness for the uppermost three stories in height and shall increase 4 in. in thickness for each successive two stories in height, measured downward from the top of the wall.

Reinforced Concrete. Shall be not less than 6 in. in thickness for the uppermost story in height and increase 1 in. in thickness for each successive two stories in height, measured downward from the top of the wall.

Nonbearing Walls. *Brick.* May be 4 in. less in thickness than bearing walls, but no such wall shall be less than 12 in. in thickness for the uppermost four stories in height. Nonbearing walls, which are supported by girders at each story, and pilastered walls of one-story buildings may be 8 in. thick.

Reinforced Concrete. If supported by girders at each story, such walls may be, but shall not be less than, 5 in. in thickness.

Fire Walls. *Brick.* To be same thickness as required for bearing walls, except business and storage buildings exceeding two stories in height, no part of which wall shall be less than 16 in. in thickness.

Reinforced Concrete. (Public Buildings.) To be not less than 9 in. in thickness for the uppermost three stories in height and increase 3 in. in thickness for each successive three stories in height, measured downward from the top of the wall.

(Business Buildings.) To be not less than 11 in. in thickness for the uppermost two stories in height and increase 2 in. in thickness for each successive two stories in height, measured downward from the top of the wall.

Party Walls. *Brick.* To be same thickness as required for fire walls. Except when supported by girders at each story, such wall may be not less than 12 in. in thickness.

Reinforced Concrete. To be same thickness as required for fire walls. Except when supported by girders at each story, such wall may be not less than 8 in. in thickness.

Roof. Roof construction shall consist of a series of steel or reinforced-concrete beams with interposed arches or slabs of such material and construction as shall meet the requirements of the Building Code of the National Board of Fire Underwriters. Roof coverings shall be of approved incombustible materials.

Skylights. To be constructed of wired glass in metal frames, or plain glass protected by wire screen placed 4 to 10 in. above and extending on all sides the same distance beyond the glazing.

Height. Not over six stories and basement.

Floors. Floor construction shall consist of a series of steel or reinforced-concrete beams with interposed arches or slabs of such material and construction as shall meet the requirements of the Building Code of the National Board of Fire Underwriters.

Metal Columns. Iron and steel columns when placed within exterior walls of a building shall be protected by approved masonry not less than 4 in. thick but the interior faces may be protected as provided for interior iron or steel columns as follows: $3\frac{3}{4}$ in. brickwork; 2 in. concrete, solid gypsum block; 4 in. hollow clay tile (if single shell, it shall be plastered with gypsum or cement mortar at least $\frac{1}{2}$ in. thick); 3 in. hollow gypsum block.

Metal Beams and Girders. Iron or steel beams and girders, supporting walls, including lintels (except spans 4 ft or less having masonry or other suitable arch support above the lintel) shall be protected by not less than 2 in. of approved masonry.

When not supporting walls, iron or steel beams and girders shall be individually protected by not less than $1\frac{1}{2}$ in. of brickwork, concrete, hollow clay tile, gypsum, cement, or cement mortar on expanded metal lath.

Partitions. To be constructed of noncombustible material.

Area. Not over 10,000 sq ft ground area between intersecting fire walls.

Floor Openings. Stairways, elevators, and other shafts. When walls enclosing shafts are bearing and of masonry or reinforced concrete, they shall be of the same thickness as specified for bearing walls. When they are supported at each floor, they may be constructed of 4-in. brick or 6-in. hollow tile plastered on both sides or 4-in. cored gypsum block. Openings in such shaft enclosures shall be protected by approved automatic or self-closing fire doors.

Heating. (For nonmanufacturing purposes) Steam or hot water safely arranged and well set with steam pipes on metal supports, free from woodwork.

Chimneys. Brick, built up solid from foundation at least 8 in. in thickness for medium-heat appliances (h-p steam), or 4 in. in thickness with terra-cotta flue lining for low-heat appliances (l-p steam).

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[*] See National Building Code, National Board of Fire Underwriters for complete details and time-temperature curves.

Lighting. Incandescent electric wiring in accordance with National Electrical Code, or gas from public plant, well arranged.

Power. Steam boiler in outside building with standard cutoff.

Automatic Sprinkler Protection. The rating of plants equipped with automatic sprinkler systems is based primarily upon the grading of the sprinkler protection estimated on the hazard of the occupancy. A system that does not grade 40 per cent or better is not considered eligible for a sprinkler rating but is put back on the schedule applying to the particular plant on a nonsprinklered basis, with credits that may apply for protection.

Credits and Charges. The schedule for grading automatic-sprinkler protection is a detailed affair and must be applied by the bureau having jurisdiction and may differ in its application in various parts of the country. The following fundamental details, however, as applied in an eastern jurisdiction, will cover the essentials, and the standards will be approximately the same for all installations. Reference is made to the "Standards for Sprinkler Equipment, Automatic and Open Systems," published by the National Board of Fire Underwriters.

SCHEDULE FOR GRADING AUTOMATIC SPRINKLER PROTECTION

1. Full standard equipment.

Water Pressure

2. *Primary pressure deficient* from gravity tank, giving less than 15 lb on highest line of sprinklers, but with tank bottom not less than 5 ft above the highest line of sprinklers (1 per cent deduction for each foot less than 35 ft), 0 to 30 per cent.

3. *Primary pressure deficient* from public water-works system, giving less than 15 lb on the highest line of sprinklers, but 8 or more lb pressure (2 per cent deduction for each pound less than 15), 0 to 14 per cent.

4. *Primary pressure* dependent upon approved central-station supervised automatic pump 5 per cent deduction.

5. *Secondary pressure deficient* from gravity tank with bottom less than 20 but 5 ft or more above highest line of sprinklers (⅔ per cent deduction for each foot less than 20), 0 to 10 per cent.

6. *Secondary pressure deficient* from public water works, giving less than 10 but 5 or more lb pressure on the highest line of sprinklers (5 per cent + 1 per cent deduction for each pound deficient), 5 to 10 per cent.

7. *Secondary pressure* dependent on nonautomatic pump, 5 per cent deduction.

8. *Water supply* common to two or more risks, charge risk not in control of supplies, 5 per cent deduction.

9. No recognized secondary supply, 30 per cent deduction.*

* Note—When public water works supply is of exceptional capacity and reliability (volume and pressure), reduce charge under 9, 50 per cent, but this reduction not to apply if any charges in grading are made under 2, 3, 10, and 21 to 29.

Capacity of Water Supplies

10. *Water supply deficient* in available capacity, 1 *to* 20 per cent deduct 10 per cent, and from 21 to 50 per cent deduct 30 per cent.

Equipment

11. Minor unsprinklered areas 10 per cent; important unsprinklered areas 10 per cent + 1 per cent deduction for each 1 per cent of total area unsprinklered.

Note 1. Make no deduction for absence of sprinklers at dynamo or switchboard when in fire-resistive rooms not containing other combustible materials.

Note 2. If building is of noncombustible construction and contents are noncombustible, make no deduction provided it is warranted that no combustibles will be located, temporarily or otherwise, in such unsprinklered area.

12. A small number of sprinklers with water shut off during cold weather, 5 per cent deduction.

13. Charge as follows when sprinklers are on dry pipe system in any one fire area:

	Per cent
a. 25 per cent or under	2½
b. From 26 to 50 per cent	5
c. From 51 to 75 per cent	7½
d. Over 75 per cent	10

Note. If any dry valve controls more than 300 sprinklers or the system has more than 375 gal capacity and does not have a quick opening device attached, double above charges.

14. Water distribution obstructed by shelves, tables, benches, partitions, or by piles of combustible stock or other contents, 10 per cent deduction (cumulative with 11).

15. Clear space of 2 ft below ceilings or bottom of joist not maintained throughout all sprinklered areas, 10 per cent deduction.

16. **Watchman and clock but no approved alarm valve, 5 per cent; no watchman and no approved alarm valve, 15 per cent.**

Note. Not to apply if central-station alarm and/or sprinkler supervisory system installed.

17. No steamer connections where necessary for use of public fire department, 5 per cent deduction.

Note 1. Not to apply if risk in unprotected location.

Note 2. Hydrants (private), if so connected as to be available for steamer connection, will be accepted in lieu thereof, if two or more double female couplings are provided.

18. Other deviations from equipment rules:

	Per cent
a. Unapproved sprinklers	5-25
b. Unapproved dry valve	5-25
c. Unapproved devices (other than above)	5-25
d. Pipe sizes in accordance with standard, previous to 1905	5
e. Supply connections not standard	5-25
f. Spacing in accordance with standard, previous to 1905	5

Note. Unapproved devices and standards are those whose field record has shown them to be serviceable but not approved by Underwriters' Laboratories. Deductions are made variable to cover the degree of probable efficiency, as follows: unapproved, 5 per cent; record, good, 10 per cent; fair, 15 per cent; indifferent, 20 per cent; poor, 25 per cent.

Vertical Openings

19. Unstopped openings between floors without approved curtain boards, 15 per cent deduction; with approved curtain boards, 10 per cent deduction; with semistandard enclosures or automatic traps, 5 per cent deduction.

Maintenance

20. Defects from lack of maintenance: minor, 5 per cent deduction; moderate, 10 per cent deduction; severe, 25 per cent deduction.

Occupations or Constructional Deficiencies

21. Large areas and unusual heights of room, with occupations likely to cause the operation of over 50 sprinklers, such as theaters, halls, erecting shops, foundries, etc., according to condition, 15 per cent deduction.

22. Contents notably preventing full sprinkler distribution, as stored cars (except when building equipped with "car line" sprinklers), barrels, tubs, coffins, large hollow piles of combustible stock such as furniture, etc., according to condition, 20 per cent deduction.

23. Unusual draught conditions, as from large blower systems or buildings with open sides, sheds, etc., according to condition, 15 per cent deduction.

processes, cotton, hemp, jute, or sisal storage, according to condition, 25 per cent deduction.

26 Occupancies allowing quick hot fires, not readily extinguished with water, as in varnish works, lard and oil refineries, and properties containing large quantities of grease, waxes, fats, nonexplosive oils, etc., according to condition, 20 per cent deduction.

27. Apprehensive explosion hazards due to presence of gun powder, dynamite, various chemicals, compounds, etc., according to condition, 30 per cent deduction.

28. Vapors from dangerous quantities of naphtha, gasoline, benzene, volatile oils, alcohols, etc., according to condition, 30 per cent deduction.

29. Manufacture of, or storage of, considerable quantity of pyroxylin products, such as "celluloid," etc., 25 per cent deduction.

FIRE

Definition of Fire. Chemical reaction in some materials is described and explained as the setting up of an activity within the molecule, which causes either the release of hydrogen or the absorption of oxygen.

Combustion. The latter phenomenon, in common parlance, is called "combustion" and as such has many phases, but the phase that presents itself most frequently is combustion resulting in fire or flaming.

Flammability of substances or liquids as the result of combination with oxygen* is termed "oxidizing," and the visible evidence of this is fire or flame, which is accompanied by the liberation or emission of heat. This flammability depends in a large degree upon the construction of the substance or liquid, its gaseous content, the elements subject to volatization, and their ability to absorb oxygen.

Combustion, therefore, takes place in the presence of oxygen, and the absence of this element or the temperature necessary to this reaction governs the intensity of

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* Liebig names the gradual combination of the combustible elements of a body with the oxygen of the air Eramecausis.

Table 5-11*

1. *Grade of Occupancy*† . . . . *Charge as per Table*‡ . . . .

2. *Height.* For seventh story charge 5 per cent of occupancy grade; for eighth story, 10 per cent; and for each additional story, 15 per cent (cumulative).

NOTE. *Not to apply if building of fire-resistive construction and equipment grades 50 per cent or more standard.*

3. *Area.* If joisted construction for each 1,000 sq ft or fraction thereof in excess of 10,000, charge 1 per cent if grading is over 75 per cent and 1¼ per cent if grading is 75 per cent or under.

4. *Floors.* If single board, charge 10 per cent of occupancy grade for each floor of such construction, not to exceed 40 per cent.

5. *Floor Openings (any kind).* If not protected, charge for each floor opening, each floor pierced, 5 per cent of occupancy grade, not to exceed 80 cents. If protected with automatic traps and/or semistandard enclosure, one-quarter charge; curtain boards, half charge.

6. *Interior Finish.* Forming concealed spaces, charge 10 per cent of occupancy grade for each floor not to exceed 40 cents.

7. *Boiler.* (Over 20 lb steam pressure) if in building, unless in fire-resistive room cut off standard, charge 5 per cent of occupancy grade if fuel is coal and 25 per cent if shavings or refuse, not exceeding $2.

8. *Extra Hazards and Internal Management.* Charge as per extra hazard table,† deducting the following percentages according to the grading of the sprinkler equipment.

80% and above = 50%
60–69.9% inclusive = 35%
70–79.9% inclusive = 40%
50–59.9% inclusive = 30%
Under 50% = 25%

9. *Tenants.* If two or more tenants use (item 1) occupancy of principal hazard and for each tenant over one charge as per table page 5, of Sprinkler Schedule of Fire Insurance Rating Organization.

NOTE. *No charge under item 9 if building of fire-resistive construction with standard floor openings and having wired glass exterior windows.*

10. *Watchman.* Without approved watchman and approved watch clock service, 20 cents.

NOTE 1. *If watchman and approved clock service but rounds are substandard, charge in the proportion that service bears to standard.*

NOTE 2. *Make no charge if deduction 20 or 21 is allowed.*

11. *Location.* If risk is located outside public fire protection, charge 10 per cent of occupancy grade, but not less than 8 cents or more than 60 cents.

NOTE. *Make no charge under 11 if risk has an approved yard hydrant and hose system and an approved private fire brigade.*

12. *Conditions not anticipated,* 10 cents.

The building charge of 1 to 12, inclusive, shall be the following per cent of the sum total of such contents charges:

	Item	Building
13. "*Fire Resistive*" With all structural members protected, 30 per cent	-- -	- - ----
14. "*Fire Resistive.*" With part or all structural members not properly protected, or unprotected, 40 per cent.	- -	-----
15. "*Slow Burning*" or "*Mill*" *Construction*, 55 per cent.	--- -	- --------
16. "*Ordinary*" *Brick.* Ordinary joisted construction (including iron on iron frame), 70 per cent.	-	-- ------
17. "*Frame*" Wood construction throughout (including iron on wood frame), but not to exceed 40 cents less than contents rate, 90 per cent.	-	- --- ----

NOTE. *Frame sheds take same rate as contents.*

18 *Base rate* for fire-resistive or slow-burning construction, 10 cents; for all other types of construction, 12 cents.

(Further deductions applying to building and contents to reduce rate)

19. *For Applied Central-station Supervision of Watchman*, 2 per cent

19a. *For Auxiliary or Central-station Fire Alarm* with watchman and applied clock, 3 per cent.

NOTE *Make no allowance under 19 and/or 19a if deductions made under 20 and/or 21.*

20. *Central-station Water-flow Alarm* with or without watchman, 5 per cent.

21. *Full Automatic Sprinkler Supervisory* service and central station alarm, 15 per cent.

22 *Standpipe and Hose* for approved system, 5 per cent.

23. *Private Fire Brigade* For approved brigade with ample apparatus and equipment, 10 per cent.

NOTE. *No credit under 23 if risk is outside city fire protection. See note under* 11.

24. For approved inspection service by automatic sprinkler contractors, 5 per cent

25. *Wire-glass Approved Windows* throughout, 5 per cent

NOTE. *No credit under 25 unless building over one story and of fire-resistive or mill construction*

26. Building only one story high with or without basement and top of roof less than 36 ft above lowest ground level, 5 per cent

27. *Waterproof Scuppered Floors*, 5 per cent

NOTE. *If floors self-draining, floor openings curbed, and stock skidded, 5 per cent additional.*

* Table 5-11 is part of Sprinkler Schedule.

† The grading of the particular occupancy is based upon the rate as an unsprinkled plant and is predicted from a table that classifies the plant as fire-resistive, slow-burning, or ordinary brick or frame, plus the grading of the sprinkler equipment. This table is on file with the Fire Insurance Rating Organization.

‡ Table 5-11 is on file with the Fire Insurance Rating Organization.

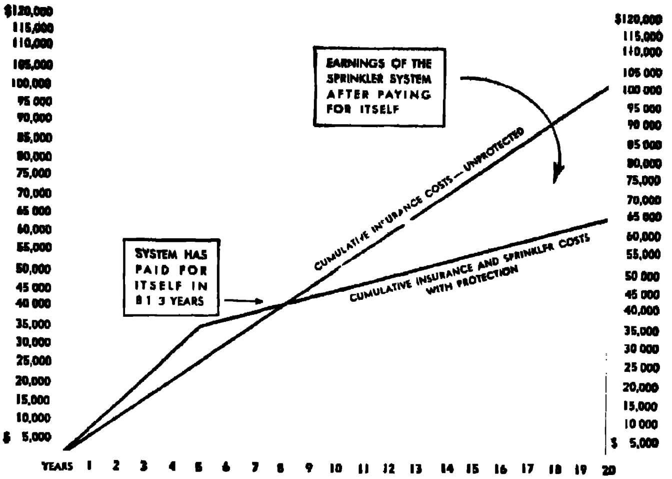

FIG. 5-1. Graph of possible insurance savings. (*Grinnell Company, Inc.*)

the heat liberation. The heat quantity is known as the "heat of combustion" (see Table 5-12), and the extinguishment of fire is therefore based upon the process of depriving a substance or liquid of its heat of combustion, thereby lowering its temperature and cooling it to a point below that at which combustion takes place, or by entirely shutting off the supply of oxygen and retarding the chemical reaction.

Table 5-12. Heat of Combustion—Nuckolls

Substance	Kg-cal	Kg-cal/ g-molecule	Kg-cal/ cu ft of mixture of gas or vapor with air*
Acetic acid	3 395 4‡	203 863	22 82
Acetone	6 969 9‡	404 693	23 75
Acetylene .	11,560 1†	300 911	27 61
Benzene	9 676 1†	755 511	24 30
Carbon disulfide	3,320 3‡	252 776	19 37
Carbon monoxide	2,412 9†	67 561	23 08
Ethane	11 239 4†	339 361	22 73
Ethyl alcohol	6,681 7‡	307 760	23 72
Ethylene	11 399 8†	319 650	24 73
Ethyl ether	8 499 1‡	629 788	25 05
Gasoline vapor	10,500 0	1,065 4§	25 5§
Hydrogen	28 386 4†	57 227	20 05
Kerosene vapor	11,500 0		
Methane	11 981 0†	192 177	21 68

* In combining proportions.

† Temperature 62°F (16 6°C).

‡ Temperature 64 4°F (18 0°C)

§ Based on tests of ordinary gasoline having a mean vapor density of 3 5 (air 1).

NOTE Constant-volume values selected from University of Illinois *Bulletin* 139 and the available literature on the subject.

This process passes through three stages of flammability, which is common to most substances, reducing the temperature to the burning point or ignition point, further to the flash point, and finally to extinction. *Flash point* and *burning or ignition point* are the two indexes by which the danger of fire may be ascertained.

Flash Point Flash point is that temperature to which a liquid or solid must be brought to cause a volatization of its lighter elements, or vapors, which will flash but not continue to burn when ignited by an outside agency.*

Burning or Ignition Point. Burning or ignition point is that temperature to which a liquid or solid must be brought to ignite it and also make it continue to burn without the assistance of the outside agency.*

These stages of combustion differ with various materials and may show wide ranges, or they may be very close, which would indicate that low flash points are danger points with explosive possibilities if the two stages are of equal temperatures.

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* These tests are made by open- or closed-cup method, Persky-Martens or Tagliabue (see *ASTM Codes* D56-21).

It is also important to consider another phase through which liquids and solids pass before reaching the flash point. This is known as the "melting or boiling phase," resulting from the application of heat. The initial stage brings about a physical change resulting in the absorption of heat and increases affinity for oxygen, liberation of flammable gases, and sometimes causes an ebullition or frothing. This action facilitates the flammability of the material and produces explosion possibilities.

The flash point shows whether or not the vapor of the liquid is combustible in the presence of air at atmospheric pressure and depends upon the volatility of the liquid and also upon the presence or absence of other volatile constituents, which may or may not mask the flash as previously mentioned.

> The flash point alone is not a measure of the fire hazard of a liquid in respect to its ignition point, flammable limits, hazards of combination, vapor density, toxicity, and chemical stability. It fails to differentiate between liquids of the exothermic and endothermic classes. It does not reveal a tendency of a liquid to spontaneous ignition.[1]

As high temperatures are needed to reduce some substances to their melting point and raise others to their boiling point, it is not to be assumed that combustion results only from the application of or contact with burning or glowing bodies or from electrical energies such as sparks or lightning. Fire or flaming may be brought about by spontaneous ignition induced by a very low degree of heat, such as solar heat or any heat of low temperature, setting up a reaction that results in raising the substance to its flash point.*

Table 5-13

Distance	Luminous, deg C	Non-luminous, deg C
½ in. above burner.	135*	54*
1½ in. above burner	421*	175*
Apex of inner cone ..	913*	1090*
Center of outer cone	1328*	1533*
Apex of outer cone	728*	1175*

* To change centigrade temperature to Fahrenheit, double the figure, deduct 10 per cent, and add 32. To change centigrade to Fahrenheit, multiply centigrade temperature by $^9{}_5$ and add 32. To change Fahrenheit to centigrade, subtract 32, multiply by $^5{}_9$.

Fire, therefore, is flaming combustion with the liberation of heat. This flame of combustion has different degrees of luminosity depending on the temperature, the presence of solids or unburned particles or carbon compounds in the flame, or the degree of oxygen present.

Flame Luminosity and Temperature. Flame luminosity indicates the intensity of combustion by the amount of heat liberated. Most visible flames liberate the smaller amount of heat, and the less visible flame, the greater amount of heat, showing a more perfect combustion and the absence of unburned carbon particles as well as a liberal supply of oxygen. The Bunsen-burner flame is typical of perfect combustion and shows the ratios between temperature and luminosity. According to Lewes these are as shown in Table 5-13.

Flame temperatures are given in Table 5-14.

There is also a slow combustion which is nonluminous and invisible except for the effects produced. Decaying processes and rusting come under this class. Certain bacterial activities which are described as slow combustion may result in flaming combustion if the action proceeds under favorable conditions for producing this result. Therefore, in the discussion of the "fire problem," we are dealing with a chemical action brought about by the application of heat, resulting in the liberation of gaseous components which burn in the presence of oxygen or air. In order to produce this burning, it is necessary to have a mixture of certain proportions of air or oxygen, and these proportions within which flame is propagated are called "the limits of flammability" or "ranges of explosibility." It is also said of a flame that

Table 5-14. Flame Temperature

Red heat	475°C–525°C— 977°F
Dark-red heat	700°C—1292°F
Cherry-red heat ...	850°C—1562°F
Pale-red heat	950°C—1742°F
Yellow heat	1100°C—2012°F
Grey-white heat	1300°C— 2372°F
Full-white heat	1500°C—2732°F

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* This is particularly noticeable in oily waste and in wood in prolonged contact with steampipes (see Combustion of Wood).

it travels through a liquid, and the speed of travel or burning varies according to the nature of the gas, the proportion of the mixture, and the character of its surroundings. This velocity is measured in meters per second as shown in the graph by Bunte (Fig. 5-3).

Explosive range refers to the definite limitations of combustibility and rate of burning of flammable vapor or dust mixed with the air. When the particles are so widely separated that those set on fire by the igniting medium will not set fire to others that are nearer, the mixture is called too "lean" for combustion and will not burn. When the particles are so close together that they exclude the oxygen necessary for combustion, the mixture is called too "rich," and it will not burn. The concentration, or per cent by volume, between the leanest and the richest mixtures that will burn is called the "explosion range" (see Table 5-15). Between the minimum and maximum limits of the explosion range will be found various phases of slow and rapid combustion.

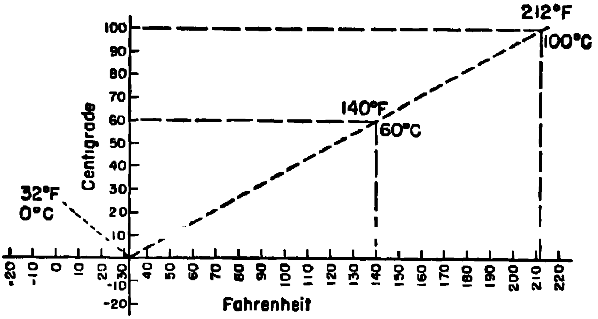

FIG. 5-2. Conversion graph, centigrade to Fahrenheit.

This "explosive wave," as it is termed, results from a vibratory movement which increases in intensity until finally these high velocities are reached. As has been stated, fire or flame, according to the latest experiments, is the result of chemical action, and its intensity is due to the speed of this action augmented by the temperature. As the temperature is applied at an increasing rate, the gases in combination with oxygen or air will supply additional heat, and as this proceeds it finally overcomes the loss of heat from radiation and then continues to burn. This final temperature is the ignition point of the gaseous mixture and will differ in its range from slow-burning

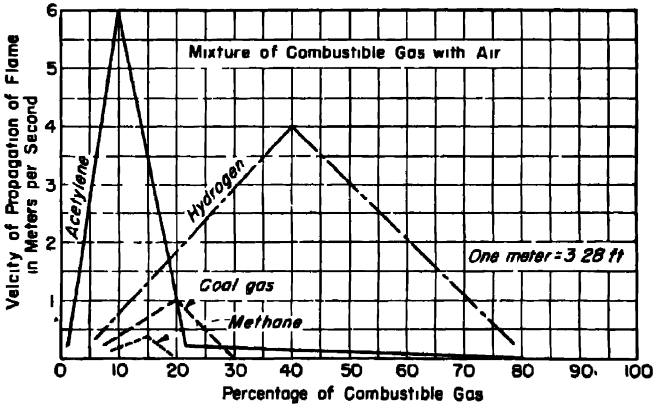

FIG. 5-3. Explosion ranges—Bunte.

Table 5-15. Explosion Ranges—Nuckolls

Material	Lower limit of volume	Upper limit of volume	Range
Gasoline*	1.4	6	4.6
Ether, ethyl†	1.9	22	20.1
Carbon disulfide†	1.0	50	49.0
Benzene§	1.5	8	6.6
Ethyl alcohol	3.5‖	19*	15.5
Carbon monoxide‡	12.5	74	69.9
Illuminating gas¶	5–7	21–31	14–26
Ammonia (NH_3)**	16	25	9
Hydrocyanic acid**	5.6	40	38.4
Methane	5.0	15.0	10
Ethane	3.2	12.5	9.3
Propane	2.4	9.5	7.1
Butane	1.9	8.5	6.6
Pentane	1.4	8	6.6
Hexane	1.3		
Heptane	1.1		
Octane	1.0		

* Nuckolls, W. K. Munn, Underwriters Laboratories.

† Coward, Carpenter and Payman, Bureau of Mines.

‡ Coward and Jones, Bureau of Mines *Bull.* 279.

§ International Critical Tables.

‖ Selected from literature on subject.

¶ Composition varies to some extent.

** Nuckolls, Matson and W. I. Wright, Underwriters Laboratories.

Table 5-16. Ignition Points of Gases

Gas	In oxygen, deg F	In air, deg F
Hydrogen	580–590	580–590
Carbon monoxide (moist)	637–658	644–658
Cyanogen	803–818	850–862
Ethylene	500–519	542–547
Acetylene	416–440	406–440
Hydrogen sulfide	220–235	346–379
Methane	556–700	650–750
Ethane	520–630	520–630
Propane	490–570	
Ammonia	700–860	

to explosive action. This difference in the burning rates of substances or gases is demonstrated by experiments with flames produced by combining gases with air. As the mixtures approach the limits of flammability the cone, as it is termed, changes in luminosity and heat intensity. During this progress the presence of unconsumed material such as carbon produces a film flame which has a center of glowing gas with the outer film of unconsumed gases. As the mass begins to reach the proper mixture for complete combustion, however, the flame or cone becomes entirely luminous and is called a "volume flame."

Table 5-16 by Dixon and Coward, 1909, gives ignition points of a number of gases.

DEFINITIONS APPLIED TO THE PROPERTIES OF HEAT

Oxidation. Oxidation is the combination of other elements with compounds of oxygen or oxides and is also a change from a lower to a higher valence.

Convection. Convection is the act of conveying the diffusion of heat through a liquid or gas by motion of its parts.

Radiation. Radiation is the sending out of rays or direct lines from a point or surface, or issue in rays.

Conduction. Conduction is the slow progress of the vibratory motion from places of higher to places of lower temperature in the same body.

Explosion. When two or more substances, one of which is flammable and the others supporters of combustion, are mixed and ignited, combustion between these bodies takes place with extreme rapidity. The heat evolved increases the pressure of the gases produced and the shock to the surrounding air causes a loud detonation, which is called an "explosion."

FIRE PREVENTION

From the proceeding text it has been pointed out that fire or flame proceeds in some sort of ratio to the amount of heat energy which is developed or applied to a solid or liquid, and the chemical action can be rapid or slow owing to the conditions attending. These conditions may vary widely, some reacting with explosive violence where the flame propagation is rapid through a highly volatile material, or slow charring may continue for a long time and never result in anything more visible than an incandescence. This condition naturally is the result to be expected when dealing with a multitude of different kinds and classes of materials, ranging from the heavy solids to light and porous materials and from liquids of varying specific gravities and flash points. With the exception of a few materials it may be safe to say that all substances are in a preferred class under normal and good conditions. If substances were unexposed to outside influences and kept in a state precluding reactions from within, there would be little danger from any source, and the fire hazard would be reduced to a minimum.

There are three steps in the process of fire prevention.

1. The analysis of the disturbing forces or causes of fire which will lead to an understanding of them and, in consequence, a method for avoiding or preventing them. In analyzing fire hazards, the latest technique is to endeavor to visualize or measure "the heat of combustion" of the materials or mass of combustibles. The heat of combustion can be determined from tables that give the relative values of various materials in calories or British thermal units, and these can be used to ascertain the probable amount of heat which would be generated in a burning mass of material.

2. The protection of the materials from ignition by taking precautions commensurate with their flash and ignition points and susceptibility to spontaneous ignition, the segregation of quantities, and prescribing construction to prevent the horizontal and vertical spread of heat and flame.

3. The necessary fire protection to cope with any outbreak which might occur, measured in terms of the nature of the materials and the amount of "heat of combustion" to be overcome.

Fire prevention, as a science, is the preconception of conditions that may cause the heating of any material from within itself or from any outside source; the preconceived knowledge of the physical and chemical properties of materials to determine their ability to create a slow or rapid combustion and the amount of the heat of combustion which may be evolved; the understanding of the susceptibility of materials to moisture, mold, water (hot or cold), smoke, charring, and damageability by fire, water, or smoke, the visualization of the possible mass of combustible material which may be affected· and the study for the removal of the cause that may be responsible for any of the conditions outlined and the removal of the material from the cause.

If plant operations make it necessary to condone a hazardous process or hazardous material, the pertinent thing to do is to safeguard the process in a standard manner, protect the hazardous material from contact with other material, and then remove all other material to safe distances.

The science of fire prevention and fire protection is defined as *comburology*,* from the Latin word *comburo*, meaning to burn, to consume, as by fire. This is the study of the occurrence and augmentation or reactions due to the application of temperature by radiated and converted heat, friction, and pressure. It also includes the reactions that are due to fermentation, spontaneous heating, and oxidation and the disturbing of the molecular equilibrium of solids, gases, and dusts and the control of electrical energy. This is also the study of flash points, ignition points, kindling points, explosive ranges, and pressures and heat of combustion of the various elements. The visible heat of these reactions, caused by the absorption of oxygen or the liberation of hydrogen or causing incandescence, is called "fire."

This study also takes into consideration the natural elements which augment conditions inherent in these phenomena. This study is therefore directed to the maintenance of equilibrium; the occlusion of oxygen; the control of temperature, friction, and pressure; the prevention of arcing and heating by electrical energy; and the application of the knowledge of these physical facts to the protection of structural members, the segregation of combustible mass, both horizontally and vertically, and the protection of combustibles from these phenomena; and the use of various antidotal mediums to be applied in the control and prevention of these phenomena.

Causes of Fire. Von Schwartz, the eminent physicist, proposed a list under which causes or phenomena of combustion could be classified. This list covers practically all the common causes of fire prevalent in America. The causes, eleven in number, are as follows:

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* This definition was proposed Dec. 27, 1943, at the Center for Safety, New York University, as a basis for a possible science.

1. *Direct Contact with Flaming or Glowing Materials.* Under direct contact it will at once be apparent that the open light is very important. Candles used for various purposes and left unguarded by plumbers and plumbers' torches for thawing pipes and sometimes used by electricians for soldering joints are examples of open lights. In industrial plants the open flame may be present in the form of an open forge, a gas stove, a pilot light at a water gauge (see *a* below), or a gas jet used for purposes of singeing or heating glue.

2. *Application of a Low Degree of Heat for a Prolonged Time.* The application of a low degree of heat for a prolonged time includes primarily the steampipe, both live and exhaust steam, which may pass through a floor or partition of wood without proper clearance (see *b* below) or may be located underneath a floor in close proximity to the wood floor or in such a manner as to cause dust to settle around it or to attract mice to build nests in contact with it owing to the warmth.

a. A disastrous fire was caused by the breaking of a gasoline supply line, the vapors extending across an open runway to the boilerhouse where they were ignited by a pilot light at a water gauge causing a flash back and an explosion.

b. The ignition point of wood is also a factor. Boiler stacks and breechings improperly insulated permit a heat radiation which is dangerous to wood structure in industrial plants and is a major source of danger. The head, or distance, between the breeching, or uptake, and the roof subjects the wood of the roof to a constant heat and will eventually produce serious results, as will the stack also as it passes through the roof. Ordinary stovepipes and hot-air heater pipes passing through or inside partitions are rightly under this heading. A simple application of this principle is the placing of paper around electric-light bulbs for decorative purposes or for shedding or shading the light. The latter is often done while working on motors in garages and is a dangerous practice.

3. *Spontaneous Heating and Ignition.* To avoid fires attributed to this cause requires some knowledge of the character of materials and their tendencies. For, although it may be said that it is difficult to avoid entirely conditions that may result in spontaneous ignition, this statement can be proved somewhat erroneous by using simple precautions. Oxidation is produced by many causes and may be in process for a long time before the temperature is raised to the flash points. Pressure such as found in stacks of printed paper, cardboard, or treated leather, as well as in coal stacks, principally soft coal, will prompt this action. Other causes of spontaneous ignition are friction between organic members due to vibration or wind motion; bacterial activity set up by the slow heating of a mass owing to pressure or germination, such as improper curing of hay or grains; chemical reactions due to the improper storage of various substances that set up an activity if in combination, such as any organic material with acids, oxygen carriers, picrates, chlorates, nitrates, fats, or oxidizing oils. Dampness or the possible wetting of materials such as lampblack, metallic oxides, and lime will cause heating and ignition. Oxidizing oils and, under certain conditions, any oil thinly spread over a surface, especially of porous material, will heat and flame. This action is noticeable in rags and waste used for wiping machinery and oily parts in industrial plants, as well as in metal turnings or iron filings coated with oil, and in various types of floor mops and wiping cloths.

4. *Explosion or Rapid Propagation.* To treat this classification properly would require a larger treatise than this, as it covers a vast number of possibilities, both chemical and mechanical, so that reference to the more common causes only will be attempted. Explosions result from rapid decomposition with release of large quantities of heat and rapid flame propagation through released vapors or by instantaneous release of pressures. To avoid this possibility, materials should be analyzed for properties that may produce one of the three characteristics mentioned. Among the principal materials which are met in the daily activities of industry and commerce[1] may be cited explosives used for blasting, such as nitroglycerin and nitrogen compounds, picrates, chlorates, and fulminates, guncotton, pyroxylin plastics, collodions, celluloids, and gases and vapors of highly volatile liquids as indicated by low flash

points or explosive ranges, dust formed by mechanical processes, and accumulations due to storage or handling. Also finely divided dust particles of almost any character can be ignited by contact with flame or sparks, but organic dusts and some metallic powders, such as magnesium, aluminum, and iron, are more susceptible.

5. *Lightning.* Lightning is a so-called "act of God." Lightning strikes according to no set rule. All are subject to its dangers, and as a cause of setting fire it is perhaps the most potent. It will ignite almost any material as its own temperature is comparable to an electric arc, and the speed of its flight provides little time to avoid it. Lightning rods serve only to assist lightning in its course, and such, when properly installed, have proved 95 per cent efficient. In addition, arresters, circuit breakers, or fuse plugs are helpful.

6. *Dust Explosions.* Dust clouds can be formed by simple combustion of materials, *e.g.*, lampblack; dry distillation or sublimation of sulfur or napthalene; mechanical pulverization, grinding, pounding, beating; mechanical purification, screening and blowing, of flour and grains; transporting materials from place to place; and bursting of bags, etc.

The following materials are susceptible to dust explosions:
Carbon of almost any kind
Lampblack
Wood dust, cork, tanbark
Grain, bran, malt, spices, medicinal barks, roots, etc.
Flour, starch, sugar, dextrin
Wool, fibers, tow, fluff from shearing
Metal or bronze and aluminum powders
Colophony, resins, celluloid, pyroxylin plastics
Sulfur, naphthalene

Any kind of organic and a great many kinds of inorganic dust floating in air can be ignited by the simplest flame or by electric or other sparks.

7. *Electric Sparks.* Electricity or electrical energy is a source of intense heat and, if released in the form of sparks or arcs due to poor insulation or broken connections in the installation, will ignite materials or vapors. Insulation may be broken down by friction or disintegrated by fumes or vapors or moisture, and short circuits are caused because of contacts of the naked wire with other conductors. Overloading of circuits through faulty installations or lightning charges will also break down the insulation. The use of incandescent lamps in places subject to mechanical injury will result in breakage and arcing.

8. *Chemical Reactions.* Chemical reactions include any and all the phenomena incident to the use, handling, or storing of substances involving even the slightest possibility of chemical change either within themselves or in combination with other materials. These reactions take place because of the inherent characteristics of the substances, *e.g.*, spontaneous ignition when slightly warmed or explosion by percussion. In combination with materials, the most prevalent cause of chemical reactions is due to affinities and natural reactions. Most of these combinations are understood by chemists and users of hazardous chemicals; but there is a wide field of industrial pursuits which involve the use of many substances which require careful handling and storage. Even in small quantities, some chemicals may cause an initial blaze sufficient to ignite surrounding material. To list these substances would make this a lengthy study and rather a duplication, for there are many chemical handbooks to which reference can be made. To exemplify the principle involved, attention is directed to dangerous combinations such as nitrate of soda with organic material such as gunny sacks or cellulose; lampblack with water; unslaked lime with water; nitric acid with organic or carbonaceous material; or sulfur with charcoal.*

9. *Friction, Pressure, Shock Falls, or Concussion.* Friction, pressure, shock falls, or concussion are closely allied with temperature and should be studied with this in mind as the production of heat is the ultimate result from each. In the case of friction and pressure, the action may be slow in the first instance, while concussion may pro-

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* A recent fire was caused by the perspiration of a workman dropping into the contents of a barrel he was mixing, presumably a peroxide.

duce a rapid action. Friction (see also Static Electricity below)[5] produces heat. The rubbing together of organic or metallic surfaces will raise the temperature and if continued will ignite the material, if its ignition point is within that possibility, or will release vapors which may be ignited, or, as in the case of wood, will reduce the cellulose to a pyrophory carbon causing ignition. Belts or shafting at points of support and flowing volatile liquids in pipes are common producers of friction. Pressures may result from mechanical means such as pumping, compression of gases, or piling. These pressures produce heat which causes expansion or molecular activity which results in ignition or explosion.

Shocks or concussion may assume a variety of forms producing similar results, different substances reacting according to their composition. Gunpowder, detonators, and explosives of similar nature are very susceptible, while there are many materials which are not explosive but will ignite from concussion.

10. *Focused Rays.* The science of physics gives a clear explanation of the result of focused rays—that of applying heat rays to produce high temperatures. This can be accomplished by reflection or concentration. Polished surfaces or mirrors will reflect the heat rays of the sun, transmitting them to other objects which may be near enough to absorb them before they are cooled by the intervening currents of air. These reflected rays will cause a rise in temperature to ignition points if applied for a given length of time.

The concentration of heat rays is also accomplished by a combination of curved surfaces of materials which will transmit them. The burning glass of boyhood days is one example, and imperfections in sheet glass (bubbles) is another. The latter will direct a concentrated heat ray upon objects or substances and cause ignition. Many instances have been recorded recently of the ignition of celluloid articles in showcases and motion-picture films in vaults from these imperfections in glass.

11. *Static Electricity.* The demands of industry have brought into use many devices and materials that have developed causes of fire. One of the outstanding causes thus produced is static electricity. Its control has baffled the fire-prevention engineer until recent years when the study of processes and materials has shed some light upon the reason for its generation. Friction, or the rubbing together of two surfaces, will produce electric charges. There are so many instances of its generation that cannot be explained by the theory of friction, however, that it is difficult to analyze the law which governs it.[*]

Belting for transmitting power has been a source of static electricity and for a long time has caused fires in cotton mills. Today this continues to be the case in industries using flammable material and liquids. It is undoubtedly due to friction between the belt and the air or the pulley, or both. The built-up charge can be drawn off in a visible spark or current by placing a suitable conductor near the belt, thus proving its presence. In order to avoid building up too heavy a charge, it is necessary to install a permanent collector in the form of a comb composed of copper or other suitable material. This should be in contact with the belt and be connected with a good ground.[†]

Mixing materials by stirring or tumbling generates a charge which remains in the material after it is removed from the mixer and may not be discharged until further used in the process.[‡]

Dry-cleaning processes which use naptha or gasoline for washing garments develop heavy charges which have caused fires and explosions. Garments of silky material augment this charge, and if they are further ornamented with metal trim, the danger is increased because of possible contacts with metal parts of the machinery, causing spark gaps. Care must be exercised to avoid these spark gaps in removing the garments from the machines also. The best method is to place the hand under the garment to lift it from the liquid, thus permitting the static to pass through the arm and body of the operator. Grounds should be placed on the machines and also on the movable shafts connected to the revolving parts of the machines.

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[*] The strange case of an enraged cat generating a sufficient charge to produce a spark that ignited an flammable gas cannot be said to have been produced by friction.

[†] A mixture of fish glue, glycerine, lampblack or ammonia hydroxide for leather belting or lampblack and spar varnish for rubber belting is said to reduce the static charge

[‡] In the rubber industry it has developed that, if milled or mixed rubber is laid upon a copper wire carpet for a length of time, there has been no development of static when the material was dissolved further for use in dipping processes.

Flammable liquids flowing through pipes will generate considerable static and also hold the charge if there is not an outlet for it. This has developed into a serious danger at gasoline filling stations and in tank cars and wagons as well as in certain industries. Where the discharge nozzle or tip of the discharge pipe is of copper and is grounded, the static is drawn off, and no trouble is experienced. There is not very much data on which to base a comprehensive explanation of this phenomenon, but the progress made so far indicates that it is a danger to be reckoned with and deserves the attention of the physiochemist.

Coating processes involving both the use of rubber compounds and pyroxylin plastics, where the application is by spreading, produce a severe problem in static-electricity control. Today these processes are safeguarded by various methods of grounding, the use of inert gases in the enclosures of coating machines, and the introduction of nonsparking metals at all friction points.

Equilibriums. Fire prevention is the study of "equilibriums" and the maintenance of those equilibriums by proper safeguards This resolves itself to a consideration of the temperatures by which materials, solid, liquid, and gas, are affected either in their structural stability, their physical form, or their action due to pressure. All materials under normal conditions may be said to be in a preferred class, or at "equilibrium." It is the disturbing of this normal condition which brings about a change which is classed as dangerous or hazardous.

Therefore precautions are instituted in almost all cases to protect the materials from outside influences, except in rare cases where a material may contain within itself the element for its own destruction, *e.g.*, metallic sodium. Standards are therefore devised to reduce to a minimum the outside influence which would disturb the equilibrium or to reduce the effects produced should the material be transformed from its quiescent state. This idea of the equilibrium of materials, if kept clearly in mind, will make the science of fire prevention an understandable one and will enable the use of standards that can be applied to all similar problems regardless of where they are located.

STANDARDS FOR THE SAFEGUARDING OF HAZARDS

The following standards have been adopted by the National Fire Protection Association and are also official standards of the National Board of Fire Underwriters:

Acetylene Equipment for Lighting, Heating, and Cooking
Air-conditioning, Warm-air Heating, Air-cooling, and Ventilating Systems
Blower and Exhaust Systems for Dust, Stock, and Vapor Removal
City Gas, Installation, Maintenance, and Use of Piping and Fittings
Combustible Fibers, Storage and Handling
Dip Tanks Containing Flammable Liquids, Including Hardening and Tempering Tanks; Flow-coat work
Dry-cleaning and Dry-dyeing Plants, Safeguarding of
Dust Explosions in Industrial Plants, Fundamental Principles for the Prevention of
Dust Explosions in Starch Factories, Terminal Grain Elevators, Flour and Feed Mills
Electric Wiring and Apparatus. (National Electrical Code)
Gasoline Vapor Gas Machines, Lamps, and Systems
Gas Shutoff Valves, Emergency
Gas Systems for Welding and Cutting
Internal-combustion Engines (Gas, Gasoline, Kerosene, Fuel Oil) and Coal-gas Producers (Pressure and Suction Systems)
Liquefied Petroleum Gases, Storage and Handling, Including Compressed-gas Systems for Cooling and Heating, Tank Trucks, and Liquefied Petroleum Gas as a Motor Fuel
Nitrocellulose Motion-picture Films, Storage and Handling
Oil-burning Equipments and the Storage and Use of Oil Fuels in Connection Therewith
Ovens for Japan, Enamel, and Other Flammable Finishes
Paint Spraying and Spray Booths
Photographic and X-ray Nitrocellulose Film, Storage and Handling
Pulverized-fuel Systems, Installation of
Pyroxylin Plastic, Storage, Handling, and Use of in Factories Making Articles Therefrom
Pyroxylin Plastic, Storage, and Handling in Warehouses and Wholesale Jobbing and Retail Stores
Small Heating and Cooling Appliances (Kerosene and Fuel Oil)

Standards and Special Committee Reports

(Published by the National Fire Protection Association)

Anaesthetic Gases and Oxygen, Construction and Installation of Piping Systems for the Distribution of in Hospitals and Similar Institutions and for the Construction and Operation of Oxygen Chambers

Anaesthetic Combustible and Operating-room Explosives

Blower and Exhaust Systems

Dust Explosions

Flameproofing of Textiles

Fire-hazard Properties of Certain Flammable Liquids, Gases, and Volatile Solvents

Fumigation of Grain-storage and Grain-processing Plants

Gasoline Stoves, Installation of, for Cooking and Heating

Gasoline Blowtorches and Plumbers' Furnaces

Lacquer-manufacturing Plants, Suggestions for the Protection of

Lightning, Protection of Life and Property against

Magnesium-dust-explosion Code

Marine Fire Hazards

Mechanical Refrigeration, Hazards and Safeguarding of

Protection of Records

Salvaging Operations

Shoe Factories, Suggestions for Their Improvement as Fire Hazards

Spark Arresters, Standard for Construction and Installation of Chimneys and Stacks

Spontaneous Ignition of Coal and Other Mining Products

Static Electricity

A Table of Common Hazardous Chemicals

Reference Books

(Published by the National Fire Protection Association)

These reference books include the National Fire Codes on flammable liquids, gases, chemicals, and explosives. They also include a compilation of 13 separate NFPA standards for the prevention of dust explosions in the industries affected, including necessary measures to secure reasonable safety to life and property from dust explosions and fire.

Fundamental principles include

Aluminum Bronze Powder

Coal Pneumatic Cleaning

Flour and Feed Mills

Magnesium Powder or Dust

Pulverized Fuel

Spice Grinding

Starch Factories

Sugar and Cocoa

Terminal Grain Elevators

Wood Flour

Woodworking

Inert Gas for Explosion Prevention

List of Dust Explosions in the United States

Educational Pamphlets

(Published by the National Fire Protection Association)

Artillery-ammunition Manufacture, A. B. Guise and J. S. Slicer, Jr.

Bleaching Wood with Hydrogen Peroxide

Drying by Infrared Lamps, Benjamin Richards

Dust-explosion Hazards of Powdered Metals, Hylton R. Brown

Fighting Magnesium Fires

Fire Gases, George E. Ferguson

Fire Hazards of Termite Control

Food Dehydration

Inspected Electrical Equipment, published by Underwriters' Laboratories, Inc. (revised semiannually)

Inspected Gas, Oil, and Miscellaneous Appliances, published by Underwriters' Laboratories, Inc. (revised semiannually)

Liquefied-petroleum-gas Systems

Magnesium and Its Alloys, H. S. Hirst and A. B. Guise

Metalworking Processes, W. D. Milne, (including fire record of metalworking industries)

New Hazards of Dry Cleaning, R. S. Moulton

Oil-tank Fire Tests

Pitch for Extinguishing Magnesium Fires, Hylton R. Brown and Irving Hartmann

Preventing Air-conditioning Fires, Carl W. Wheelock

Railroad Fire Hazards, deWill Rapalje
Safeguarding Gas Appliances, F. E. Vanderveer
Smoke and Water Damage, F. E. Roberts
Spontaneous Ignition of Chopped Hay
Spontaneous Ignition in Mines and Mineral Products
Static Electricity as a Fire Cause, E E. Turkington
Synthetic Rubber, Gerald Hallowell
Treatment of Burns with Aniline Dyes, Robert H. Aldrich, M.D.
Venting of Tanks Exposed to Fires
Venting Dust Explosions

CONSTRUCTION

The high temperatures recorded in conflagrations reach between 2200 to 2300°F. Any material that is used in construction for the purpose of protection or the prevention of the passage of fire and heat, therefore, must have a resistance sufficient to withstand the "hazard of the occupancy" and the "fire severity" to which it may be subjected.

Fire ratings are based upon the time element of exposure to fire and heat and range from ½ to 4 hr. (These fire ratings and construction details for buildings are set forth in the National Board of Fire Underwriters, Standard Building Code.) The standard methods of fire tests of building construction and materials are E119.47 of the American Society for Testing Materials, American Standard A21-1942 as approved by the American Standards Association, and British Standard Test for Fire Resistance (BSS 476). In the American specifications, average indicated furnace temperatures are required to conform to the following values within allowable tolerance limits:

1000°F	at	5 min
1300	at	10 min
1550	at	30 min
1700	at	1 hr
1850	at	2 hr
2000	at	4 hr
2300	at	8 hr

The temperature should continue at 2300°F.

The fire-resistance point is taken to be reached if

1. An average temperature rise of 250°F or a maximum rise of 325°F, measured with thermocouples under asbestos pads, is attained on the unexposed side of the wall.
2. Passage of heat, flame, or gases hot enough to ignite cotton waste occurs.
3. Failure under the design load occurs (load-bearing construction only).

The fire resistance of building materials or the protective coverings for steel members used in construction should conform to the standard tests if their equilibrium is to be maintained.

Steel expands eight parts in a million for each degree Fahrenheit, which means that, if it is heated to 1000°F, it can expand 9½ in. in a 100-ft length.

Steel under tension can fail under a temperature as low as 1200°F, the critical temperature of steel. Steel columns have a maximum useful limit of 900 to 1100°F.

Maintaining the equilibrium of the building itself and its structural members is commensurate with the degree of volume of fire, heat, gases, smoke, and the effects of water used in the extinguishment of the fire. It is therefore essential to consider the factors that contribute to these conditions. These factors are area, height-horizontal segregation, and vertical segregation.

The confining of a fire and its effects to the smallest possible area is one of the first principles of fire prevention.

Table 5-17 is the standard table for area for square foot of the plot, depending upon the type of building and the type of construction.

Table 5-17. Area Limits per Floor of Buildings in Square Feet

Types of Construction

Fireproof	Semifireproof	Street fronts	Heavy timber	Ordinary and noncombustible		Wood frame	Unprotected metal
			Per story	One story	More than one story	Outside the fire limits	
No limit		One street	6 500	6 000	5 000		One story, no limit More than one story, same as noncombustible
		Two streets	8 000	7 500	6 000		
		Three streets	10 000	9 000	7 500	5 000	
		Four streets	10 000	10 500	9 000		

NOTE For area modifications and also for area limits for certain special occupancies for sprinkler requirements based on occupancy see National Board of Underwriters Building Code 1919

Table 5-18. Building Height Limits*

Classification of occupancy	Types of construction					
	Fireproof	Semifireproof	Heavy timber	Ordinary	Noncombustible	Wood frame and unprotected metal*
Public buildings	No limit	75 ft except no limit if less than 100 persons above 75 ft	Not to exceed 4 stories or 55 ft	Not to exceed 3 stories or 35 ft Churches 2 stories 45 ft Schools 2 stories		Not to exceed 2 stories or 35 ft
Institutional buildings	No limit	75 ft	Not to exceed 2 stories or 35 ft			Not to exceed 1 story or 35 ft
Residence buildings	No limit	No limit	75 ft	Not to exceed 3 stories or 45 ft*	Not to exceed 3 stories or 45 ft	Not to exceed 2 stories or 35 ft Dwellings other than prefabricated may be 3 stories
Business buildings	No limit	75 ft except office buildings no limit	75 ft	Not to exceed 4 stories or 50 ft		Not to exceed 2 stories or 35 ft
Storage buildings	No limit	50 ft 75 ft if sprinklered	35 ft 75 ft if sprinklered	35 ft 50 ft if sprinklered		Not to exceed 1 story or 35 ft

* National Board of Fire Underwriters Building Code also applies special heights within fire limits to certain special occupancies for sprinkler requirements based on occupancy

NOTE With a 2 hr noncombustible first floor height may be 4 stories or 55 ft in multifamily houses with a 2 hr noncombustible first floor and other floors 1 hr with fire partitions subdividing floors into areas not exceeding 3 000 sq ft the height may be 5 stories or 65 ft

Wired Glass. The first patent for wired glass was granted in 1855 to an Englishman Owing to difficulties of manufacture, however, its use did not materialize until 1893 when the award was made to Scott and Sherman by the Franklin Institute for the machine and process for making it

Its fire-resistive qualities were discovered by accident. It will stand up to a temperature of approximately 1400°F (by test). There are two types of wired glass—smooth and roughened surfaces.

Wired glass is generally installed in frames of solid metal or hollow metal specially designed to hold the glass under fire conditions. Although the standard requirement for wired glass recommends its use in moderate or light exposures, it has stood up well under many adverse conditions. With a double retardment (such as a fire shutter), it is recommended for severe exposure.

Wired glass does *not* prevent the passage of radiated heat.

Sizes of individual glass lights are limited in the standard for its use.

Class E—moderate exposure, 720 sq ft exposed area, 54 in. vertical, 48 in. horizontal

Class F—light exposure. 2,916 sq ft exposed area, 54 in. in either direction (Underwriters, Laboratories)

Window openings at outside fire escapes should be protected by wired glass in metal frames.

Glass Blocks. Glass blocks are good for panel walls but not for bearing walls. They are acceptable for protection in light exposures, *i.e.*, class F (above). They have a relatively high thermal insulation and are suitable for window openings not exceeding 120 sq ft in area or 12 ft in width and height. Expansion joints must be permitted at sides and tops.

Roofs. Roofs should have a degree of fire resistance equal to or greater than that of the buildings on which they are placed.

A classification of roof coverings as set forth by the Underwriters' Laboratories is as follows:

Class A—Effective against severe fire exposure. Not readily flammable; do not carry or communicate fire. Afford a slight degree of heat insulation. Do not slip; no flying-brand hazard. Do not require frequent repairs to maintain their fire-resistive properties.

Class B—Effective against moderate fire exposure. Same as class A with moderate degrees of heat insulation. May require infrequent repairs.

Class C—Effective against light fire exposure. Same as class B with at least a slight degree of heat insulation. May require occasional repairs.

Various materials are acceptable to meet these classifications, and they are listed in the standards of Underwriters' Laboratories.

Wood Shingles. Wood shingles have contributed to 59 conflagrations in the United States and Canada. The reasons for the susceptibility of wood shingles to fire are their dryness and high winds. Fire retardants for wood shingles or wood in general are

Diammonium phosphate
Monoammonium phosphate
Monomagnesium phosphate
Phosphoric acid
Zinc chloride

Snow Loads. Snow loads constitute an important factor in roof construction. Snow loads for some sections of the United States are shown in Table 5-19.

Table 5-19. Snow Loads

	Psf area
Southern and Pacific states	5
Central states	30
New England states	40
Northwestern states	45

Skylights. Skylights should be so constructed as to prevent the entrance of falling firebrands. They should also act as vents over elevators, stairways, and stages in theaters and over fire sections of piers and wharves. Skylights in flat roofs should be constructed of wired glass in metal frames or ribbed glass (plain) in metal frames protected by wire screening of No. 12 gauge set on angle irons 1 ft above the sky-light and extending 1 ft on all sides. Texas or monitor skylights should be protected by screening of not less than No. 12 gauge or wired glass in horizontal sides if possible.

Skylights over stages in theaters are of special design and are usually equipped with automatic devices for quick opening in case of fire. These devices are of sliding-track or butterfly design with fusible link and weights under remote control. Similar devices should be installed on piers and wharves.

Exits. The standard for exits is set forth in the Exit Code of the National Fire Protection Association.

There are three additional elements which must be taken into consideration in relation to fire prevention as applied to buildings for the maintenance of the equilibrium of the structure and its equipment which is attached to or supported by the structure, *viz.*, hurricanes, termites, and dry rot. The latter two elements may undermine the integrity of the structural wood members and may cause collapse because of wind movement, excessive weights from stock piling, or water used in fire extinguishment.

Hurricanes. Tests at the U.S. Bureau of Standards have shown that, as wind blows against a building, it produces a positive pressure on the windward side (the direction from which the wind is coming) or on some portion thereof but produces a negative or suction pressure on the other sides of the building (the leeward or direction in which the wind is going). A roof will have an area of positive pressure on a windward slope if the pitch approximates 30 deg or steeper, but other portions of the roof will also be under suction pressure. The wind-tunnel tests have indicated that positive wind pressures vary up to a maximum of what is known as "wind velocity pressures." Suction pressures, on the other hand, have been shown to exceed the velocity pressures in some instances. The total wind force acting on a building as a whole is the sum of the positive force on the windward area plus the negative force on the leeward area. External projections, appendages, and openings should therefore be avoided. A reasonable amount of streamlining is desirable.

In cities and towns subject to hurricanes, except for buildings and structures where height does not exceed $1\frac{1}{2}$ times the least width, the design wind pressure upon vertical surfaces shall be as shown in Tables 5-20 and 5-21 which are to supplement Standard Building Code requirements.

Table 5-20

Maximum height above ground, ft	*Normal wind pressure over entire height, psf*
25	25
75	35
Over 75	45

The wind pressure on sprinkler tanks, sky signs, or other similarly exposed structures and their supports shall be taken as not less than 50 psf of the gross area of the plane surface, acting in all directions. On circular tanks, towers, or stacks the pressure shall be assumed to act on six-tenths of the projected area.

Table 5-21. Wind Velocity Pressures*

(Based on air at 15°C and 760 mm Hg)

True wind speed, mph	*Velocity pressure, psf*
30	2.30
40	4.09
50	6.39
60	9.21
70	12.5
80	16.4
90	20.7
100	25.6
110	30.9
120	36.8
130	43.3
140	50.2
150	57.5
160	65.8
170	74.2
180	83.0
190	92.6
200	102.4

* National Board of Fire Underwriters, 1937.

Dry Rot. The U.S. Department of Agriculture defines dry rot as follows: Dry rot is the term given to decay in wood which is the result of a fungus which attacks the wood structure. Wood that has been kept dry, weatherproofed, or thoroughly saturated with moisture is not subject to decay. Dampness or mild to warm weather are favorable conditions for the growth of fungi. The fungus is a plant, and although wood is the food of the plant, it cannot become active unless there is at least 20 per cent water present, based upon the weight of oven-dried wood. The fungus *Poria incrassata*, however, has the ability to seek water and promote its own growth. Posts and poles set in the ground will become rotted at the ground line if this fungus is present, regardless of the type of wood.

Postings in buildings are subject to rot at the floor level where the floors are damp or mildewed. The sapwood of all timbers is subject to this decay, but the cedar, catalpa, chestnut, southern cypress, juniper, black locust, and redwood resist the decay longer. Wood preservatives that penetrate to the heartwood will prevent this decay and maintain the integrity of the wood. Creosote treatment is recommended.

Termites. In the spring and occasionally in the fall, male and female termites, white-winged with black or brown bodies, seek points for the establishment of new colonies. These swarming or flying termites are not capable of attacking wood. The descendants of these winged migrants, the workers, *i.e.*, cream-colored insects, are the wood-mining and destructive members of the colony. These insects shun the light and are concealed in the shell of the wood. With the common ground-nesting or subterranean form of termite, the invasion of buildings is mainly direct to the woodwork in contact with the ground from outdoor colonies through tunnels made by the workers under the ground.

From the report and the work of the U.S. Department of Agriculture,[6] the conclusion is drawn that proper construction or reconstruction is the most effective method of prevention of or remedy for attack by termites. In the construction of new buildings, maximum immunity from termite infestation can be secured by eliminating all wood from foundations, cellars, and basements. Where it is not feasible to eliminate all such wooden members, reasonable security can be secured by treating all structural wood within 18 in. of the ground with creosote or an equally effective toxic preservative.

Wood treated with coal-tar creosote under pressure and in accordance with the specifications of the American Wood Preservers' Association will give satisfactory protection under all known conditions.

Zinc chloride, although not so effective as creosote, is used on structural members, particularly when the wood is to be painted. The introduction of poison dusts into the termite galleries is the most effective known remedial treatment for wood infested with the dry-wood termite. The most effective poison dusts are Paris green, arsenical smelter dust, and finely ground sodium fluosilicate. Sodium fluosilicate is nonpoisonous to man but is effective only if the infested wood is dry. Subterranean termites cannot be satisfactorily controlled by poison dusts. This is particularly true where the infested wood on or in the ground is damp. Cleanliness should be observed and care should be taken in handling poisonous dust. It should not be breathed nor allowed to get on the clothing. If either arsenical smelter dust or Paris green is swallowed, an emetic should be given and a physician called. The danger to firemen in buildings that have been treated with poison dusts is only slight and not so great as carbon monoxide so frequently encountered.

The use of insecticides or fumigants, if flammable, for termite control introduces a severe fire hazard if they flash below 100°F. Precautions include thorough ventilation to prevent combustible mixtures of air and vapor and elimination of open lights and all devices that may produce an arc or a spark. All liquids used for termite control should be tested and approved by Underwriters' Laboratories. The use of such soil poisons as orthodichlorobenzene in liquid or crystal form placed in hollow trenches about exterior foundation walls will give temporary relief.

Horizontal Separation—Walls. For exterior walls of brick or concrete the standards for walls, as set forth in Building Codes should be followed.

An 8-in. brick wall has a rating of 4 hr. The equivalent in reinforced concrete is 11 in.

1. *Bearing Wall.* A bearing wall is a wall that supports any vertical load in addition to its own weight.

2. *Nonbearing Wall.* A nonbearing wall is a wall that supports no load other than its own weight.

3. *Curtain or Panel Wall.* A curtain or panel wall is a nonbearing wall between columns or piers, not supported by girders or beams.

4. *Fire Partition.* A fire partition is a partition that subdivides a building to restrict the spread of fire or to provide an area of refuge. It is not necessarily continuous through all stories or extended through the roof. It has a fire resistance of at least 2 hr.

5. *Fire Wall.* A fire wall is a wall of brick or reinforced concrete which subdivides a building or separates a building to restrict the spread of fire. It starts at the foun-

dation and extends continuously through all stories to and above the roof, except where the roof is fireproof or semifireproof, and the wall is carried up tightly against the underside of the roof slab. The height of such a wall above the roof should be 3 ft (parapet) and be properly coped. Wall thickness should be 12 in. (insurance standard).

6. *Parapet.* Parapets are required on all walls with the following exceptions:

Walls connecting with roofs of fireproof construction or semifireproof construction

The adjoining wall of a building, the roof of which is 3 ft lower than the roof or any opening in the side wall of the building adjoining or adjacent to such wall

Walls facing on a street having a width of 50 ft or more

Walls of a building that is 50 ft or more distant in all directions from property lines and from other buildings on the same property

Walls of a detached dwelling or building not exceeding 1,000 sq ft in area

Walls of a building where the roof has an angle of more than 20 deg with the horizontal

On 8-in. walls of dwellings, parapets should be 2 ft. On 12-in. walls of dwellings, parapets should be 3 ft. A parapet is not only a separation but also acts as a barrier for use by fire departments.

7. *Wing Wall.* If a building has frame sides, the fire wall should extend 5 ft beyond the frame outside wall. This is termed a "wing wall."

If a wall is of brick but with wood cornice and roof, the fire wall should be corbelled out at the eaves and extend 3 ft above the roof. Exterior walls at intersections with fire walls should be blank for a distance of 20 to 30 ft. If any windows are in exterior walls, these windows should be protected by wired glass in metal frames for 20 ft on each side of the wall. This also applies to roof structures. A fire wall is a special device and should have special consideration.

Openings in Fire Walls—Fire Doors. Openings in fire walls should be protected in a standard manner. The protection of such openings can become valueless unless the protective device is designed to cope with the heat of combustion which may be generated on either side of the wall. Flash fires and radiated heat will pass through openings too fast for ordinary devices.

For doorways, the standard calls for two class A fire doors, one on each side of the wall. This may be modified in the case of boiler and engine rooms of fireproof construction. Class A doors are three-ply wood with a lock-jointed tin envelope; class B doors are two-ply wood with a lock-jointed tin envelope.

The size of openings for standard doors, owing to expansion and other fire effects, shall be limited to 120 sq ft for class A three-ply doors and 80 sq ft for class B two-ply doors. Door laps should be 5½ in.

All sills, styles, and lintels should be of incombustible material. Fire doors should be automatic in operation and also equipped with devices for quick closing in case of flash fires or radiated heat. The attachment of thermostatic or rate-of-rise devices is desirable where conditions demand quick action.

Fire doors are constructed of sheet metal, tin clad, rolling steel, steel plate, and Kalamein Their fire rating is set by the rating bureau having jurisdiction.

Fire-door failures are due to one or more of the following reasons:

1. Fire doors not on each side of wall, thus not providing two lines of defense with air space between
2. Fire doors substandard
3. Poor hardware
4. Fire doors not closed owing to automatic-action failure, blocking, or human element
5. Combustible sills, styles, and lintels
6. Fire intensity too severe
7. Explosives
8. Fire travel too fast for automatic devices
9. Lack of care and maintenance
10. Dry rot

Belt Holes. Belt holes should be protected by standard tin-clad shutters and be equipped with automatic closing devices similar to doorways. Pipe holes should be protected by metal thimbles and collars extending at least 1 in. on each side of wall.

Piers and Wharves. Since piers and wharves are similar to buildings and incorporate flue action in their design, the areas of such structures should be restricted. Transverse fire walls should be installed at intervals of 300 ft, or to restrict the area of one fire division to 5,000 sq ft. Fire walls should extend through the pier deck and continue in the substructure to low water.

Vertical Separation. If a building is more than one story in height, the problem becomes one of superimposed horizontal area. The possibility of damage and destruction due to flue action, converted heat and fire, and smoke and water seepage are increased also (see Table 5-18).

Vertical Openings. Vertical openings consist of elevators, stairways, chutes, conveyors, pipe holes, pipe shafts, ventilating shafts, ramps, and fire towers.

1. *Elevator.* An elevator is a device within or in connection with a building that is used for carrying persons or things upward or downward. Elevators vary in size and are subject to much activity in an industrial establishment and therefore should have special treatment (see American Standard Safety Code for Elevators). The walls of an elevator should be of brick of not less than 4 in. or equivalent thickness (increased for height), extending above the roof not less than 3 ft, and having a fire-resistive rating of not less than 2 hr. The enclosure at the top should have skylights of $\frac{1}{8}$-in. wired glass or plain glass covered by a screen of No. 12 wire set 1 ft above the skylight and extending on all sides at least 1 ft. Windows in these enclosures, not less than 2 ft above roof, may be of plain glass unless the window is within 10 ft of a lot line. In that case it should be wired glass. If the elevator does not extend to the bottom of the building, it should be enclosed with the same construction as the lowest floor.

2. *Stairways.* A stairway consists of one or more flights of stairs and the necessary landings and platforms connecting them to form a continuous and uninterrupted passage from one story to another in a building or structure. There are many varieties of stairways, and standard treatment is difficult, except that the standards for elevator shafts also apply to stairways. All stairways should be of incombustible material and enclosed by wired glass in metal frames or fire-resistive partitions to suit the severity of the occupancy.

3. *Chutes or Conveyors.* Chutes or conveyors that cannot be treated in a standard manner may be treated by installing metal traps or flaps.

4. *Pipe Holes.* Pipe holes should be made as small as possible and closed off by close-fitting incombustible material at the ceiling and floor lines. Metal sleeves should extend 6 in. above the floor if the floor is waterproofed.

5. *Ventilating Shafts.* Ventilating shafts should be constructed of incombustible material, and windows in the shafts should be of wired glass in metal frames.

6. *Ramps.* Ramps present the same problem as any vertical opening and should be treated as such if possible.

7. *Fire Tower.* Smokeproof Tower. Philadelphia Tower. The enclosing walls of a fire tower should be of approved masonry or reinforced concrete, and there should be no openings in such walls, except for the necessary doors and windows. A fire tower should have a roof of fire-resistive construction. The fire tower should extend to the street level. Self-closing fire doors, swinging in the direction of travel from the building to the fire tower, should be provided at both building and fire-tower ends of the balcony or vestibule. Fire towers should be accessible and should never be used for storage. The doors should always be kept free (see Fig. 5-4).

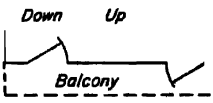

Fig. 5-4. Smokeproof tower.

Frame construction should have the end of wooden transverse beams or joints resting on masonry walls cut to a bevel or chamfer of 3 in. in depth so that release of beam will not destroy the integrity of the wall (see Fig. 5-5).

Water Seepage and Drainage. Water from broken pipes and sprinkler ruptures caused by freezing and breakage and water for extinguishing fires causes great damage, and its elimination by gravity is essential, particularly where floors have been waterproofed. Water is not only destructive because of its wetting quality, but it will be absorbed by stocks, even though they are skidded. Also, its weight (62.5 lb per cu ft) will increase the dead load and cause collapse of building.

*Scuppers.** The installation of scuppers is essential to permit the water to drain off, and floors should be pitched slightly from the center to the outer wall to assist the flow. A scupper is a cast-iron assembly set in the wall at floor level, with an opening 4 by 4 in. closed by a hinged flap at the outer opening. Care should be taken to keep scuppers clean and free from obstruction of any kind. Birds will make nests in them if flaps are not seating properly.

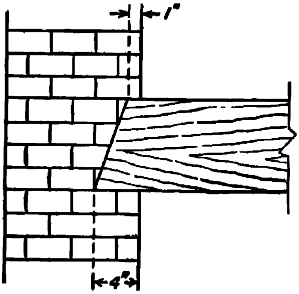

Fig. 5-5. Self-releasing beam.

For floor areas of 500 sq ft or less use two scuppers.

For floor areas of 750 sq ft or less use three scuppers.

For floor areas of 1,000 sq ft or less use four scuppers.

For floor areas over 1,000 sq ft additional scuppers should be used, one to 500 sq ft of area in hazardous locations; one to 1,000 sq ft of area in moderate locations; and one to 2,000 sq ft of area in ordinary locations.

Two scuppers are needed to discharge water from one 2½-in. hose stream (250 gpm).

Table 5-22. Discharge from a 4- by 4-in. Scupper, by Test

Depth of water, in.	*Discharge*
	33
	71
	132
	188
1 4 (top of opening)	210
5	218
	245

FLAMMABLE LIQUIDS AND GASES

Fire prevention, as applied to flammable liquids, gases, and dusts, is a study that must take into account the physics of the materials. Complete equilibrium as to temperature and pressure, as in closed containers, can be used as a basis, for it is the disturbance of the equilibrium which presents the problem in the control of these materials. To enumerate all the factors involved in this study is beyond the limits of this chapter, but certain features are presented which will be of initial value to those in plant management.

Flammable Liquids

There is no sharp line of demarcation between a liquid and a gas. Any liquid at a high temperature and low pressure becomes a gas. Any gas at a sufficiently low temperature and high pressure becomes a liquid. The essential physics of flammable liquids is to maintain the equilibrium of the material at atmospheric pressure. This requires the exclusion of sources of ignition, the prevention of evaporation in enclosed spaces or in proximity to combustible material, and the protection of liquids or gases in closed containers from fire heat or the sun's rays which may cause expansion of the contents and the bursting of the container, resulting in widespread fire. Heat is absorbed when a liquid is evaporated to form a gas. Heat is given off when a gas is condensed to a liquid.

In general, the rate of diffusion of liquids and gases varies with the density. Vapor density is the relative density of the vapor as compared with air, a figure less than 1 indicating a vapor lighter than air, a figure greater than 1 indicating a vapor heavier

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* Scuppers of larger dimensions can be used, but the above requirement will suffice to cover a normal condition.

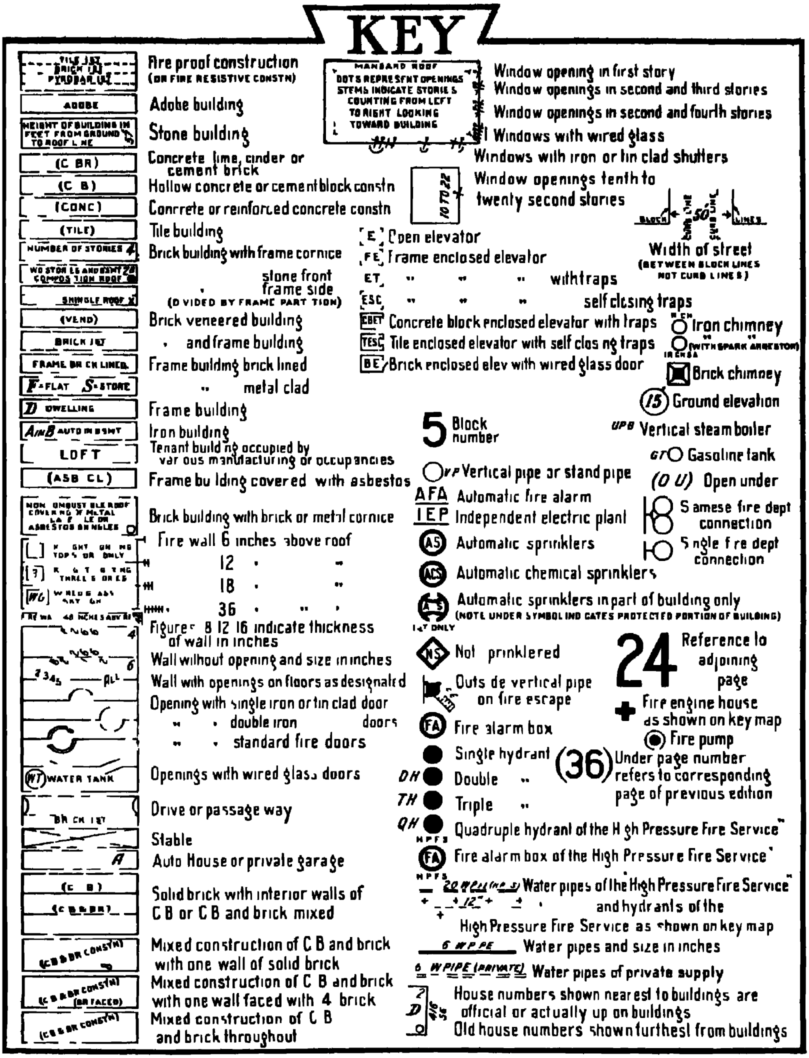
KEY
Fire proof construction (or fire resistive constn)
Adobe building
Stone building
Concrete lime, cinder or cement brick
Hollow concrete or cement block constn
Concrete or reinforced concrete constn
Tile building
Brick building with frame cornice
stone front
frame side
(divided by frame partition)
Brick veneered building
and frame building
Frame building brick lined
metal clad
Frame building
Iron building
Tenant building occupied by various manufacturing or occupancies
Frame building covered with asbestos
Brick building with brick or metal cornice
Fire wall 6 inches above roof
12
18
36
Figures 8 12 16 indicate thickness of wall in inches
Wall without opening and size in inches
Wall with openings on floors as designated
Opening with single iron or tin clad door
double iron doors
standard fire doors
Openings with wired glass doors
Drive or passage way
Stable
Auto House or private garage
Solid brick with interior walls of C B or C B and brick mixed
Mixed construction of C B and brick with one wall of solid brick
Mixed construction of C B and brick with one wall faced with 4 brick
Mixed construction of C B and brick throughout
Window opening in first story
Window openings in second and third stories
Window openings in second and fourth stories
Windows with wired glass
Windows with iron or tin clad shutters
Window openings tenth to twenty second stories
Width of street (between block lines not curb lines)
Open elevator
Frame enclosed elevator
with traps
self closing traps
Concrete block enclosed elevator with traps
Tile enclosed elevator with self closing traps
Brick enclosed elev with wired glass door
Iron chimney
Brick chimney
Ground elevation
Block number
Vertical steam boiler
Gasoline tank
Vertical pipe or stand pipe
(O U) Open under
AFA Automatic fire alarm
IEP Independent electric plant
Siamese fire dept connection
Single fire dept connection
Automatic sprinklers
Automatic chemical sprinklers
Automatic sprinklers in part of building only
Not sprinklered
Reference to adjoining page
Outside vertical pipe on fire escape
Fire engine house as shown on key map
Fire alarm box
Fire pump
Single hydrant
Double
Triple
Quadruple hydrant of the High Pressure Fire Service
Under page number refers to corresponding page of previous edition
Fire alarm box of the High Pressure Fire Service
Water pipes of the High Pressure Fire Service and hydrants of the High Pressure Fire Service as shown on key map
Water pipes and size in inches
Water pipes of private supply
House numbers shown nearest to buildings are official or actually up on buildings
Old house numbers shown furthest from buildings

Fig. 5-6 Signs and symbols used on insurance maps

Color Legend

Yellow	frame	Green (olive)	fire-resistive
Pink	brick	Drab	adobe
Light blue	concrete	Pink-yellow center	brick veneer
Lighter blue	cement block	Pink-blue center	brick, concrete-block lined
Gray	iron-clad		

Table 5-23

Material	Specific gravity	Lb/gal
Alcohol ethyl	0.833	6.95
Benzene	0.88	7.3
Butane	0.585	5.
Crude oil	0.82–0.97	6.85–8.11
Cylinder oil	0.91–0.97	7 58–7.93
Ethane	0.446	3 71
Diesel oil	0.90	7 51
Ethyl fluid	1 71	14.1
Fuel oil	0 96	8.0
Gasoline (motor fuel)	0.71–0.82	5.93–6.86
Gas oil	0.83	7
Kerosene	0.81	6.75
Lubricating oil, light	0.86–0.90	7.17–7.51
Heavy	0.89–0.93	7.41–7.76
Mercury	13 5939	114.
Methyl alcohol	0.791	6 6
Naphtha	0.729	6.07
Propane	0.509	4 24
Water, at 60°F	1.	8.34

than air. Vapor density is one-half the molecular weight. Vapor density of a gas is the ratio of the weight of the gas to the weight of an equal volume of hydrogen. Molecular weight (mol wt) of a gas is equal to twice its vapor density.

Specific Gravity. For liquids, specific gravity is based on the specific gravity of the liquid tested at 60°F as compared with water at 60°F.

Degree Baumé. According to the U.S. Bureau of Standards degree Baumé is as follows:

For liquids heavier than water, degree

$$\text{Baumé} \quad 145 - \frac{145}{R}$$

For liquids lighter than water, degree

$$\text{Baumé} \quad \frac{140}{R} - 130$$

R is the ratio of the density of a liquid to that of distilled water, both taken at 60°F.

Table 5-24. Thermal Expansion of Flammable Liquids

Liquid	*Per cent increase in volume/100°F rise*
Acetone	8.5
Amyl acetate	6.8
Benzol-benzene	7.1
Carbon bisulfide	7.
Ether	9.8
Ethyl acetate	7.9
Ethyl alcohol	6 2
Fuel oil	4.
Gasoline, ordinary	6.
Natural	8
Liquid petroleum gas, butane, propane	9
Methyl alcohol	7 2
Toluol (toluene)	6.3

Table 5-24*a*. Calorific Value of Liquids and Gases

Petroleum products	*Btu/lb*
Gasoline	19,800–20,520
Crude and fuel oil	18,610–19,710
Kerosene	19,710–19,870
Coal-tar oil	17,890–18,400
Gas oil	19,260–19,580
Benzene	18,050
Carbon disulfide	6,130
Ethyl alcohol	19,930
Ethyl ether	22,000
Ethylene	21,340
Methyl alcohol	9,550
Naphthalene	17,320
Gases	*Btu/cu ft*
Natural	710– 2,250
Oil	510– 800
Coal	450– 670
Producer	100– 180
Water	300– 670
Hydrogen	61,060
Acetylene	20,750

Hazardous Locations. A hazardous location is defined as one in which there is present any highly ignitable material, including gases, liquids or dusts. The analysis of the existing or contemplated process or mechanical operation to ascertain whether any hazard of fire or explosion may be present calls for the application of standard treatment set forth* for ameliorating these conditions, including the mechanical equipment such as electric motors, ventilation, grounding, and explosion venting.

Under the National Electrical Code and Underwriters' Laboratories, hazardous locations are defined as shown in Table 5-25.

Table 5-25. Hazardous Locations

National Electrical Code	Underwriters Laboratories	Hazardous locations
Class 1	Group A	Atmospheres containing acetylene
Class 1	Group B	Atmospheres containing hydrogen or gases or vapors of equivalent hazard such as manufactured gas
Class 1	Group C	Atmospheres containing ethyl ether vapor
Class 1	Group D	Atmospheres containing gasoline, petroleum, naphtha, alcohol, acetone, lacquers, solvent vapors and natural gas
Class 2	Group E	Atmospheres containing metal dusts
Class 2	Group F	Atmospheres containing carbon black, coal, or coke dust
Class 2	Group G	Atmospheres containing grain dusts
Class 3		Locations handling ignitible fibers, combustible flyings
Class 4		Locations where easily ignitible combustible fibers are stored

Only liquids having a flash point lower than 200 F are generally termed "flammable," but any combustible liquid, when heated to above its flash point, will produce flammable vapors, e.g., heavy fuel oil when heated above 300°F may release vapors as flammable as gasoline at its flash-point temperature, which is about 0°F (see Table 5-27). All combustible liquids, therefore, must be treated as hazardous with special treatment given to those of a density of 1.25 or lower.

Table 5-26. General Classification for Flammable Liquids*

Class	*Numerical scale*
Ether†	100-110
Gasoline	90-100
Kerosene‡	30-40
Paraffin Oil§	10-20

* Underwriters Laboratories.

† Pure ether is rated 100.

‡ A standard kerosene of 100°F (closed cup) is rated 40.

§ A paraffin oil of 440°F flash (closed cup) is rated 10.

Ventilation

Ventilation may be either natural or artificial. Natural ventilation has the advantage of being in continuous operation and is not dependent upon manual starting or on power supply. For liquids heavier than air, the floor venting should be for a height of 7 ft and may consist of pipes set in the enclosing walls at floor level, pointing slightly downward and venting to the outer air, and other practical apertures. For liquids lighter than air, the venting should be at the ceiling and roof and above 7 ft from the floor for convection currents pushing upwards. Pivoted windows without fixed latches are a desired type of ventilation medium. Mechanical ventilation is more positive and should be used where operations are extensive. It does not depend on temperature and wind conditions. Mechanical ventilation should follow the principle noted above and be figured on the rate of diffusion of the vapors, e.g., 1 gal of solvent evaporated in 2,300 ft of air.

Explosionproof Electrical Equipment (see Motors). Since it is impractical to construct airtight enclosures for motors and other electrical equipment, it is necessary so to construct the enclosure that, if gas or vapor within it is ignited, the resulting flame is prevented from propagating to the outside surrounding atmosphere. This is a fundamental requirement for motors and other electrical equipment of the explosionproof type. The design of equipment to meet this requirement is based on the principle that the propagation of flame through an opening is prevented when the dimensions of the opening are within certain narrow limits.

Explosion pressures of many vapors or gases show wide differences. Explosions

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* National Fire Protection Association.

Table 5-27

Liquids	Flash point °C	Flash point °F	Apparent ignition temperature °C	Apparent ignition temperature °F	Classification	Volatile	Mean iodine value, oxygen absorption (spontaneous ignition)	Fermentation value
Acetate, amyl	25	77	399–427	750–800	55–60	Yes	77	
butyl normal	29	84	371–440	700–825				
ethyl	– 2	28	427–496	800–925				
methyl	– 10	14	454–496	850–925				
Acetic acid, glacial	42	107	427–496	800–925				
Acetic anhydride	53	127	315–357	600–675				
Acetone	– 16	3	538–566	1000–1050	Gasoline, 90	Extremely	90	
Alcohol, amyl (fuse oil)	38	100	371–385	700–725	Kerosene, 40	Yes	76	
iso	45	114	343–385	650–725	Kerosene, 35–40			
Sec	34	94	343–385	650–725				
butyl, iso	31	88	371–427	700–800				
normal	38	100	343–385	650–725	Kerosene			
denatured	16	61	399–432	750–810	Ethyl alcohol	Yes	80	
ethyl	12	54	371–427	700–800	70	Yes	78	
methyl	11	52	427–482	800–900		Yes	85	
propyl iso	16	61	399–454	750–850	Ethyl alcohol, 70			
normal	25	77	371–427	700–800	Amyl acetate, 55–60			
Aniline	84	138	538–607	1000–1025			65	
Aniline dimethyl	74	165	371–399	700–750				
Benzene		– 5				Extremely	95–96	
Benzaldehyde	64	147	192	377				
Benzol	– 20*	– 4*	538–571	1000–1060	Gasoline, 95–100	Extremely		
Bone oil		490				No	25	Low
Carbon disulfide	– 30	– 22	100–106	212–223	100+	Extremely	100	
Castor oil		450				No	10	Low
Coal tar		– 16						
Cocoanut oil, refined		548				No	93	Very high
Corn oil		490				No	69	Very high
Cottonseed oil	252	–486	343–427	650–800	Spontaneous ignition	No	62	High
Cylinder oil, mineral		575				No	1	
Ether, ethyl	– 45	– 49	180	356	100	Yes		
sulfuric		– 49				Extremely	97	
Fuel oil		150				Yes	83	
Fuel oil		122				Yes		
Furfural	61	143	315–357	600–675				
Gasoline	– 43	– 45	280	536		Extremely	95	
Glycerine	176	350	343–399	650–750		No	5	
Hexane	– 26	– 15	260–296	500–565	Gasoline 90–95			
Kerosene		110				No	70	
Lard oil, commercial	184	363	343–399	650–750		No	52	High
pure		494				No	19	Low
Linseed oil, boiled	206	–403	343–399	650–750	Spontaneous ignition		75	
raw	222	–432	343–399	650–750	Spontaneous ignition	No	98	High

Table 5-27—(*Continued*)

Liquids	Flash point		Apparent ignition temperature		Classification	Volatile	Mean iodine value, oxygen absorption (spontaneous ignition)	Fermentation value
	°C	°F	°C	°F				
Lubricating oil, mainly mineral		350				No	12	
Menhaden oil		435				No	85	Extremely
Methyl salicylate	104	−219	454–510	850–950				
Mustard oil, wild		566				No	67	High
Naphthalene		176				Decidedly	85	
Neat's-foot oil		470				No	15	Low
Nitrobenzene	92	198	482–510	900–950		Moderately	60	
Oleo oil		598				No	19	Low
Olive oil	252	485	343–427	650–800	Spontaneous ignition	No	13	Low
Palm oil	162	323	315–371	600–700	Spontaneous ignition	No	40	Extremely high
Palm-kernel oil		398				No	33	Very high
Peanut oil		504				No	18	High
Perilla oil		522				No	72	High
Petroleum crude		− 1				Extremely	93	
Ether	− 56	− 69	329	624	Gasoline, 95–100			
Pyridine	23	74	482–593	900–1100	Amyl acetate	Yes		
Rape oil, pure		465				No	58	Low
Red oil		372				No	50	High
Rosin oil		266				No	76	
Solvent naphtha		60				Yes	90	
Soybean oil		566				No	70	High
Sperm oil		466				No	30	Low
Tallow		509				No	22	Low
Tallow oil		492				No	26	Low
Toluene	9	48	566–607	1050–1125	75–80		92	
Toluidine	94	202	482–538	900–1000		Moderately	64	
Toluol		46				Extremely	92	
Tung oil		552				No	71	High
Turpentine		93				Yes	85	
Whale oil		476				No	77	Very high
Xylene	30	86	482–621	900–1150				
Xylidine		206				Moderately	63	
Xylol		80				Yes	88	

* Flash point is subject to an error of plus or minus 5°C.

of acetylene, for example, may be weak, developing little pressure, or violent, developing a high pressure. If explosive mixture is in a state of mechanical agitation or turbulence at the time of ignition, a higher explosion pressure is developed. It has been found that mixtures containing approximately 1.8 per cent by volume of gasoline vapor in air are much more likely to produce discharge of flame at the joints or other openings than mixtures of the order of 2.2 per cent by volume.[7] It is difficult to make a standard to fit all conditions, so tests should be made and equipment approved for the hazardous location in which it is to be used.

Explosions. The violence of an explosion depends upon the concentration of a vapor with air, and the explosion range is determined by this concentration. For gasoline these proportions range from slightly over 1.4 to about 6 per cent by volume, or a range of 4.6. Therefore, 1 gal of gasoline completely vaporized and mixed with air will form an explosive mixture in volume of from 500 to over 2,000 cu ft.

Table 5-28. Explosion Pressures for Gasoline Vapor and Air Mixture

Per cent of vapor in air by volume	*Pressure, psi*
1	56
1½	65
1¾	73
2	83 4
2¼	100
2½	90
2¾	85
3	68
3½	50
4	30

The curves for vapors from other flammable liquids are similar in form, although the explosion limits vary considerably.

To avoid the possibility of explosions in the use of liquids for dry cleaning or cleaning and washing of parts in factories and garages, the following mixtures are suggested:

Seventy per cent carbon tetrachloride plus 30 per cent or less of gasoline. (This mixture must be tested at intervals, as the carbon tetrachloride will evaporate faster than the gasoline.)

Three parts ethyl dichloride plus 1 part carbon tetrachloride is a safe mixture. Other solvents of low flammability are trichlorethylene, tetra chlorethylene, Stoddards Solvent, and B/L spirits.

Explosion Venting.[1] Properly designed and proportioned and properly distributed vents will release explosion pressures without structural damage. The diagonal scoring of double-strength clear-glass window lights on the outside with a diamond, starting the scoring about two inches from the corners and leaving a 2-in. gap in the center, reduces the strength of the lights to about one-fifth so that the scored glass will probably be blown out without severe damage to the steel frame.

The required ratio of explosion-vent area depends upon the strength of the structure. For reinforced concrete (heavy), provide 1 sq ft for every 80 cu ft of volume. For reinforced concrete (light), brick plank on timber construction, provide 1 sq ft for every 65 cu ft of volume. For light construction provide 1 sq ft for every 50 cu ft of volume. These requirements are for dusts of grain-dust characteristics. Average heat content of flammable liquids is 19,000 Btu per lb. Average thermal expansion of flammable liquids is 4 to 9 per cent increase in volume per 100°F temperature rise.

Oil Burners

The primary hazard of the oil burner is the discharge of unburned oil into a hot firebox, where it may vaporize and form an explosive mixture. In domestic oil burners, the danger is faulty ignition or formation of carbon on ignition points. In industrial plants, burners may be turned off while the firebox is still hot, and oil continues to flow. The installation of oil burners in dwellings and mercantile and industrial properties where the chimney of flue construction has not been designed for this type of fire heat calls for rigid inspection to determine heat radiation which may expose combustible material.

Solvents

Any liquid with a flash point under 100°F may be classed as a solvent. In all mixtures where solvents are used, the same precautions and standards must be used as for flammable liquids.

Table 5-29

Fuel oils	Flash point, deg F	Ignition point, deg F	Extinguishing mediums (Factory Mutual)*
Fuel oil 1 (similar to kerosene)	100–165	490	2, 3, 4
Fuel oil 2	110–190	494	2, 3, 4
Fuel oil 3	110 230 125 200 110 230	498	2, 3, 4
Fuel oil 4	Not used		
Fuel oil 5 (industrial type)	130+ 150+		1, 2, 3, 4
Fuel oil 6 (industrial type)	150+	765	1, 2, 3, 4

* Extinguish mediums are (1) water spray or fog or automatic sprinklers, (2) foam, (3) CO_2 and (4) dry chemicals.

Notes. Fuel oils 5 and 6 are for industrial types of oil burners and are used with live steam and preheated oil for atomization. They may involve large storage tanks under and aboveground. Fuel oils 2 and 3 are for domestic use. Tanks for 550 gal are permitted indoors.

Solvent Recovery. Solvent-recovery processes involving materials in the liquid or vapor state are unusually hazardous and should be isolated in detached buildings. The recovery of the solvent from the vapor may be accomplished in three general ways.

1. Adsorption by materials such as activated carbon or silica gel, vaporization by means of steam, followed by condensation, by cooling, with subsequent recovery of the solvent by decanting the separated layers of condensed steam and solvent.

2. Dissolving the vapors in suitable solvents in scrubbing equipment followed by separation in distillation apparatus.

3. When vapors are in high concentration, they may be recovered by cooling either with water or refrigeration equipment or by first compressing and later cooling. In either case the solvent is obtained directly without further separation or purification by distillation.

Equipment. Although differing considerably in detail, solvent-recovery equipment may be classified into two general groups according to fire hazard.

1. The adsorption and scrubbing types, which involve vapor mixtures of low concentration, generally below the lower explosion limits.

2. The direct-condensation types, which involve much richer mixtures, so that the atmosphere is within or must pass through the explosive range whenever operations are started or stopped.

Low-vapor-concentration Methods. In the first group, particularly the adsorption method, the higher the vapor concentration, the lower the capital and operating costs and the more efficient the recovery. This involves a small margin of safety, and such operations can be permitted with every reasonable safeguard against explosion, including ample explosion vents, and should be under continuous supervision and control of vapor conditions by a reliable vapor indicator.

High-vapor-concentration Methods. Solvent recovery from concentrated vapor mixtures by cooling or by compression and cooling is especially hazardous because explosive conditions are encountered whenever the equipment is started, when it is shut down for repairs or for any other reason, and whenever air enters owing to some defect or breakdown in the equipment. Inert gas is commonly used to keep air out of the equipment.

Distillation. Separation or purification of the recovered vapors by distillation is generally accomplished by means of closed stills heated by steam and operated at low pressure or under a vacuum. The vapor is usually recovered in a water-cooled condenser and conducted directly by gravity to the solvent storage tanks. The danger from distillation equipment is mostly from possible explosion when the equipment is opened for repairs or inspection when it contains vapors mixed with air. Other dangers are from the bursting of the vaporizing tank owing to neglect or failure of auxiliary devices and from liquid escaping from glass gauges. Solvent stills should be equipped with relief valves set well within safe working pressure limits and piped outdoors. All solvent-recovery systems should be isolated and ventilated; all possible sources of ignition should be removed; vaporproof globes should be installed on all lights; and nonferrous metals and tools should be used around the equipment.

Dip Tanks. Painting or coating by dipping objects in tanks of paint, varnishes, or lacquers involves the hazard of exposed surfaces of flammable liquid and the evaporation of the solvents. This also applies to quenching tanks. Small dip tanks can be protected by automatic covers held open by fusible links or thermostats. Covers are not effective where they interfere with stock or stock conveyors. Drainboards should be treated as part of the dip-tank operation. Overflow pipes of such capacity as to drain the liquid without clogging should be provided on all dip tanks. Dipping operations in connection with large units and involving drying ovens are especially hazardous operations and call for special treatment. Fire protection for dip tanks and drainboards can be either foam, CO_2 gas, water spray, or automatic sprinklers. Water spray of high velocity will cool but will also emulsify the oil-reducing salvage. All extinguishing agents can be applied automatically. Quenching oils have a flash point of approximately 365°F. These oils have a good salvage value if allowed to settle for 24 or 36 hr after a fire.

Spray Booths. Spraying operations are hazardous because of the volume and concentration of the work and because spraying produces a residue of flammable character. The solvents used generally are highly flammable unless the spraying medium is a paint not using those solvents of greater volatility than kerosene. Spray operations vary in size from one booth to large assemblies which present special hazard conditions. A single booth may become dangerous because of its use in congested areas or hazardous locations other than that created by the booth itself.

The necessity of elaborate ventilating systems and special lighting arrangements add to the hazard of spraying. A separate exhaust duct should be provided for each booth, leading by the shortest route to the outside of the building. If more than one booth exhaust is connected to a manifold exhaust, this manifold should be treated as part of the entire installation. Exhaust fans should be of the nonsparking (nonferrous) type. Wooden fan blades can be used in emergencies. Motors should be located outside the booth, with the fan drive by shaft or enclosed belt or chain extending through the wall of the booth. Explosionproof motors are no exception to this rule, as they may become coated with spray residue, reducing the cooling effect. Lamps should be of the vaporproof or explosionproof type and not placed where they can be contacted by the spray or broken. Booths should be cleaned frequently, the scraping being done with nonferrous tools, or cleaned with a solvent of no greater flammability than kerosene. Most spray residue, other than ordinary paints, if not thoroughly removed from the booth and if not disposed of, is subject to spontaneous ignition. The use of baffles in spray booths calls for frequent change and cleaning. Water-type spray booths, where flammable liquid spray passes through water to collect residue, should be treated as above.

Location. Spraying operations should not be carried on above large or susceptible stock piles or equipment. Large quantities of water are liable to be used in extinguishment of fire, and such materials would be greatly damaged if floors were not watertight. Spraying assemblies and the rooms in which they are located should be protected by automatic sprinklers.

Transportation. The Interstate Commerce Commission (ICC) has control over shipments of flammable liquids, gases, and explosives. The ICC issues a standard table showing substances that cannot be shipped together. The character of substances is designated by a system of labels as follows:

Red—Flammable liquids with flash points of 80°F or lower
Yellow—Flammable solids (except explosives) and oxidizing materials
White—Corrosive liquids (mineral acids)
Green—Nonflammable compressed gases
Explosives—Marked Explosives

This color scheme is also used within the industry to designate piping conveying hazardous liquids.

Tank-car Shipments. Tank-car shipments are classified in accordance with the vapor pressure of the liquid by the Association of American Railroads, with different tank construction for each class. Vapor pressure is determined by the ASTM, using the Reid Method.

Table 5-30. Reid's Specifications for Vapor Pressure

(Class 1—less than 16 lb vapor pressure, psi, at absolute 100°F. Class 2—not exceeding 40 lb vapor pressure)

Substance	Vapor pressure	Classification
Acetone	7.6	Class 1
Benzene, 60°Bé	0.5	Class 1
55°Bé	1.3	Class 1
Benzol	3.2	Class 1
Carbon disulfide	11.1	Class 1
Casing-head gasoline	15–40	Class 2
Gasoline	9.5–13.5	Class 1
Ethyl ether	16.8	Class 2
Ethyl methyl ketone	3.5	Class 1
Methyl acetate	7.1	Class 1
Naphtha (petroleum), less than	0.5	Class 1
Naphtha (VMP), less than	1.0	Class 1
Methyl ether (Pressure above 40-lb limit for class 2)	Above 103 lb	
Petroleum ether	16–17	Class 2
Toluol (toluene)	1	Class 1

Flammable Liquid Storage Underground. A tight substantial tank is the first requirement in storing flammable liquids underground. Tanks should be thoroughly coated on the outside with a rust-resisting compound. To minimize the danger of leakage from tanks seeping into near-by cellars and lower-than-grade floors or basements, the distances apply as specified in the flammable-liquid ordinance of the NFPA (see Table 5-31).

Table 5-31. Maximum Capacity of Tanks

Location	Maximum allowable storage	
	Class 1, 2, 3, under 100°F flash point, gal	Class 3, above 100°F flash point, gal
If top of tank is above the lowest floor level of any building that is		
Less than 10 ft away	550	50,000
10 to 20 ft away	2,000	75,000
20 to 25 ft away	5,000	100,000
25 to 30 ft away	15,000	150,000
30 to 40 ft away	20,000	200,000
40 to 50 ft away	50,000	500,000
More than 50 ft away	Unlimited	Unlimited

Flammable Liquid Storage Aboveground. Distances between aboveground storage tanks should be equal to at least the diameter of the larger of the two tanks for tanks of over 100,000 gal capacity. For small tanks see Table 5-32. For outside storage, attention must be given to the slope of the terrain from the tank storage to buildings or merchandise stored in yards.

Table 5-32

Capacity of tank, gal	*Minimum distances between tanks, ft*
0– 18,000	3
18,001– 24,000	5
24,001– 48,000	10
48,001– 75,000	13
75,001–100,000	15

Dikes. Dikes around tanks should be large enough to hold the entire capacity of the tank. Vents of standard capacity and design and flame arresters should be standard equipment.

Standards Applying to Flammable Liquids. Standards concerning flammable liquid set forth by the NBFU are the following:

Dry cleaning
Dip tanks
Paint spraying
Discharging flammable liquids
Containers for storage and handling
Installation of oil burners
Pyroxylin-plastic storage
Pyroxylin-plastic manufacture
Motion-picture film
Photo X-ray film

The National Fire Protection Association publishes the following pamphlets applying to flammable liquids:

Preventing welding fires
Combustible anesthetics
Spray application of flammable finishes
Flammable liquids

Flammable Gases

The hazard of flammable gases is generally similar to that of flammable liquids as previously noted (see gas, natural and artificial). The hazard of gases is the possibility of their escape into the atmosphere to form explosive mixtures with air and possible ignition mediums. This is occasioned mostly by breaks or leaks in piping or containers or the escape of unburned gases in gas-burning appliances. Gases lighter than air present a less difficult problem in ventilation than those which are heavier than air. Gases are more or less toxic and present a health hazard which calls for the supplying of oxygen to relieve asphyxiation. All gases expand when heated or increase in pressure if in a closed container of any design. All equipment designed for the handling or storage of gases should be strong enough to withstand any pressure to which they may be subjected and so located or protected as to minimize the danger of mechanical injury.

Pressure Releases. All containers should be provided with some form of pressure relief to prevent rupture or explosion if exposed to heat, sun's rays, or fire. Pressure releases may take the form of a disk of metal designed to rupture at a given temperature or a fusible plug designed to melt at a given temperature.

NOTE: Scharffenburger Bung release for barrels and small containers.

In larger systems various forms of safety-relief valves are used, including thermostatic alarms. See NFPA *Quarterly*, October, 1943. The relief capacity must be equivalent to the vaporization rate or the volume increase.

The volume of a true gas at constant pressure is directly proportional to the absolute pressure, or at constant volume (as in closed containers) the absolute pressure is proportional to the absolute temperature. The expansion of an unconfined gas lessens the velocity of the gas molecule, releases the kinetic energy, and lowers the temperature of the gas. Compression produces the opposite effect.

Flame Arresters. Flame arresters are required on tank vent pipes to prevent flash backs into the tank when an explosive mixture is present. A wire screen of 40-mesh psi will prevent the passage of flame through small openings. Screens of 40 mesh have a free area of about 35 per cent of their total area. This must be allowed for in the design of vents. Screens have a tendency to clog because of dirt and oily residue. Arresters constructed of banks of parallel metal plates or tubes, having a large surface of metal to dissipate heat, are more effective with openings larger than screens and are less liable to clog.

NOTE: Flame arresters are not required by NFPA.

Atmospheres in Tanks. Tests of atmospheres in tanks may be made by chemical methods or by flammable-vapor indicators These indicators, which are useful not only in tanks but in manholes or in locations where the presence of flammable vapors may be suspected, measure the amount of flammable vapor present and are generally calibrated in terms of per cent of low explosion limits They give different readings for different gases and therefore should be used only within the limits of their suitability. Carbon bisulfide gives low readings and requires special calibration Flammable-vapor indicators must be carefully operated and maintained and should be checked and calibrated periodically to assure correct readings.

A common type of indicator operates by combustion within the apparatus of a sample of the atmosphere in a small enclosure in the presence of an electrically heated resistance element The change in resistance due to the heat of combustion determined by a thermocouple or a Wheatstone bridge is measured to indicate the relative amount of flammable vapor present In another type the amount of vapor is determined by the thermal conductivity of the mixture electrically measured Instruments available for determining the percentage of flammable vapors in the atmosphere are Factory Mutual flammable-vapor indicator, Gasoline vapor indicator, Hall (devised by H H Hall of Standard Oil Company of California), the UCC combustible-gas indicator (Union Carbide & Carbon Company) and the U S Bureau of Mines indicator[9]

Table 5-33. Dangerous Gases Found in Sewers and Manholes*

Gases	Properties	Source
Ammonia	Poisonous	Refrigerator plants
Benzol	Poisonous explosive	Automobiles tanks
Carbon dioxide	Suffocating	Combustion decomposition
Carbon monoxide	Poisonous explosive	Fuel gas flue gas automobile
Ethane	Explosive	Natural gas mfrd fuel gas
Gasoline	Poisonous explosive	Automobiles tanks
Hydrogen	Explosive	Artificial fuel gas electrolysis of water
Hydrogen sulfide	Poisonous	Coal gas decomposition
Methane	Explosive	Nat gas and mfrd gas decomposition
Natural gas	Suffocating	Natural gas mains
Nitrogen	Suffocating	Combustion decomposition
Phosgene	Poisonous flammable	Decomposition combustion
Sulfur dioxide	Poisonous	Burning insulation
Unsaturated hydrocarbons	Explosive	Manufactured fuel gas

* *NFPA Quarterly* April 1930

Flammable gases may be produced in

Oil burners
Industrial furnaces
Range oil burners
Oil-burning stoves
Gasoline stoves
Dry cleaning
Finishing processes
Gasoline engines
Coating and spreading processes
Reclaiming
Solvent recovery
Degreasing
Extraction processes
Rubber cement and varnish mixing
Impregnating under pressure

Manufactured gas contains carbon monoxide Natural gas does not ordinarily contain carbon monoxide

Acetylene Acetylene has a calorific value in Btus of 20,700 per lb

The quantity of calcium carbide[10] permitted to be stored inside a building is 600 lb In a one-story building outside the main buildings, 5,000 lb may be stored

Acetylene may react with copper to produce

Table 5-34. Commercial Gases

Gas	*Calorific value, Btu per cubic foot*
Natural gas	710–2250
Gas oil	510– 800
Coal gas	450– 670
Producer gas	100– 180
Water gas	300– 670

an explosive compound; therefore, iron or steel piping should be used on all installations. Acetylene has the widest explosive range of any gas used in industry, from 2.5 to 82 per cent. Acetylene can explode by decomposition without the presence of air at pressures above 15 psi. The calcium-carbide-to-water type of acetylene-gas generating systems is the best type. Calcium carbide residue must be disposed of in a safe place; its reclamation is said to be a practical process. Calcium carbide fires can be extinguished by CO_2, but the carbide must be removed to avoid reignition due to remaining moisture.

Hydrogen. Hydrogen has a calorific value of 61,060 Btu per lb. Its ignition temperature is from 982 to 1000°F. All piping used in installations where hydrogen is present must be gastight to prevent leakage. Hydrogen evolved from zinc-coating processes can be a source of fire. Battery charging causes the dissociation of hydrogen. Therefore it should not be done within 10 ft of an open flame.

Oxygen. Oxygen under pressure should not be brought in contact with oil, oil lubricants, or greases.

Liquid Petroleum Gases. Liquid petroleum gases have a calorific value of 20,000 Btu per lb. Their thermal expansion is 9 per cent volume increase per 100°F. Butane and propane are heavier than air. Cylinders of such gases aboveground must be kept from freezing.

Nitrocellulose Film. Nitrocellulose film consists of such substances as celluloid, pyralin, fiberloid, xylonite, viscoloid, nixonite, and pyroxylin plastics. The difference between these substances is their nitrogen content, which ranges from 10.5 to 13.5 per cent. Tests made on nitrocellulose film for X-ray purposes, showed gases produced by decomposition to be as shown in Table 5-35.

Table 5-35*

	Per cent by volume
Nitrogen peroxide	6.9-8.9
Nitric oxide	1.4-8.2
Carbon monoxide	47.4-59.1
Carbon dioxide	21.3-24.5
Oxygen	None
Hydrogen	0.9-3.2
Methane	1.0-2.7

* Chemical Warfare Service, United States Army, after Cleveland Clinic Fire, 1929

Note: At high temperatures 1,000 lb = 8,500 cu ft of gas.

Common Refrigerants. Table 5-36 lists most of the substances used as refrigerants and gives data on their flammable and toxic properties. For purposes of comparison, similar data are given on carbon tetrachloride, a commonly used extinguishing agent.

These data are taken from Underwriters' Laboratories Report MH2375 on the Comparative Life, Fire, and Explosion Hazards of Common Refrigerants and from the supplementary Report MH 2630 on dichloromonofluoromethane. For more complete information, see the original reports, which may be purchased from Kinetic Chemicals, Inc., Wilmington, Delaware, for whom they were made.

Other information of interest is given in the following paragraphs, also taken largely from Underwriters' Laboratories Report MH2375.

Ammonia is an irritant gas having a sharp, penetrating odor which gives adequate warning of its presence.

Butane, ethane, and propane have practically no odor and do not give warning of their presence in air in dangerous concentrations. Butane and propane are anaesthetics.

Carbon dioxide is odorless, but its physiological effects when breathed in mild or moderate concentrations serve in a measure as a warning of its presence.

Dichloroethylene is an anaesthetic gas having an odor that resembles chloroform. It does not give adequate warning of danger.

Dichlorodifluoromethane (F-12, Freon) has an odor resembling that of chloroform.

Ethyl bromide does not give adequate warning of its presence in air in dangerous concentrations.

Ethyl chloride is an anaesthetic that gives some warning of its presence, but a person might tolerate exposures to it in dangerous concentrations until helpless.

Methyl bromide has an odor, but it is not distinctly unpleasant and does not give definite warning of its presence in dangerous concentrations.

Methyl chloride is an anaesthetic that does not give definite warning of its presence in dangerous concentrations, but when used as a refrigerant it usually contains substances that give adequate warning

Table 5-36

Name of substance	Flammability	Toxicity classification group No.*
Ammonia	†	2
Butane	‡	5
Carbon dioxide	Nonflammable	5
Carbon tetrachloride	Nonflammable	3
Dichloroethylene	§	4‖
Dichlorodifluoromethane (F-12)	Nonflammable	6‖
Dichloromonofluoromethane (F-21)	¶	Between 4 and 5‖
Dichlorotetrafluoroethane (F-114)	Nonflammable	6‖
Ethane	‡	5
Ethyl bromide	§	4‖
Ethyl chloride	§	Between 4 and 5‖
Methyl bromide	¶	2‖
Methyl chloride	§	4‖
Methyl formate	‡	3
Methylene chloride (dichloromethane)	¶	Between 4 and 5‖
Monofluorotrichloromethane (F-11)	Nonflammable	5‖
Propane	†	5
Sulfur dioxide	Nonflammable	1

* Sulfur dioxide (group 1) has a toxicity that is comparatively high (concentrations of the order of $\frac{1}{2}$ to 1 per cent for duration of exposure of the order of 5 min. will cause death or serious injury). Substances in group 2 are less toxic, in group 3 still less toxic, etc. An idea of the danger from refrigerants in group 5 may be obtained from the fact that carbon dioxide, which is in group 5, has been extensively used in industry and as a fire extinguishing agent with a record that indicates that its actual life hazard under ordinary conditions is small.

† An explosion of ammonia air mixtures of high concentration (16 to 26 per cent ammonia by volume) is possible, but usually under practical conditions combustion of ammonia is slow and weak.

‡ In a class with illuminating gas and the vapor of gasoline.

§ Moderately flammable; much less hazardous than gasoline but more hazardous than ammonia.

‖ These refrigerants are decomposed by contact with flame or very hot surfaces, producing fumes which are toxic and irritating but which give definite warning of their presence before dangerous concentrations are reached.

¶ Fire hazard is very small from a practical standpoint.

Methylene chloride (dichloromethane or Carrene No. 1) has a physiological action which gives some warning of its presence.

Methyl formate is an irritant and gives definite warning of its presence.

Sulfur dioxide is exceedingly irritating to the eyes and throat even in very low concentrations, so that persons breathing it make every effort to escape before serious injury occurs.

In considering flammable gases and vapors, one of the important points is whether they are heavier or lighter than air, and if so, how much heavier or lighter. Table 5-37 gives in one column the vapor density relative to air under the same conditions of temperature and pressure. The vapor density of acetone, for example, is given as 2.0, which means that acetone vapors are twice as heavy as air. That of methane is given as 0.55, which means that methane gas is about one-half as heavy as air.

Gas escaping from a pipe or tank and vapor given off by a liquid, such as gasoline, tend to diffuse in the air in about the same way that a drop of ink in a glass of water diffuses through the water so that all the water becomes colored. Some vapors diffuse very slowly, while others diffuse rapidly. It will be noted from Table 5-37 that ammonia gas diffuses in air about twice as fast as carbon bisulfide and that the lighter the gas, the faster it diffuses. Hydrogen, the lightest of all gases, diffuses about $3\frac{1}{2}$ times as fast as carbon monoxide.

It has been found that flammable gases and vapors will burn or explode only when the proportion of gas or vapor to air is within certain limits, known as the "flammable" or "explosive" limits. Looking, for example, at the figures in Table 5-37 for gasoline, it is seen that gasoline vapor will burn or explode when the proportion of vapor to air is between 1.4 and 6 per cent. This means that, if the proportion is less than 1.4 parts of vapor to 98.6 parts of air or is richer than 6 parts of vapor to 94 parts of air, there can be no burning or explosion. However, with mixtures too

Table 5-37. Some Characteristics of Certain Flammable Gases and Vapors and of Common Refrigerants*

Gas or vapor	Vapor density (air = 1)	Rate of diffusion (air = 1)	Explosive limits in air by volume	
			Lower	Upper
Acetic acid	2.1	0.69	4.0	
Acetone	2.0	0.71	2.0	9.0
Acetylene	0.90	1.05	2.6	82
Alcohol (ethyl alcohol)	1.6	0.79	3.5	19.0
Ammonia	0.59	1.30	16.0	25
Amyl acetate	4.5	0.47	1.1	4.0
Amyl alcohol	3.0	0.58	1.2	5.0
Benzene	3.0	0.58	1.4	5.9
Benzol (benzene)	2.7	0.61	1.4	8
Butane	2.0	0.71	1.6	6.5
Butyl acetate	4.0	0.50	1.7	
Butyl alcohol	2.6	0.62	1.7	18
Carbon bisulfide	2.6	0.62	1.0	50
Carbon dioxide	1.5	0.82	Nonflam	
Carbon monoxide	0.97	1.02	12.5	74
Dichlorethylene	3.35	0.55	5.6	11.4
Dichlorodifluoromethane (F 12)	4.2	0.49	Nonflam	
Dichlorotetrafluoroethane (F 114)	5.9	0.41	Nonflam	
Ethane	1.03	0.98	3.3	10.6
Ether (ethyl ether)	2.6	0.62	1.9	22
Ethyl acetate	3.0	0.58	2.5	11.5
Ethyl bromide	3.76	0.51	6.0	11.0
Ethyl chloride	2.2	0.67	3.7	12.0
Ethylene	0.97	1.02	3.0	34.0
Ethylene oxide	1.5	0.82	3.0	80
Gasoline	3.5	0.53	1.4	6
Hydrocyanic acid	0.9	1.04	5.6	40
Hydrogen	0.069	3.81	4.1	74
Hydrogen sulfide	1.17	0.92	4.3	46
Illuminating gas	0.45	1.24	5.0	31
Kerosene	4.5	0.47	1.1	6
Methane	0.55	1.35	5.6	13.5
Methyl acetate	2.6	0.62	4.1	14
Methyl (wood) alcohol	1.1	0.95	6	36
Methyl bromide	3.3	0.55	13.5	14.5
Methyl chloride	1.7	0.76	8.1	17.2
Methylene chloride (dichloromethane)	2.9	0.58	†	†
Methyl formate	2.1	0.69	4.5	20
Monofluorotrichloromethane (F 11)	4.7	0.46	Nonflam	
Pentane	2.5	0.63	1.45	7.5
Propane	1.5	0.82	2.3	7.3
Propylene	1.45	0.83	2.2	10.7
Sulfur dioxide	2.2	0.67	Nonflam	
Toluol (toluene)	3.14	0.56	1.4	7
Turpentine	4.7	0.46	0.8	
Xylol (xylene)	3.7	0.52	1.0	5.3

* NBFU *Bull* 52

† Practically nonflammable at ordinary temperatures

rich to burn there is a possibility of some portion of the mixture being diluted with additional air so as to bring it within the explosive limits

At or near the explosive limits the explosion will be mild or there will be quiet burning, but between these limits explosions of varying degrees of violence will occur For gasoline the most violent explosion will occur with a mixture of about 2 25 per cent In general, it may be said that the wider the range between the lower and upper explosive limits the greater the hazard of explosion Vapors with a lower limit below 2 per cent have also been shown to have a high hazard even though they have a relatively narrow range

A vapor such as gasoline that is heavier than air tends to settle in low spots and, although gradually mixing with the air, if not affected by drafts will remain in such low spots for a considerable time. Such heavy vapors will flow in a stream along a floor and down stairs or along the ground and, if reaching a source of ignition, will propagate flame back to the source even from considerable distances. In one fire, the vapor was reported to have ignited in this way from a point 162 ft from the source. The vapor in such low spots or in such a stream will frequently be too rich to explode, but even where this is true there will be, above and around the rich mixture, a mixture that is in the explosive range, and as the vapor diffuses a greater amount will come within the explosive range. If ignition is delayed until the vapor is diffused sufficiently to get a flammable mixture over a considerable area, ignition will then result in a serious explosion. The higher the relative rate of diffusion, the more quickly will the scene be set for such an explosion; and the larger the space that contains such a flammable mixture, the more violent will be the explosion.

A gas such as illuminating gas that is lighter than air tends to rise but diffuses fairly rapidly so that in a building the gas would be soon distributed throughout the room or space in which it is discharged. A small gas leak in a room with some ventilation might not be sufficient to produce an explosive mixture, but if enough is liberated, an explosive mixture will be formed over a large area very quickly.

If the temperature of a gas or vapor is higher than that of the air in which it is liberated, the gas or vapor is in effect made lighter and will act accordingly.

Need for Gas Masks in Fighting Fires, Their Value and Limitations[11]

Noxious and poisonous gases of various kinds are encountered in fighting fires. The most common of these is carbon monoxide, an odorless product of combustion which is present in varying degrees depending upon the material and the location of the fire. Cellar and basement fires and those involving oil or material in a compact form are producers of large quantities of carbon monoxide, and as such fires can seldom be ventilated, it is obvious that some form of protection must be provided against these and other gases. One source of other gases is mechanical-refrigeration equipment which is now found in many stores and restaurants and in a great many apartment houses and residences. Many systems contain ammonia or sulfur dioxide, both toxic refrigerants. Although these, through their distinctive odor, give adequate warning of their presence before dangerous concentrations are reached, fire service may require that entrance be made.

Because of the irritant and poisonous nature of the gases occurring during a fire and the further probability of other gases being liberated, no modern fire department can be considered as being properly equipped if it does not have sufficient protective equipment for its men.

Subsequent to the development of gas masks for war work, there has been material improvement in the canister type of mask, which consists of a rubber face covering and a canister so arranged that all the air breathed comes through the canister which contains absorbing substances for removing poisonous gases. Type N, as approved by the U.S. Bureau of Mines, gives protection against all the poisonous gases that firemen are likely to encounter, including carbon monoxide. This mask is sometimes called an "all-service" mask.

All canister-type masks have two important limitations which must be well understood if they are to be used with safety. (1) They are utterly useless in atmospheres deficient in oxygen. This condition is found in tanks that have held readily volatile material, such as gasoline, in sewers and manholes in which gases may accumulate and there is very little ventilation, or where carbon dioxide or other gas has seriously diluted the oxygen of the air. (2) If there are high concentrations of gases, the materials in the canisters are rapidly used up, and considerable heat is developed.

For those infrequent occurrences where a gas mask is not suitable, an oxygen breathing apparatus must be used. This is a self-contained breathing apparatus which supplies oxygen from a small cylinder and which purifies the air breathed.

Modern fire-department practice recognizes that the use of gas masks and oxygen breathing apparatus is an essential feature of effective fire fighting. Its value in preventing injury or death to firemen is primary, but material reduction in fire loss is also possible because of the ability to penetrate to the seat of the fire. Less ventilation of the structure is required where they are brought into use, and rescue work can be performed with masks where otherwise there might be loss of life.

For effective operation two gas masks should be provided for each company, and the members trained in their use. In the larger departments these would be supplemented by oxygen breathing apparatus, carried either by ladder companies or on special apparatus.

The effective use of gas masks and helmets requires training and practice. The men must know how to use them as well as when not to use a gas mask but to use oxygen breathing apparatus.

Inert Gas Systems

Fire and explosion can be prevented by the creation of an atmosphere that will not support combustion by reducing the oxygen content of the normal air which contains 21 per cent oxygen. Any inert gas can be used for this purpose, but consideration of availability and cost limits such use to carbon dioxide, nitrogen, and mixtures of carbon dioxide and nitrogen produced by combustion, as in flue gas, internal-combustion-engine exhausts, and other inert-gas producers.

Table 5-38. Dilution of Atmosphere with Carbon Dioxide*

Substance	*Maximum permissible oxygen, per cent*
Pittsburgh coal dust	16
Pyrethreum flower dust	15.5
Acetone	15
Cotton lint or dust in air	15
Ethyl alcohol	15
Gasoline vapor	15
Methane	14.5
Cork dust	14.1
Wheat, corn, or oat elevator dust	14
Ground-oat hulls	13.7
Ether	13
Hard-rubber dust	13
Wheat starch	12
White dextrine	12
Sulfur	11
Ethylene	10
Carbon disulfide	8
Cotton in bulk to prevent smouldering and reignition	8
Jute	8
Carbon monoxide	5.9
Hydrogen	5.9

Note: If nitrogen is used alone, these percentages can be lowered 10 per cent.
* NFPA Standard.

Extinguishment of Dust Fires. The introduction of inert gas into grinding equipment and transport systems reduces the normal oxygen content to a point at which an explosion cannot occur. A reduction of oxygen to less than 15 per cent is necessary to prevent explosions of carbonaceous dusts and to less than 10 per cent to prevent smouldering fires.[12]

Carbon dioxide, carbon tetrachloride, and water should not be used on aluminum fires. During fire-fighting operations, caution should be exercised in use of heavy streams, as clouds of dust may be set up causing explosions.

DUSTS

Dusts have many states of division which range from particle sizes, which will hardly float in air, to the finely divided state known as "flowers," which will pass through a screen of 400 mesh per sq in. The finer the state of division and the more rapidly and violently the dust is formed, the greater the possibility of a powerful explosion.

Flammability of Organic Dust. The chief reason for the flammability of dust, according to Von Schwartz,[13] is the occlusion of atmospheric air or other gases by the minute particles of dust which act like an absorbent sponge. Laden with gas or vapor, these dust particles form a highly flammable material which ignites with

Table 5-39. Effects of Inert Gases and Low Oxygen Content on Explosibility*

Combustible gas or vapor	Formulas	Per cent of oxygen by volume below which explosions are impossible, at ordinary temperature and pressure			Volumes of inert gas necessary to render one volume of combustible nonexplosive under all conditions of dilution, with air at ordinary temperature and pressure		
		Nitrogen	CO_2	Flue gas	Nitrogen	CO_2	Flue gas
Hydrogen	H.	5.0	5.9	5.1	16.5	10.2	11.0
Carbon monoxide	CO	5.6	5.9	5.6	4.1	2.2	2.4
Ethylene	C_2H_4	10.0	11.7	10.2	15.3	9.0	9.8
Methane	CH_4	12.1	14.6	12.3	6.0	3.2	3.5
Ethane	C_2H_6	11.0			12.8		

* Jones and Perrott U.S. Bureau of Mines

NOTE. The standards for reference from the National Board of Fire Underwriters are the following:
50 Acetylene Equipment
53. Gasoline Vapor Machines
54. Piping and Fittings for City Gas
51 Gas Systems for Welding and Cutting
58 Liquid Petroleum Gases

The standards set up by Pamphlets from the Underwriter's Laboratories include Spontaneous Ignition and Prevention, List of Approved Gas and Oil Appliances

great rapidity forming an atmosphere of hydrocarbons and carbon monoxide that furnishes with the remaining air an explosive mixture, which is ignited and exploded by the flame of the burning dusts.

Consequently, dust explosions occur in two stages—the ignition of the dust particles and the explosion of the resulting and readily formed gaseous products. These phases follow in such rapid succession as to form one operation. Both organic and metallic dusts become far more explosive when mixed with substances liberating oxygen.*

The formation of dust is always accompanied by a liberation of heat (in contrast to gases), and if this heat is confined and is not dissipated, the temperature may reach the ignition point of the material of which the dust is formed.

Dust explosions from shock, percussion, or friction, unaccompanied by the production of sparks, cannot take place.

Dust-explosion Pressures. Tests were made by the U.S. Department of Agriculture to find out dust explosion pressures. The quantity of dust used in these tests was 5,000 mg per liter which corresponds to 500 oz per cu ft. The results of these tests are shown in Table 5-40.

Table 5-40. Dust Explosion Pressures

Type of dust	*Explosive pressure, psi*
Grain dust	37–46
Starches	38–47
Sugars	22–45
Woods	36–44
Sulfur and lime sulfur	31–32
Hard rubber	36–37
Cork	38–40
Metals	3–49
Fertilizers	34–51
Milk powders	31–42
Cocoa	23
Flours	26–42
Meals	21–42
Spices	28–43
Drugs	22–43
Rosins, waxes, soaps	31–42
Coal, carbons	24–48

Metal Powders. Ignitions and explosions of metal powder may occur under the same conditions required to produce explosions of carbonaceous dusts. Metal powder or dust in a quiescent state may be ignited by a spark or the application of sufficient heat to raise the temperature to the ignition point. If undisturbed, the powder will burn quietly, and the fire will spread at a rate depending to some extent on the availability of oxygen to complete combustion. The same sources of ignition are generally capable of starting a dust explosion, but to obtain such a reaction, the powder must be scattered in the air in the proper proportions. It is generally conceded that the finer the dust, the more flammable it is, the smaller the flame or spark required to ignite it, and the longer it remains in suspension in the air where it can be most readily ignited.

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* This is confirmed by Dr. R. V. Wheeler, of the British Coal Mines Committee, Eskmeals. England writing on starch

The most widely used methods of producing metal powders is the milling process, which includes the production of powder in stamps, crushers, rolls, ball, bar, and disk mills, attrition mills, eddy mills, and jet pulverizers. In the last two types of equipment, air currents are employed in the grinding operation.

The Hall process provides for the mixing of a liquid with the material entering the mill to form a slurry or paste and thus eliminate dust ignitions within the mill. If it is necessary to dry or further condition the product after milling, the explosion hazard may be present to some extent.

The Hametag process employs a stream of inert gas to carry the powder away from the mill as it is produced. The purpose of this is to prevent both oxidation and explosions.

It has been estimated that 1 lb of some of the finest metal powders has an expanded surface of about 220,000 sq in.

Screening and packaging operations constitute dust ignition hazards similar to those in other dusty industries employing the same type of equipment, such as screens, bolters, sifters, reels, and dust collectors.

Experiments have shown that flame propagation occurs most readily in dust clouds in which the particles are small, of uniform size, and uniformly distributed. Under such conditions flame velocities of 2,000 to 3,300 fps have been measured, with indications of much higher velocities, even up to 6,000 fps. The ignition temperatures of the metal powders while in a quiescent state were found to range from 410 to 1742°F, 12 out of 16 igniting at temperatures below 932°F. Ignition of a dust cloud usually requires a higher temperature or a larger source of ignition than is necessary to cause inflammation of a quiescent layer of the same material. In a series of tests, it was found that the temperature necessary for ignition of a dust cloud generally ranged from 599 to 1652°F. Pressures produced in metal-dust explosions range from 20 to 72 psi.

Venting by explosion vents or hinged windows in buildings of substantial construction has been installed at the ratio of 1 sq ft of vent to each 25 cu ft of room volume.

In recent tests to determine the efficiency of inert gases to reduce the oxygen content of atmosphere, such as CO_2 and nitrogen used in preventing ignition of carbonaceous dusts or flammable gases, it was found that some of the metal powders could be ignited by sparks in pure carbon dioxide, and for a few powders pure nitrogen provides only a narrow margin of safety against ignition. Magnesium will burn in nitrogen to form magnesium nitride. Helium and argon were found to be more effective than CO_2 or nitrogen. The results of these tests indicate that dependence for protection against dust ignitions in metal powders of atmospheres with reduced oxygen content by the introduction of inert gases should not be accepted unless laboratory tests have demonstrated their effectiveness.

The following recommendations for the prevention of ignition and explosion are not conclusive owing to the lack of factual knowledge relating to metal dust, but some degree of protection can be obtained by their observance:

1. Hazardous processes should be segregated in small detached buildings or separate units with fire walls and fire doors.
2. Vents such as light roofs or large hinged windows should be provided, designed in the proper proportion to release explosion pressure without structural damage.
3. Equipment should be maintained as nearly dust-tight as possible and frequent cleaning should be provided for to prevent accumulation of dust at points where it may be thrown into suspension to form a cloud.
4. All possible sources of ignition should be eliminated by using: (*a*) electrical equipment approved for such locations; (*b*) indirect heating to prevent dust from collecting on heated surfaces; (*c*) ground connections on all equipment to remove static electricity.
5. Inert gas should be used to prevent dust ignitions wherever feasible.
6. Conveying, collecting, processing, and cleaning equipment designed to perform

Table 5-41. Ignition Temperature and Explosibility of Metal Powders with Comparable Data for Several Carbonaceous Dusts

Substance	Ignition temp., °F, of		Maximum pressure, psi	Pressure rise, psi/sec	
	Quiescent powder	Dust cloud		Average	Maximum
Aluminum, atomized	*	1292	58	1,050	2,250
Stamped	1085	1193	62	2,170	5,700†
Antimony	629	779	20	90	150
Cadmium	482	1058	‡		
Chromium	1247§	1652	‡		
Copper	518‖	1292	‡		
Dow metal (magnesium-aluminum alloy)	896	806	56	1,600	2,570
Iron, milled	860	1436	‡		
Reduced	554	599	36	230	430
Iron carbonyl	410	608	31	660	1,200
Lead, atomized	518	1310	‡		
Stamped	572	1076	‡		
Magnesium, milled	977	1004	65	1,600	3,160
Magnesium, stamped	887	968	72	1,450	4,760†
Magnesium and aluminum, milled	878	995	62	2,350	3,000
Manganese	.....	842	31	220	400
Silicon	1427	1742	62	440	1,180
Tin, atomized	806	1166	34	500	850
Titanium	806	896	52	750	1,650
Zinc	896	1112	35	710	1,350
Zirconium	410	¶	42	600	2,570
	Carbonaceous dusts				
Cornstarch	615	716	51	830	1,590
Peat moss	...	887	58	680	1,120
Pittsburgh coal dust	...	1130	40	370	780
Wood pulp	522	1040	53	580	1,000

* Temperature rise in powder at 1292°. No other evidence of ignition up to 1652°.

† This extremely high value should be accepted with caution because of possible errors in determining maximum slope of pressure-time curves.

‡ This powder not ignited by standard spark in Hartman apparatus. Ignition produced by pallet of magnesium and barium peroxide.

§ Reaction noted at 1247°. No other evidence of ignition.

‖ Powder oxidized rapidly at 518°. No other evidence of ignition.

¶ Dust cloud ignited when blown through furnace at room temperature.

Note: In comparing the ignition temperatures for a quiescent powder and a dust cloud, it will be noted that there is no consistent relation between the two figures. The only explanation that can be offered at the present time is that in some cases prolonged heating of the powders on a hot surface produces a condition that permits the ignition at a lower temperature than is otherwise possible. On the other hand, in a moving dust cloud, the powder may come in contact with the heated surface for only a short period. Also, the shape of the powder permits the exposure of thin edges when the dust is in suspension, and thin edges or fine projections can be ignited at a temperature somewhat lower than that required to ignite the product when the particles are packed together.[14]

Bronze dust, if free from aluminum, is practically harmless. Sulfur dust or vapor forms explosive mixtures with air and is hazardous in contact with oxidizing materials. It should be isolated from chlorates and nitrates. [Gunpowder is formed from sulfur, saltpeter, and charcoal (carbon)]. Water spray or fog is effective on sulfur fires.

Table 5-42. Explosion Pressures of Coal, 24 to 48 Psi

Type of coal	Specific gravity	Density	Btu
Anthracite	1.4–1.8	97	
Bituminous	1.2–1.5	84	
Anthracite (Penn.)	....	..	12,500–13,830
Anthracite (semi)	....	..	13,120–13,380
Bituminous	....	..	13,710–14,810
Bituminous	....	..	10,020–14,700
Bituminous (sub)	....	..	8,690–11,140

its function with no possibility of producing metallic sparks or frictional heat capable of igniting the powder being handled should be employed.

7. Fire-fighting facilities of the type found best adapted to the particular needs of the plant should be provided.

Water and ordinary types of fire extinguishers are not effective on metal-powder fires, and in many cases their use will intensify the fire.

Table 5-43

Plastics	*Group*	*Plastics*	*Group*
Cellulose acetate	1	Casein	2
Cellulose acetate, butyrate	1	Vinyl chloride	2
Acrylate	1	Vinyl acetate	2
Polystyrene	1	Copolymers	3
Polyvinyl alcohol	1	Polyvinyl chloride	3
Phenol formaldehyde	2, 3	Cold molding	3
Urea	2		

Table 5-44. Relative Explosion Hazards of Powders Used in the Plastics Industry[1]

Type of powder[2]	Ignition temperature of dust cloud		Minimum energy required for ignition, joules[3]	Minimum explosive concentrat., 0.03/cu ft	Maximum pressure psi[4]	Rate of pressure rise, psi/sec[4]	
	°C	°F				Average	Maximum
Hexamethylene tetramine[8]	410	770	0.01	0 015	64	940	2,570
Shellac, rosin, gum	390	735	0 01	0 015	58	1,240	2,990
Phenolic resins[5]	500	930	0.01	0 025	61	1,370	3,160
Coumarone-indene resins	520	970	0.01	0.015	63	1,370	2,990
Cellulose acetate molding compounds[6]	320	610	0 01	0 025	62	1,180	2,260
Pentaerythritol[8]	450	840	0 01	0 030	65	980	2,170
Cellulose acetates[6]	410	770	0 015	0.035	68	800	1,740
Lignin resins	450	840	0 02	0 040	69	760	2,700
Ground cotton flock[9]	470	880	0 025	0 050	67	870	2,990
Ground wood flour[9]	430	805	0 02	0 040	62	830	2,080
Phenolic molding comp	490	915	0 01	0.030	63	900	2,080
Synthetic rubber	320	610	0.03	0.030	59	740	1,870
Phthalic anhydride[8]	650	1200	0 015	0 015	49	1,270	1,690
Vinyl butyral resin	390	735	0.01	0 020	60	470	1,020
Methyl methacrylate molding comp	440	825	0 015	0.020	57	570	1,200
Urea molding comp	450	840	0 08	0 075	63	710	1,800
Polystyrene molding comp	560	1040	0 04	0.015	50	740	1,640
Ground alpha pulp[9]	480	895	0 08	0.060	60	520	1,450
Urea resins	470	880	0 08	0 070	65	340	850
Polysterene resin	490	915	0 12	0.020	44	350	650
Vinyl resins[7]	550	1020	0 16	0.040	49	250	490
Rennet casein[8]	520	970	0.06	0.045	49	190	500
Vinyl molding comp	690	1275					
Chlorinated paraffin	840	1545					
Asbestos, asbestine, mica	No ignitions obtained in any test; these powders present no dust-explosion hazard.						

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[1] Table is based on tests made in dust-explosion laboratory of Bureau of Mines, U.S. Department of Agriculture, Pittsburgh, Pa. (see Bureau of Mines *Report of Investigations No.* 3751).

[2] The powders are arranged approximately in order of decreasing dust-explosion hazard. All tests were made on minus 200-mesh dusts.

[3] 1 joule = 1 watt-sec = 0.00095 Btu. Dust clouds are ignited by static sparks from condenser discharge.

[4] Data on pressures and rate of pressure rise are for a dust concentration of 0.500 oz/cu ft.

[5] Samples included phenol formaldehyde, phenol-anhydro-formaldehyde aniline, and phenol furfural.

[6] Samples included cellulose acetate, cellulose acetate butyrate, and ethyl cellulose.

[7] Samples included polyvinyl acetate and vinyl chloride acetate copolymers.

[8] Ingredients of synthetic resins.

[9] Fillers for molding compounds.

Coal Pulverizing.[15] The chemical composition of coal varies greatly as does its physical structure, and both features contribute to the possibility of spontaneous ignition, the presence of sulfur, in the form of pyrites or marcasite, free gases, oxygen, and moisture being factors that contribute to this possibility also

Coal should not be stored in temperatures in excess of 75 to 80°F and should be kept free from combustible material Piles should be tested for heating at regular intervals Coal being a product of carbonaceous vegetable matter when pulverized creates a severe dust-explosion hazard, and all precautions suggested for explosive dusts should apply to coal particularly

Table 5-45. Permissible Percentage of Oxygen in Atmospheres to Prevent Explosions of Powders Used in the Plastics Industry*

Type of powder	*Maximum permissible oxygen per cent*
Alpha pulp ground	7
Casein rennet	7
Cellulose acetate	7
Cellulose acetate molding comp	7
Cotton flock ground	7
Coumarone indene resins	11
Hexamethylene tetramine	11
Lignin	7
Methyl methacrylate molding comp	7
Pentaerythritol	7
Phenolic resins	9
Phenolic molding comp	7
Phthalic anhydride	11
Polystyrene resin	7
Polystyrene molding comp	9
Rubber synthetic	11
Shellac rosin gum	9
Urea resins	11
Urea molding comp	9
Vinyl butyral resin	5
Vinyl resins	11
Vinyl molding comp	15
Wool flour ground	7

*See Bureau of Mines *Report of Investigation No* 3751

PLASTICS

The relative combustibility of any particular plastic composition depends not only on the synthetic resin used but also on the plasticizers, fillers and pigments incorporated in it In a preliminary test, the various types of plastics subjected to test flame (yellow gas flame) were found to fall into three groups (see Table 5-43)

Group 1 Those plastics which are ignited by the test flame in approximately the same time and burn at about the same rate as cellulose acetate

Group 2 Those plastics burning with a feeble flame, which is extinguished by a slight draft and which may or may not propagate away from the point of ignition

Group 3—Those plastics which burn only during the application of the test flame

Table 5-46. Causes of Fire

	Per cent
Smoking and matches	29 41
Misuse of electricity	10 11
Exposure fire originating off premises	7 56
Sparks on roofs	7 08
Overheated or defective chimneys and flues	7 06
Lightning	6 49
Stoves furnaces boilers and their pipes	6 16
Open lights	5 90
Petroleum and its products	5 53
Hot ashes and coals open fires	4 48
Ignition of hot greases	1 16
Spontaneous combustion	7 08
Gas natural and artificial	1 32
Sparks arising from combustion	1 01
Rubbish and litter	0 76
Miscellaneous	1 65
Friction sparks from running machinery	2 34
Explosions	1 70
Fireworks balloons etc	0 28

CAUSES OF FIRE

According to the latest statistics the classified causes of fires are as shown in Table 5-46

The per cents in Table 5-46 are for reported fires where the cause is ascertainable and do not represent either the total number of fires or the total property loss in the United States Excluded are all unreported losses as well as fires resulting from unknown causes It is estimated that the aggregate property loss in the United States during a decade was in excess of 3 billion dollars The causes of fire in productive industry of which there is definite knowledge are those taken from the enumerated fire losses where the cause is given, and the collection of these known causes is of primary importance

Electrical Fires. Some of the causes of electrical fires are arcing, sparking, overheating, switchboards, transformers, generators, motors, wiring, switches, lightning arresters, switchboxes, and panel boxes.

The National Electrical Code is the governing standard for all electrical installations in the United States. This code differs from most standards as it is unquestioned in its authority on the subject because of its wide use and approval by the electrical industries and engineering bodies.

Electricity passing over a circuit generates heat. This heat is generated in direct proportion to the resistance of the conductor and to the square of the current. Resistance of a conductor varies with its length. Double the length, and the resistance is doubled; treble the length, and the resistance is trebled. Resistance varies inversely with the cross-sectional area of a conductor. Doubling the cross-sectional area reduces the resistance to one-half. As the temperature rises, the resistance rises, except in carbon where the resistance falls as the temperature rises. The sizes of wires or conductors are expressed in the Brown and Sharpe Gauge "B-S," the smaller numbers indicating the larger wires.

Arcing. Arcing is caused by the interruption of the current, its intensity depending upon the amount of current and voltage. The arc temperatures can be very high, and if the arc comes in contact with combustible material, fire will ensue. To eliminate the possibility of arcing, all connecting joints and tie-ins must be made secure, and the possibility of exposed conductors (wires) should be avoided (see Motors). Ends of conduit should be properly nippled to avoid rough edges. There should also be proper insulation where conductors pass through partitions or are held on supports, particularly metal supports.

Sparking. Sparking may occur if conductors come in contact with metal, such as the conduit itself, the metal used in the insulation of the conductor, metallic supports, nails, or junction boxes.

Overheating. Overheating is usually caused by overloading the circuit. The National Electrical Code limits the current a conductor shall carry and therefore the amount of heat generated. The resistance of conductors depends upon four factors:

1. Kind of material
2. Length
3. Cross-sectional area
4. Temperature

Taking the resistance of silver, the best electrical conductor, at 1 ohm, the resistances of several other materials are expressed in Table 5-47

Table 5-47

Material	Ohms	Insulators	Billion billion ohms
Copper	1.1	Bakelite	0.031
Aluminum	1.8	Glass	20
Iron	6.5	Mica	30
Platinum	7.0	Amber	3,300
German silver (white copper)	18	Sulfur	10,000
Mercury	60	Hard rubber	100,000
Nichrome	70		
Carbon	2,300		

Switchboards. Switchboards should be examined for defects such as loose connections, lack of cleanliness, and freedom from combustible material. This also applies to panel boards. The use of marble or slate is not recommended because of the presence of iron in these materials. Ebonized asbestos or mica is preferable. Protection for over and under 600 v should follow National Electrical Code. Shunts or tie-ins on current, for purposes other than switchboard design, should be avoided. Features such as overloaded relays, lightning arresters, grounding, bus bars, and wiring should be constantly examined for defects.

Fire Streams and High-tension Electric Lines.[17] In this day of high-voltage lines and equipment, it is pertinent for every plant manager to have some knowledge of the danger involved when a fire occurs in electrical equipment or near electrical lines.

Numerous tests have been conducted on this question, and from these tests certain facts have been deduced. These are as follows:

A stream of water will carry current back to the nozzle, and this current may be sufficient to injure or kill a person.

The amount of current will depend upon certain conditions:

1. The voltage of the wire or device
2. The distance from the nozzle to the electrically charged line or device
3. The size of the stream
4. Whether the stream is solid or broken
5. The purity of the water

For voltage up to 600, which is classed by the National Electrical Code as the limit of low potential, there is no danger unless the nozzle is brought within close distance to the line or device. This is particularly true where the fire fighter is wearing rubber boots. For ordinary lighting wires of 120 v to ground, the nozzle can be within a few inches of the charged line. For such voltages as are ordinarily used for trolley service, *i.e.*, 550 v, the nozzle can be handled without discomfort at a point 3 to 4 ft from the wire. However, it is well to consider every wire as a hot wire because of the possibility, at fires, of crosses with high-voltage wiring.

For voltages higher than 600, the distance that a nozzle should be kept from the electrical line or device would be as shown in Table 5-48.

Table 5-48 applies to fresh water; sea or salty water or the discharge from a soda-acid extinguisher may have such high conductivity that no rule can be applied as to the safe distance for a solid stream. If it becomes necessary to use water known to have high conductivity, the stream should be broken up.

Table 5-48

Voltage	Safe distance, $1\frac{1}{4}$-in. nozzle, ft	Safe distance, $1\frac{1}{2}$-in. nozzle, ft
1,100	6	9
2,200	11	16
3 300	15	22
5,500	18	27
6,600	19	29
11 000	20	30
22,000	25	33
33,000	30	40

The spraying of a stream may be an important factor in electrical fires, especially those involving transformers, oil-cooled switches, burning insulation, and wood or other combustible material near electrical equipment. Tests indicate that the spray or fog nozzle gives protection to the holder and also that such a nozzle is very efficient in cooling the equipment and extinguishing the fire.

Some recent installations of substations, transformers, and switch points on high-tension networks have been equipped with spray or fog nozzle systems with thermostatic systems to give automatic operation.

Table 5-49

Voltages	Distance, in	Voltages	Distance, in.
7,500	6	88 000	52
15,000	12	110,000	64
25,000	17	132 000	77
37 000	24	154 000	89
50,000	32	187 000	106
73,000	44	220,000	124

The data available indicate that for fog or spray nozzles the general clearances between any portion of the equipment and live electrical apparatus should not be less than as given in Table 5-49, and whenever possible these clearances should be increased.

Of the various hand extinguishers listed by Underwriters' Laboratories, Inc., the ones suitable for electrical (class C) fires are those which discharge carbon dioxide and those with a carbon-tetrachloride base. Dry-powder extinguishers are also suitable.

Foam, soda acid, and hand-pump tanks with antifreeze solutions are not suitable for electrical equipment.

Transformer. Transformer fires are due to possible arcs in the casing, which also can cause explosions. Gauges should be checked to see that the oil is maintained above the arcing point of the contacts. The oil should be tested to ascertain whether a lighter oil has been substituted for a high-test oil. Oil used in transformers is for the purpose of conducting heat to the casing and thence to the atmosphere.

Generators. The operation of generators is attended by many hazards, the worst of which is the breaking down of the windings which causes a disalignment of the stator and the disruption of the apparatus. Oil-feed systems for lubrication should be checked for leaks. The transit oil used in generators has a temperature limit of 360 to 470°F. Cooling systems are of the air-blast type—with hydrogen and CO_2 gas. The presence of hydrogen increases the fire hazard.

Motors.[18] There are several hazards attending motors. Motors are liable to produce arcs and sparks from lack of maintenance and from burnouts due to overloads and to low voltage at terminals. There is often improper overcurrent protection. Motors of over 5 hp should be equipped with an overcurrent device not greater than 25 per cent of the rating of the motor. Single phasing of multiphase motors is another hazard. Overcurrent protection should be installed to prevent single phasing which is generally caused by blowing of a fuse where a multiphase motor is protected by fuses. Motors should be protected from dampness, corrosion, accumulations of dust and lint, combustible material, and oily waste. The location of motors should be studied and frequently checked as they may be found in out-of-the-way places, such as small enclosures and closets, or they may be supported from ceilings or on top of elevator shafts.

Motors in hazardous locations should be of the explosionproof type but preferably located outside of rooms or enclosures where flammable gases are present. Shaft drives through walls can be installed to accomplish this. Explosionproof motors in spray-painting atmospheres should be cleaned frequently so that cooling effect is not retarded. All motors should be adequately grounded. Where starting devices are used on d-c current resistances, the starting device should be free of combustible material because of its overheating possibilities.

Wherever specific maintenance instructions for a particular motor are available, they should be followed closely. In the absence of such specific instructions, the following periodic checks should be made. These are based on average conditions with regard to severity of duty and dirtiness of location. The frequency of the inspections can be varied to suit actual plant conditions.

Weekly inspections and operations:

1. Examine commutator and brushes.
2. Check oil level in bearings but do not overlubricate.
3. See that oil rings turn with shaft, and note any excessive end play in shaft.
4. See that shaft is free of oil and grease from bearings.
5. Examine starter, switch, fuses, and other controls, as well as ground connections and motor loads.
6. Open-type motors in dusty locations should be blown out thoroughly.
7. Check air gap between rotor and stator. The difference in width of gap around circumference will indicate extent of wear on bearings. This check should be made weekly on motors operating with excessive belt tension; otherwise check air gap semiannually.
8. Start motor and see that it is brought up to speed in normal time.
9. Check motor and bearing temperature under actual load conditions by feeling for heat.

Semiannual inspections and operations:

1. Clean motor thoroughly, blowing out dirt from windings, and wipe commutator and brushes.
2. Inspect commutator clamping ring.
3. Check brushes and renew any that are more than half worn.
4. Examine brush holders and clean them if dirty. Make sure that brushes ride free in holders.
5. Check brush pressure.
6. Check brush position.
7. Drain, wash out, and renew oil in sleeve bearings.
8. Check grease in ball or roller bearings.

9. Check operating speed or speeds.
10. See that end play of shaft is normal.
11. Inspect and tighten connections on motor and control.
12. Check current input and compare with normal.
13. Run motor and examine drive critically for smooth running, absence of vibration, worn gears, chains, or belts.
14. Check motor foot bolts, end-shield bolts, pulley, coupling, gear, and journal setscrews and key.
15. See that all motor covers, belt, and gear guards are in good order, in place, and securely fastened.

Annual inspections and operations:

1. Clean out and renew grease in ball- or roller-bearing housings.
2. Test insulation by Megger.
3. Clean out magnetic dirt which may be hanging on poles.
4. Check clearance between shaft and journal boxes of sleeve-bearing motors to prevent operation with worn bearings.
5. Clean out undercut slots in commutators.
6. Examine connections of commutators and armature coils.
7. Inspect armature bonds.

Motors should generally be given complete overhauling at intervals of 5 years or so, normally.

Arc-welding Sets. The general maintenance schedule relating to motors and motor controls will apply generally to arc-welding sets. Proper ventilation should be provided, and machines operated outdoors should not be covered with tarpaulins in such a way as to impede ventilation.

Wiring. Wiring in general should be protected against corrosion due to acids, corrosive vapors, or electrolysis, heat and moisture, and loosening of supports. The junction-box covers should not be removed, and wiring should not come in contact with metal. The first requisite is good workmanship and compliance with National Electrical Code.

Switches. The defects of knife and snap switches are arcing due to loose connections, pitting and burning of contact points, overheating, and poor mechanical conditions, such as corrosion. The mercury switch is adaptable for hazardous locations.

Lightning Arresters.[19] Lightning arresters should be inspected at connections in early spring just before the lightning season and every month during summer. If ground resistance exceeds 5 ohms make necessary improvements to lower it to 5 ohms or less. Never run ground wire through iron or steel pipe for protection against mechanical injury unless wire and pipe are bonded together.

Copper used in lightning arresters should have a weight of at least 3 oz per ft. Aluminum should have a weight of at least 2½ oz per ft. Steel should have a weight of at least 4½ oz per ft. Grounding should be to a plate placed at least 8 ft in the ground, if possible, and kept wet.

If lightning disables service, all switches should be turned off as resumption of current may start machines or apparatus at an inopportune moment.

Switch Boxes, Panel Boxes, Etc. All metal boxes of the wiring system should undergo monthly inspection; in locations that are particularly dusty, boxes should be cleaned out weekly. All covers should fit tightly and be in place. All unused knockout holes should be plugged tightly. Bond wires and ground connections should be checked.

Fuses and Switches. All fuses should be inspected twice each year to ascertain that circuits are not overfused. Fuse clips should be clean and tight, and all connections should be checked to prevent overheating. Fuses and switches should be inspected closely for any evidence of overheating as a result of high-resistance contacts or overloads. Refillable fuses should be examined, and charred cases replaced, after making certain that cases are refilled with links of proper rating and that fuses are assembled tightly. Switches should be inspected at same time fuses are checked. Operation of quick-break features should be checked.

Control Equipment. Control equipment should be kept clean. In dusty locations all starting switches, compensators, controllers, and air-circuit breakers should be cleaned and blown out at least weekly. Dust or dirt should not be allowed to accumulate on high-voltage bushings or terminals. An accumulation of dirt, oil, grease, or water should not be permitted on the operating parts of control equipment. In damp or corrosive atmospheres, control parts should be painted quarterly or semiannually. In clean locations semiannual cleanings will suffice. The mechanical condition of all control equipment should be checked weekly if subject to vibration, or monthly otherwise. All nuts and screws should be tightened, and electrical and mechanical connections on both panels and resistors should be examined. Condition of pigtail connections should be checked closely, especially in corrosive atmosphere. Any damaged or worn parts should be replaced, and condition of magnet contact surfaces should be inspected. Control mechanism should operate freely, and oil or grease should be applied if necessary. Contact surfaces should not be lubricated. Equipment subject to severe operating conditions should be checked; contacts should be inspected weekly. A monthly check is sufficient for average operating conditions. Contactor tips should be adjusted, filed clean, or renewed if badly worn or burned. Contactor circuit should be examined closely for any high-resistance joints. A new combustion a-c/d-c millivoltmeter is available for checking resistance of bolted joints and contact surfaces. A semiannual inspection of oil level and condition in starting compensators, oil switches, and circuit breakers should be made. Leakage and evaporation loss should be replaced; oil should be replaced if dirty or gummy. In dirty or damp location, oil level and condition should be inspected quarterly or monthly according to severity of condition.

The following overload devices should be tested:

Circuit breaker tripping points should be checked monthly. Dashpot oil should be replaced if thick and gummy. Orifices in plunger should be kept clear. Leather bellows should be kept soft and pliable with neats'-foot oil. Heater coils for thermal overload relays should be checked to see that they are the correct size. Overload relay settings should be checked every 6 months.

Spontaneous Ignition. Heating and ignition, involving a combustible material, due to chemical action rather than a heat source, such as a flame, spark, radiation, a hot surface, or friction, are known as "spontaneous." The process is known as spontaneous heating if the ignition point is not reached and as spontaneous combustion if the ignition occurs. Spontaneous heating starts with a slow oxidation or fermentation which generates heat. As the heat increases, the chemical action is accelerated until the ignition temperature is reached. The processes and conditions that may lead to spontaneous heating or ignition, taken in their widest sense, are

1. Moisture
2. Bacterial activity —Agricultural products, fodder, manures
3. Germination
4. Storage in large heaps—Agricultural products, coal, tobacco, oleaginous substances
5. Protracted drying—Wood, organic substances
6. Contained sulfur—Lampblack, coal
7. Contained finely divided carbon—Metallic sulfides
8. Containing fat or oil—Organic substances, fibers, colors (paints), clothing
9. Occlusion of oxygen—Coal and metals
10. Absorption of moisture—Quicklime, potassium, sodium, carbides (acetylene)
11. Fineness of division—Metals, bronzes, dusts, fats, and oils
12. Recent calcination—Carbonized substances, metallic powders, metallic sulfides, lampblack
13. Exposure to the sun—Phosphorus in fragments, oxyhydrogen gas
14. Concentration of the sun rays (burning glass)—All readily ignitable substances
15. Friction, pressure, shock fall—Numerous detonating, explosive substances; spontaneous ignition here generally results in explosion

16. Electricity (sparks)--Explosive vapor mixtures, dry-cleaning works, resinous bodies

Examples 1 to 9 mostly exhibit slow development (chronic spontaneous ignition). Numbers 10 to 14 progress with moderate rapidity, while 15 and 16 are sudden and of explosive character (acute).[30]

Substances liable to spontaneous ignition have been classified by A. H. Nuckolls of Underwriters' Laboratories as follows:

Group 1—Calcium oxide (unslaked lime), barium oxide, sodium peroxide (with water)

Group 2—Hydrides of phosphorous and silicon (in air), sodium and potassium (in water), turpentine and ammonia (in water), hydrogen and chlorine (in light), phosphine with vapor of hydride of phosphorous (in air)

Group 3—Vegetable oils and fats (rancid oils and fats more susceptible), metals in finely divided states (oil soaked), metallic sulfides, charcoal (hardwood by retort methods is worst; in new charcoal mostly; sacked charcoal should be piled 6 in. high only), coal (except hard) sprayed with a high-flash-point mineral oil prevents spontaneous ignition

Group 4—Hay and grain, owing to enzymes, ferment during germination period and when wet

Other Substances Subject to Spontaneous Ignition. Some other substances subject to spontaneous ignition are the following:

Fish scrap (green)
Hemp, jute, and sisal
Sugar and gunny sacks
Wood sawdust (in icehouses)
Lampblack
Leather scrap
Wool waste
Oily waste
Printed matter in piles, owing to linseed oil and ink
Nitrate of soda

Smoking. *Cigarettes and Matches.* There are manufactured in the United States every year 700 million miles of cigarettes. These are burned down to about one inch while being smoked and then discarded. This amounts to 513,520 million potential fires from this cause every year. It has become the leading cause of fire and involves the home, industrial plants, and the forests. The match is another contributing factor, as the type now used, the book match, has a wide distribution. The sulfur-head match, subject to ignition from dampness, is responsible for the mice and match theory and for fires where they are stored in poorly protected premises. It is suggested that signs be posted in industrial plants to discourage the discarding of lighted cigarettes and matches. These signs might read:

"Cigarettes are dangerous if not put out before throwing them away. Please be careful."

"Don't be a 'Ciggyflipper' " (an accepted word).

"Only smoke in safe places."

Oily Waste. The fire hazard of cotton waste and wiping rags impregnated with oil is due to the phenomenon of oxidation. Cotton waste that is "slightly" coated with oil presents a large surface within the mass to the process of oxidation, and the heat of this process will reach the ignition temperature of the oil and the cotton. Cotton waste or rag saturated with oil does not present this danger. Oily waste when not in use should be kept in a standard waste can, which is a metal can without any soldered joints. It has a self-closing cover and sets on legs 3 to 4 in. from the floor. (This is a standard piece of equipment for all machine shops.)

Packing Material. Any kind of packing material that is combustible, such as excelsior, should be kept in standard bins. These bins, designed to hold at least 1 bale of excelsior, are constructed of wood covered with lock-jointed tin (similar to fire-door construction) with automatic covers. Large supplies of packing material should be kept in fire-resistive sections of an inside area or kept outside the building.

Open Lights, Flames, and Sparks. Any open light or spark-producing apparatus is a possible source of fire, particularly in the presence of flammable vapors and on or in the vicinity of the leeward side of any air or wind motion which may carry flammable vapors such as gasoline, brazing torches, kerosene torches, welding torches, gauge glass lights, gas jets, pipettes of glass blowers, and inert-gas producers burning gas or oil. Salamanders used in buildings during construction or razing of buildings should be set on incombustible material and never left unattended. Chimneys, stacks, and refuse burners should be equipped with spark arresters.

Table 5-50. Iodine Absorption

(Indicating tendency to spontaneous ignition)

Acetone	90	
Alcohol, amyl	76	
Denatured	80	
Ethyl (grain)	78	
Methyl (wood)	85	
Amyl acetate	77	
Aniline oil	65	(Vlackos)
Benzine	95	
Benzol (benzene)	96	
Bone oil	25	(Vlackos)
Carbon bisulfide	100	
Castor oil	10	(Vlackos)
Coal tar	93	
Cocoanut oil (refined)	32	(Vlackos)
Corn oil	69	(Vlackos)
Cottonseed oil (refined)	62	(Vlackos)
Cylinder oil (mineral)	1	
Ether, sulfuric	97	
Fuel oil	83	
Fusel oil	75	
Gasoline	95	
Glycerin	5	
Kerosene	70	
Lard oil (commercial or animal oils)	52	(Vlackos)
Lard oil (pure)	19	(Vlackos)
Linseed oil	75	(Vlackos)
Lubricating oil (mainly mineral)	12	(Vlackos)
Menhaden oil	85	(Vlackos)
Mustard oil (wild)	67	(Vlackos)
Naphthalene	85	
Neat's-foot oil	15	(Vlackos)
Nitrobenzol	60	(Vlackos)
Oleo oil	19	(Vlackos)
Olive oil (pure)	13	(Vlackos)
Palm oil	40	(Vlackos)
Palm-kernel oil	33	(Vlackos)
Peanut oil	18	(Vlackos)
Perilla oil	72	(Vlackos)
Petroleum (crude)	93	
Pyridine	89	
Rape oil (pure)	58	(Vlackos)
Red oil	50	(Vlackos)
Rosin oil	76	(Vlackos)
Solvent naphtha	90	
Soybean oil	70	(Vlackos)
Sperm oil	30	(Vlackos)
Tallow	22	(Vlackos)
Tallow oil	26	(Vlackos)
Toluidine	64	
Toluol	92	
Tung oil	71	(Vlackos)
Turpentine	85	(Vlackos)
Whale oil	77	(Vlackos)
Xylidines	63	
Xylol	88	

NOTE: Fifty grams of cotton impregnated with linseed oil rises in temperature to 23.5°C during the first hour. In 15 hr, however, it rises to 170°C.

Bearings. Bearings are a source of fire, particularly neglected bearings (from a mechanical sense) or ones not oiled regularly. On driven shafts, the offside bearing is the one most likely to heat. All shafting should be properly aligned to prevent undue friction at the bearing. Bearings that may be concealed on the inside of apparatus should be made accessible by providing hand holes for inspection and lubrication. Bearings should be kept clean and free of dust and lint.

Belts. Any moving belt is a potential generator of static electricity and should be tested for it. When it is present, the proper grounding apparatus should be installed. This consists of combs and points properly connected to the ground. Pulleys of wood should be kept clean and free of oil. Rope drives should have sufficient clearance from combustible material to avoid friction (see Static Electricity).

Belt Dressings. The problem with belt dressings consists in developing a material which can be easily applied to belts, which will have sufficient electrical conductivity to remove or neutralize static electrical charges as rapidly as they are formed, which will resist wear, which will not decrease the coefficient of friction between the belt and the pulley, and which will not be injurious to the belt. A good rubber belt dressing can be prepared by mixing 18 parts of lampblack with 82 parts of good spar varnish. The dressing must be dry on the belt before running. A nonflammable thinner for the varnish can be prepared by mixing equal volumes of carbon tetrachloride with VMP naphtha.

A good leather belt dressing consists of

100 cu cm liquid fish glue
100 cu cm sulfonated castor oil
80 cu cm glycerine

170 cu cm water
82 g lampblack
20 cu cm 2 per cent ammonium hydroxide

This can be applied while the belt is moving. If the belt has not been properly maintained, this dressing will cause the belt to stretch.

Cheap and low-grade stick or semiliquid belt dressings, which generally contain the ingredients shown in Table 5-51, should never be used on leather belting of any kind.[23]

Table 5-51

Stick type	Per cent	Semiliquid type	Per cent
Graphite	22	Tar	36
Rosin	27	Rape seed or cotton oil	54
Vegetable oil	44	Rosin	10
Paraffin	7		

Rubber belts[22] are more prone to the accumulation of static electricity than leather belts. The rubber belt has been given a certain amount of immunization by the incorporation of acetylene black in the friction compound. This makes the belt a good conductor, thus permitting the diffusion of the static charge. This type of belt is recommended for hazardous locations and can be installed without grounding.

Air-conditioning ducts and filters constructed of combustible material are subject to ignition from sparks, either from the internal apparatus or those sparks sucked into air ducts from bonfires and carelessly discarded cigarettes. Screens should be provided on all intake ducts.

Focused rays from bubbles in glass are frequent causes of fire, and where flammable or combustible material is stored, this feature should be checked.

Heating furnaces or cupolas of stationary types, whether coal-, oil-, or gas-fired, should have clearances as prescribed in the standard classification of furnaces (see Ovens and Dryers).

Overheated Materials. Overheated materials such as boiling or cooking resins, greases, asphalts, or pitch are liable to froth and flow down to the combustion unit or chamber and ignite. Proper clearances should be maintained around all kettles. They should be set on incombustible material and properly diked to prevent the contents of the tank from flowing out if frothing occurs. Molten metals can be a source of sparks and radiated heat, they should be safeguarded and kept from contact with combustible material. Metal, lead pots, zinc coating (subject to evolution of hydrogen gas), forges, oil-tempering furnaces, and bonderizing furnaces (metal spraying) are types of molten-metal hazards.

Sparks on Roof. The antidote for the danger of sparks on roofs is noncombustible roofing, tin, tar, or gravel, and asbestos or copper shingles. Some wooden shingles are treated with chemicals to make them slow-burning, but in the course of time, the heat of the sun nullifies this feature so that all wood-shingle roof coverings become a serious menace. The exposure of one wood-shingle roof to another is a grave danger in sections where they are much used. This calls for drastic action for their elimination.

Defective Chimneys and Flues. (Direct ignition with flaming or glowing material, prolonged heat, sparks.) Chimneys and flues as a source of fire are associated with the heating of buildings or power plants and with carrying off heated gases and vapors. They are in a direct sense a passageway connecting the source of heat with the outer air. Their construction varies with the conditions, but they are most often constructed of brick, stone, or cement block, with a suitable burned-tile lining. Some are made of metal and in rare cases of wood, the latter construction being necessary for conveying corrosive vapors. Under this heading will also be found sooty and dirty chimneys originally designed for one purpose and used for another.* A defective chimney or flue, therefore, may involve any one of three conditions—poor construction, soot or dirt, or overburden of original design.

Defects of construction consist of poor design, poor cement between bricks, inadequate thickness, no lining, or insufficient draft, thus permitting the escape of fire or sparks or heat which will ignite flammable material adjacent to the chimney.

Sooty or dirty chimneys may result in the ignition of the soot or dirt and cause an intense fire in the chimney, thus inducing disruption by expansion, the escape of fire or sparks, or the dropping of burning soot into the connecting flue or breeching causing it to become overheated and set fire to flammable material.

Overburden of original design creates a condition similar to poor construction with similar results.

The construction of a chimney is of the utmost importance, for it is a flue passing up through the building from cellar to roof and through attics and roof spaces, some of which are open and others closed. In many instances such spaces are boxed in with lath and plaster or wainscoting. These concealed spaces make it difficult to observe defects such as mortar falling out between the bricks, thus permitting sparks and flame to ignite the woodwork. This is also possible where the flue from a stove enters the chimney passing through this boxing. A chimney should be built in a straight line from the ground up, extending well above the roof. The bricks should be laid with care in a good smooth cement mortar. Unless it has 8 in. of brickwork around each throat or flue, the chimney should be lined with a hard-burned-tile pipe laid in smooth cement mortar with all joints tight. A chimney should never be used as a support for beams or joists nor should these ever be permitted to enter the chimney.

Ovens and Driers. Under ovens and driers come all classes of heat-treating mediums involving enclosures, piping flues, and stacks and involving the source of heat, the material to be treated, and the method of removing gases and flammable liquids.

The various types of ovens are graded on a temperature basis:

Low grade—Those developing a temperature up to the melting point of lead 600°F

Medium grade—Temperature of 601 to 1500°F

High grade—Temperature of 1501°F and over

Almost all devices will fall within these grades, but owing to special function and performance, many devices will require special treatment. Box-type ovens include all ovens that are closed when in operation. Continuous-conveyor ovens include all ovens that are open at both loading and unloading ends. Ovens should be located abovegrade and so installed that an explosion can be relieved with the least damage to building and equipment. Ovens should be accessible on all sides. There should be sufficient space between the top of the oven and the ceiling to permit the installation of automatic sprinklers or proper use of hose streams. Ovens should not be in contact with posts, beams, trusses, or other structural supports for the building. Regardless of the type of oven or furnace, certain clearances must be maintained. These clearances are shown in Table 5-52.

Table 5-52

	Without setting of brick or equivalent			With setting of brick or equivalent		
Grade	Above	Sides and rear	Front	Above	Sides and rear	Front
Low	48 in.	36 in.	48 in.	36 in.	24 in.	48 in.
Medium	48 in.	36 in.	96 in.	36 in.	24 in.	96 in.
High	15 ft	10 ft	30 ft	15 ft	10 ft	30 ft

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* The introduction of oil burners has put a heavy burden upon the ordinary dwelling chimney many were never designed to resist the intense heat or constancy of flame.

The front end of the oven or furnace is the loading end; because of the opening and closing of doors and the possible emission of heat and flame, the clearance is naturally greatest at this end. Ovens and furnaces in certain industries will be constructed on the earth or solid brick or concrete foundations.

Where ovens are supported on wooden or combustible floors, they should be separated therefrom by a space of at least 18 in. or insulated by one, two, or three courses of 4-in. hollow tile, the tile cells laced end to end to form ventilating ducts to permit circulation of air under the oven. In large furnaces in which the material is melted by straight fire heat or an oxidizing flame, incombustible areas should be maintained around the furnaces in case the furnace walls should break permitting the contents to flow out.

Core Ovens. Core ovens are used for drying cores or sand molds in foundries. They should be constructed of incombustible material, and the interior racks should be of metal. Flasks and fallow boards should be kept clear of core ovens.

Caul Boxes. Caul boxes are used for drying patterns and wood molds. They should be made of incombustible material with interior racks of metal. They should be cleaned frequently.

Dry Rooms. Dry rooms are of various types but involve similar hazards.

Lumber Driers. Lumber driers should be made of incombustible material and cleaned frequently as interior surfaces become coated with resinous distillates. They should be equipped with steam jets or fog, manually or automatically controlled.

Annealing Furnaces. Annealing furnaces run high temperatures and should be made of incombustible material and have clearances similar to ovens. If electrically heated, control apparatus should be of the temperature type to prevent overheating and should be equipped with pilot lights to show when current is on.

Special Types. Ovens using japan or enamels with dipping are a special type. They should be of brick, concrete, or steel and approved incombustible material. All walls should be free of hollow spaces to avoid accumulation of vapors. If heated by steam, this rule is not mandatory. Ovens of this type should be provided with explosion reliefs, hatches, or permanent openings having an effective area of not less than 1 sq ft for each 15 cu ft of capacity. Vent pipes should be treated as "heat flues" and kept well away from combustible construction or materials and be insulated where necessary. They should not pass through floors, if possible to avoid, but be as short as possible in passing to the outside of the building. To avoid an explosive atmosphere in box-type ovens and maintain a reasonable factor of safety (2), 4,600 cu ft of air should be provided for each gallon of solvent evaporated based on an oven temperature of 70°F. The lowest explosive limits of the vapors, from the commonly used naphtha, benzine, gasoline, and turpentine substitutes, are about 1.0 to 1.4 per cent vapor in air (by volume) at 70°F -or as an average, 1 gal of solvent evaporated in 2,300 ft of air.

Heating Units (see standards). A reliable pilot light or electric ignitor should be provided for gas or oil burners and should be so arranged that the main fuel valve cannot be opened unless pilot is in operation.

Grounding. All metal parts of ovens and conveyors that may be exposed to flammable vapors should be thoroughly grounded. Grounding conductors should be not less than No. 8 gauge copper and should be protected from mechanical injury.

Fire Extinguishment for Ovens. Water, carbon dioxide, or nitrogen, foam carbon tetrachloride, or steam can be used to extinguish fires in ovens. Where temperatures are over 250°F, open sprinklers should be installed which are controlled by automatic valve or releases. At temperatures over 250°F condensations from finishes will cause loading or coating of sprinklers.

The use of flammable solvents to clean ovens should be prohibited.

Welding.[24] Welding and cutting operations present hazards due to sparks of molten metal igniting paper, rags, oily waste, excelsior, and other loose material or the sparks lodging in cracks in woodwork or falling through holes in floors. The flame of the torch, a high temperature, 2000 to 2400°F in the range of orange yellow or yellow white, may come in contact with combustible material, or a backfire from the torch may cause a fire. Where welding is done by portable systems, such as those on trucks or near or attached to workbenches, all combustible material should be protected against sparks by metal plates or asbestos blankets, and a fire extinguisher or pails of water and sand should be kept near by. Canvas tarpaulins should not be used as covers. The welding equipment should be securely fastened in an upright position on the truck or at any bench, and only experienced operators should be allowed to use it. All hose should be kept in perfect condition and free from leaks. If there is any doubt of the integrity of the hose, it should be discarded. Tape should never be used to fix up a hose. Hose lines can become burned by the flame of the torch in moving equipment about. Oxygen hose is colored green; acetylene is red for identification. Where oxygen is used under pressure, all grease and oil should be positively kept away and removed if found to be present, as grease or oil in contact with oxygen under pressure will cause a violent explosion.

In fixed systems where welding is done as a prime process, the entire operation should be installed in a fireproof explosion-vented room, cut off from the rest of the plant.

Cylinders of oxygen should not be stored in the same room with cylinders of acetylene, unless separated by fire-resistive partitions (see Standard Welding Room Design in Standards). Cylinders should be kept away from stoves, heaters, radiators, furnaces, or forges. All space around them kept clear of combustible material, oil, and grease. When cylinders are stored in the open, they should be protected from accumulations of ice and snow and from the direct rays of the sun. Where the generation of acetylene is carried on, this should have positive protection and should be installed in a separate fireproof explosion-vented room, even if this must be a part of the enclosed area of the entire process room. This section should be kept dry and waterproof. The storage of 600 lb of calcium carbide is permitted in generator rooms or in a separate compartment near by in one-story buildings, preferably without cellar or basement. No water should come in contact with carbide outside the generator. No open lights should be brought near carbide or open containers. Water should never be used to extinguish a carbide fire.

Friction. Wherever there is metal-to-metal contact in mechanical apparatus, particularly in bearings, the use of the right type of lubricant is essential. A heating bearing is the sign of neglect, or it may be that the oil or lubricant is not reaching the points of contact. Drip and wick oilers are liable to clog; the ring or splash types are less liable to give trouble unless the oil gets low. Inspections should be made frequently to ascertain if drip cups are full, if the lubricant is reaching the bearing, and if sufficient oil or lubricant is present. Friction caused by the clogging or stoppage of conveyor belts in elevators at the pulleys is a cause of fires. To avoid the continuous operation of the power mechanism in such installations, it is recommended that a "Bethlehem starter" be set between the power unit and the belt mechanism.[25] This device is inherently a limit torque coupling. It will not transmit to the driven machine or impose on the motor more than the maximum load for which it is designed. It is therefore an absolutely dependable overload protection for the motor and the driven machine.

Rubbish and Litter (sparks, direct ignition with flaming or glowing material, prolonged heat, spontaneous ignition, focused rays). The undue accumulation of discarded material, whether it be the usual household rubbish or industrial-plant refuse, is always a breeder of conditions that may result in fire. The rubbish may contain oily waste or greasy rags and paper which can set up a reaction causing spontaneous ignition. Matches are frequently caught up with the rubbish and by dampness or friction may ignite the surrounding flammable material. A carelessly thrown cigarette, cigar, or match may lodge among the litter and cause fire.

If there is no specific place, such as a fire-resistive receptacle, to put rubbish, it may be thrown anywhere and come in contact with steampipes, defective electric wiring, or be subject to the focused rays of glass bubbles. All these direct causes of fire are rightly attributed to rubbish and litter because if it were not permitted to accumulate the fires would not occur. Rubbish and litter should be removed from the premises and kept in metal receptacles until final removal, which should be often. Wastepaper is a menace because of its heterogeneous character and the possibility of spontaneous ignition, for it may consist of greasy lunch papers, oil-soaked paper, and even wiping rags. Also cigarettes may be thrown into the pile or be gathered up when it is collected. Wastepaper should be kept in fire-resistive rooms and disposed of daily.

Static Electricity. Static electricity is generated by friction or the bringing together and then separating of two unlike substances. It is not altogether possible to prevent, but static sparks can be avoided by preventing the accumulation of the static charges. The static charges are found to be present in processes involving the flow through pipes or restricted orifices of flammable liquids such as gasoline, etc., in dust that is in motion, in finishing and coating processes, and on belts and dryers in motion. They are also found in mills employing spinning and roving operations. The usual methods of preventing the accumulation of static charges is by grounding all moving parts and containers and by bonding connecting apparatus. Humidity may also be applied in the form of vapor, spray, or evaporation at times when humidity is low (winter) or in sections that may become low in humidity. A relative humidity of 50 to 60 per cent may be sufficient, but the minimum for safety may vary with conditions. It is believed that a normal percentage of carbon dioxide must be present to make humidification effective.[26]

Neutralizing. Air is ordinarily a nonconductor of electricity, but if sufficient ionized conductivity is produced, it will prevent the accumulation of static charges. Ionization can be produced by electrical discharges or by radioactive substances. Static neutralizers employ a high-voltage transformer of small capacity to produce brush discharges. This equipment is liable to ignite flammable vapors or dusts. Gas flames produce an ionizing effect but should not be used in the presence of flammable vapors or dusts. The human element enters the study of static electricity as it has been found that some people are more subject to the accumulation of static charges than others. People with dry skin appear to be more liable to this accumulation than those having moist skins. Similarly, people of blonde coloring are more liable than those of brunette coloring, who are generally not negative to the charge. The grounding of the person by portable ground wires, using a metal-grounded plate to stand upon with metal-pegged shoes, and grounding portable knives on coating machines are effective. A method of grounding the personnel in a hospital operating room has held some attention because of the hazard that has developed in the case of anaesthetics which are explosive. A specially constructed floor, both grounded and of material to prevent static charges, has also been given attention.

Hospitals. The use of flammable and explosive anaesthetics in operating rooms is liable to cause damage. Such anaesthetics are ether, ethylene, cyclopropane, propylene, ethylene chloride, and nitrous oxide. Where oxygen is used, because it is a supporter of combustion, its presence may add to the rapidity and intensity of a fire. Fire in hospitals may be caused by the use of X ray, electric cauteries, radio knives, high-frequency machines, and open gas flames in nearby laboratories and kitchens.

Table 5-53. Properties of Anaesthetic Gas—Explosive Limits*

Gas	Per cent in air		Per cent in oxygen		Per cent in nitrous oxide	
	Lower	Upper	Lower	Upper	Lower	Upper
Ether	1.85	36.5	1.7	39.5	3.8	25.7
Ethylene	2.75	28.6	3.1	79.9		
Ethyl chloride	3.7	12.0				
Cyclopropane	2.4	10.3	2.45	63.1	3.0	28.0
Propylene	2.2	9.7		52.8		

* See *Anaesthetics in Hospitals*, NEPA.

Electrostatic Voltmeter. The ordinary voltmeter designed for measuring voltages of an electrical circuit is not suitable for measuring the voltage of a static charge.

For this purpose a reliable electrostatic voltmeter must be used. An electrometer or electrostatic voltmeter has been successfully used when calibrated to 10,000 or 50,000 volts. Although the calibration varies to some extent when the instruments are moved, they have been found to be sufficiently accurate for most electrostatic voltage determinations. Another voltmeter consists of a needle system with the needle moving over a graduated scale so arranged that the operator may arrest the movement of the oscillating needle without waiting for it to come to rest by natural damping. The voltmeter scale is graduated in equal divisions which, together with the calibration curve which accompanies the instrument, give the static voltage. This type is available for a range up to 5,000 or 10,000 volts. The range of these instruments may be increased by connecting one or more condensers in series with the voltmeter, but unless the condensers are each of the same capacity, a large error may be made in such measurements. Before each reading the condensers should be completely discharged in order to ensure accuracy.

Neon-tube Testers. Probably the most satisfactory and inexpensive portable instrument for detecting the presence of static electricity in manufacturing plants is the neon-type tester commonly used for testing automobile spark plugs. It consists of a small glass tube containing rarefied neon gas into which the two electrodes extend. The electrodes are protected by suitable resistors built into the equipment itself. The neon tube glows when one lead is brought into contact with a body charged with static whether or not the other lead is grounded. It is sensitive enough to glow when held in the hand of a person walking over a high-pile rug in a dry atmosphere when one of the leads is grounded or brought into contact with a conducting material.

One method of testing for static at any given location is to touch one of the leads with the hand and to hold the other lead in close proximity to the suspected equipment or material. A glow in the neon-tube tester indicates the presence of static. The amount of the charge cannot be determined by the neon-tube tester alone. The use of this type of tester may be hazardous at some locations where explosive dusts or flammable vapors are present, since the static spark may jump from the object containing an accumulation of static to the lead of the tester or the operator.

Gold-leaf Electroscope. The gold-leaf electroscope is another instrument that may be used to detect the presence of static. It is primarily for use in the laboratory, consisting as it does of two delicate pieces of gold leaf suspended from a single electrode in a glass tube or one piece and a permanent bar. In the presence of a static charge, the two pieces of gold leaf will be repelled from each other sufficiently to be seen very easily by the observer. It is sensitive enough to detect a very small charge. Since the capacity of the gold-leaf electroscope is so small, there is practically no spark when it becomes charged in the presence of static or subsequently discharged. For this reason it may be used with less hazard for atmospheres where flammable vapors or dusts may be present, but when used in the industrial plant, it must be handled with care since it is easily injured. The gold-leaf electroscope does not indicate the quantity of static but merely its presence.

Ohmmeter. Ground wires are often "rung out" by using a small hand-operated magneto and bell set. While this indicates that an electrical path to ground exists, it does not give any indication of the resistance of the path. Since it is essential that grounds for static have a low resistance, the use of the magneto-bell set or "ringer" is not recommended for testing static grounds. For this purpose an ohmmeter, or meter reading the resistance in ohms, must be used. A common type consists essentially of a direct-reading ohmmeter of the permanent magnet moving-coil type mounted in a suitable case with a hand-driven generator or provided with other means for supplying direct current at suitable potential for the test. The instrument is so simple and direct in its operation that anyone can use it successfully by following directions. The ohmmeter can be obtained with direct reading scale having ranges as shown in Table 5-54.

Table 5-54

Scale, ohms	*Smallest scale division*
0.0–1.0	0.01
1.0–100.0	1.00
100.0–10,000.0	100.00
10,000.0–1,000,000.0	10,000.00

Switches on the instrument change the scale. The instrument has two binding posts, one of which is connected to the equipment and the other to known positive ground such as an underground water main. The magneto built into the instrument is rotated by the small hand crank, and the resistance of the instrument is read directly. Since the current flowing is infinitesimal, the resistance of the instrument leads is negligible. Static grounds, to be most effective, should have a low resistance. Ohmmeter tests should not be made in operating buildings while any exposed flammable solvent or vapor is present in the equipment or immediate vicinity of the equipment being tested, since the ohmmeter itself may produce a slight spark. Readings of resistance should be made of all ground connections and all parts of grounding systems when the systems are installed, and regular and periodic readings should be made thereafter to make sure that high resistances do not develop over a period of time because of corrosion, broken connections, or loose connections caused by vibration or in some other way.

Infrared Driers.[27] Drying by the radiant heat from infrared lamps has now become common in industry. Lamp driers are used in a variety of industrial processes, especially in the automobile industry and where finishes of paint, lacquers, and enamels are used. The principle, in effect, is the utilization of the radiant waves from lamps whose light waves are at the red end of the spectrum in the nearly invisible field of light rays. The near-infrared field includes wave lengths of from about 6,500 to 14,000 Å. Most of the heating lamps used permit a utilized energy below 20,000 Å. The violet rays at the other end of the spectrum are about 4,000 Å.

The term Angstrom unit (Å) indicates a standard scale for measuring the length of light waves devised by Angstrom, a Swedish physicist. The Angstrom unit is one-billioneth of a centimeter. Another standard is called the millimicron, which is ten times an Angstrom unit. The temperatures usually listed at the filaments (of the lamp) are about 1500°K. The symbol K is used to indicate the color temperature, which is the same as Fahrenheit, so far as it can be measured by color-temperature measuring devices. The sun is 6000°K. Mazda lighting lamps are from 2600° to 3200°K. Carbon lamps are 2000°K. Some unenclosed resistance heaters are made as low as 1200°K. The 250-w heating lamps on the market rate at approximately 11,500 Å (770 Btu per hr). The temperatures of surfaces painted or being treated usually run 250 to 350°F.

Drying by infrared lamps has some advantages over ordinary gas- or oil-heated ovens. Some of these are quicker results on certain types of goods, low cost, better working conditions, less floor area required, less odor and smoke, no oil- or gas-fuel hazards. Explosions are unlikely to occur as enclosures are not required. The drying results from the heat of the infrared lamps. The lamps used are usually 250 to 260 w rated at voltages of 105 to 110. Some 500- to 1,000-w lamps are used for special purposes. This method of drying was first developed by the Ford Motor Company. The lamps are usually arranged in drying tunnels, in the form of open cages, to fit the particular type of work to be treated. The lamp bases are usually ventilated so that the mountings may be kept cool. The drying effect is due to the heat produced by the interruption of the rays by the metal or body of the work being treated so that it may be said that the paint or coating is dried from the inside out. The lamps are said to have a life of from 10 to 20,000 hr.

Suggestions for Installations. All electrical equipment should be in accordance with the National Electrical Code. Installations should not be confined to small rooms unless they are thoroughly ventilated. Ventilated hoods or enclosures should be built at the first part of the conveyor with exhaust systems to discharge the first evaporation of solvents where large amounts of solvents are used. Lamp drying units should be well separated from spraying and dipping processes. Controls should be provided to shut off the lamps automatically if and when the conveyor stops and also if stationary goods are not removed from the drier on time. Goods on conveyors should be rigidly supported. Fire protection may consist of automatic sprinklers or approved water-spray systems over the lamp tunnels and under ventilating hoods, or an approved CO_2 gas system should be provided.

Electrostatic Coating Process. The electrostatic coating process, or electrical "detearing," supplies an automatic method of freeing materials of accumulations and fatty edges, which are spray- or dip-coated with alkydsynthetics, oleo resins, or polymerized types of finishes. In the process the article on a conveyor is dipped or flow coated in any suitable manner. It is then carried over the usual drainboard and is permitted to air-dry or set sufficiently so that the flow has practically ceased. At this point the grounded article is carried over an insulating grid to which a high voltage can be applied. An electric field having a difference of potential of 50,000 to 100,000 volts or over between the grounded goods, being painted, and the electrodes is maintained. Only a low amperage is used, about 10 ma.

One type of apparatus consists of a small oil-immersed power pack requiring less than 4 amp at 220 volts (single-phase) in the primary. This pack can produce 72,000 to 126,000 volts (alternating current, 8 steps). The device has a rectifying tube to produce direct current from a-c power. It is wired to electrodes hung in the spray booths or to other electrodes near which painted goods pass on the conveyor. Sparks can jump to the goods if they are within 6 or 7 in. of the electrodes. This introduces a serious spark hazard in the painting area and process. This force is particularly strong at points where drainoff is apt to occur and the excess paint is removed from the piece and a smooth coating remains.

Tentative Suggestions for Installations of Electrostatic Apparatus in Painting Processes[28]

Transformers, power packs, and all electrical controls are to be located outside the immediate painting area (not inside spray booths, well away from dip tanks, etc.).

The electric installation is to be in accordance with the National Electrical Code in so far as it applies.

Electrodes are to be rigidly supported in permanent locations and well insulated.

Minimum spaces between electrodes and painted goods are to be at least twice the distance of the length of the possible spark gap.

Installations are not allowed where goods being painted are handled manually. All goods subject to the processes are to be handled on conveyors and be so fixed to the conveyors that the minimum spaces from goods to electrodes will be maintained under all circumstances. Also guards should be provided where necessary to produce this result.

Automatic control should be provided to make the entire electrostatic equipment "dead" when vent fans and conveyors stop.

Standard ventilation should be maintained for the areas containing the electrostatic apparatus; the exhaust-fan motors for the ventilating system are to be connected in series with the voltage pack primary (rules for blower, exhaust, and ventilating systems to apply).

All painting processes supplemented by electrostatic apparatus should be protected by automatic sprinklers or other approved automatic extinguishing systems.

Where necessary, adequate fencing, railings, or guards are to be provided to maintain safe isolation of the process from other work in the vicinity and to guard personnel.

Signs designating the work zone as dangerous as regards fire and accident shall be posted at the process.

Electrode insulators shall be kept clean and dry.

Drip plates and screens shall be removable and taken to a safe place for cleaning.

Cleanliness shall be maintained and stock or goods not stored within 5 ft of the process.

Exposure. In fire-prevention parlance this literally means the spread of fire from one building to another. As a cause of fire, *i.e.*, of causing other fires, it looms large on the records. A fire in progress, in order to spread and cause other fires, must have opportunity for so doing. This is given it by the failure to safeguard surrounding property or by using flammable material for construction or roof coverings. Brick or concrete construction will resist fire, but if the openings in walls are not safeguarded by wired glass in metal frames or standard metal-clad shutters, fire will enter the building. When wood construction is used, the safeguarding of the openings in walls as noted above, is a material help to the fire-fighting forces as the time element enters, and before the fire can burn through a frame wall, water can be applied to cool the surfaces. Water curtains on wood buildings are also of value, applying the same principle. Incombustible roof coverings also are essential to prevent ignition from falling brands or sparks, as is the protection of roof openings by wired glass in metal frames or metal screens over heavy ribbed glass. The prevention of fires from exposure therefore calls for construction safeguards which will not permit ignition or entrance of fire to buildings.

Steam- and Hot-water Pipes (prolonged heat, spontaneous ignition, chemical reaction). Faint but prolonged heat is the classification under which fires from steam- and hot-water pipes are placed. This type of fire is also recorded as spontaneous combustion owing to the fact that the process of decomposition or charring is very slow, and ignition resulting from this, not being easily understood, is characterized as such. Steam- and hot-water pipes maintain a constant heat of low degree. As a source of fire they are to be considered more as a means of causing other material to ignite by contact, which breaks down or volatizes the flammable constituents, igniting at the temperature of the radiated heat or, upon cooling, absorbing oxygen setting up rapid heat or *reaction*. Woodwork or any combustible material that is liable to remain in contact with heated pipes may be ignited at an unexpected moment. It therefore constitutes one of the fire causes that is to be feared, as it is not always detected in its most insidious stage. The inherent hazard of steampipes in contact with wood is the slow action of applied heat which produces pyrophoric carbon. The temperatures of steampipes which can produce this result are shown in Table 5-55.

Table 5-55

Atmospheric pressure, lb	*Temperature, deg F*
14.7	212
25	267
50	298
100	338
150	366
200	387
300	422
500	470
1,000	546

Clearances around steampipes of from 2 to 18 in. must be maintained.

Pipes Conveying a Heating Medium. Pipes conveying a heating medium, such as steam, water, or oil, where they pass through floors or partitions, must have a sufficient clearance around them to prevent their inclusion in the list of causes of fire. The prolonged heat will eventually reduce wood to a charcoal or pyrophoric carbon which will ignite, or when the heat is withdrawn the wood will absorb oxygen and set up a reaction resulting in fire.

Pipes running in a horizontal direction through concealed spaces are liable to come in contact with dust accumulations or siftings or because of their warmth may attract mice to build nests close to them which can be a possible cause of fire.

Woods of commerce, being nature's product, are complex things made up of various materials which go to form a composite whole which is called wood, but varying in degrees of construction to such an extent that no one wood is like another, either in its construction or action under various treatment. Wood is that product which is made up of cellulose or lignin in combination with various resins and aromatic substances which have served the living tree. Therefore, in analyzing the burning point of wood, we may reach false conclusions if we assume that all woods are alike in this respect.

The long- and short-leaf or winter or summer, growths will differ greatly in action under heat owing to the resistance to volatization of the resins and aromatics. For general purposes it may be assumed that the usual woods used in commercial structures will act in the same way with one exception, *i.e.*, the matter of bulk, for large members will not succumb to disintegration so quickly as smaller ones owing to carbonization of outer fibers whereby a protective layer of carbon is formed which will protect or envelop the inner layers and prevent further volatization. Therefore, the degrees of heat necessary to ignite the so-called "wood" would depend upon the construction or kind of wood and its bulk or volume.

It is a well-known axiom that wood which is slowly heated is more susceptible to a lower degree of ignition temperature than woods heated very rapidly. This is entirely due to volatization of the resins. In the former, this causes a complete volatization, leaving the cellulose in a very dehydrated state. In the latter it merely brings to the surface the resins which burn at their ignition point but eventually ignites the cellulose structure, thus bringing up the interesting subject of the ignition points of resins, aromatics, and cellulose, which vary in considerable degree.

Ordinarily it may be assumed that at 110°C the water and aromatics are dispelled. At 150°C the carbon monoxide (CO), carbon dioxide (CO_2), and resins are dispelled. At 230°C the external browning or carbonization starts. At 270°C pyrophoric carbon is formed. This is pure cellulose in punk form which has a tendency to self-ignition while undergoing this heat process. At 300°C charcoal or carbonized cellulose is formed. Where there is an actual contact of flame, the process is very rapid but the cycle remains the same.

Table 5-56. Density of Gases Lighter or Heavier Than Air (Von Schwartz)

(Density of air taken as unity: g = gas, v = vapor)

	Lighter	*than air*	
Hydrogen (g)	0.069	Wood gas (g)	0.700
Coal gas (g)	0.400–0.600	Acetylene (g)	0.898
Pit gas (firedamp)	0.553	Hydrocyanic acid (g)	0.945
Ammonia (g)	0.588	Ethylene (g)	0.967
Water vapor (v)	0.622	Carbon monoxide (g)	0.967
	Heavier	*than air*	
Ethane (g)	1.036	Butane (g)	2.004
Nitric oxide (g)	1.039	Carbon oxysulfide (g)	2.104
Oxygen (g)	1.105	Sulfur (v)	2.200
Methyl alcohol (v)	1.120	Chlorine (g)	2.448
Phosphuretted hydrogen (g)	1.185	Ether (v)	2.565
Sulfuretted hydrogen (g)	.192	Carbon disulfide (v)	2.645
Allylene (g)	382	Arseniuretted hydrogen (v)	2.697
Propylene (g)	451	Benzol (v)	2.770
Propane (g)	.520	Seleniuretted hydrogen (g)	2.784
Nitrous oxide (g)	.527	Amyl alcohol (v)	3.147
Ethyl aldehyde (v)	.530	Chloroform (v)	4.215
Hyponitrous acid (g)	.590	Phosphorus (v)	4.355
Alcohol (v)	.613	Telluretted hydrogen (g)	4.498
Methyl ether (g)	1.617	Selenium (v)	5.700
Methyl chloride (g)	1.738	Sulfur vapor	6.650
Cyanogen (g)	1.800	Tellurium	8.896
Butylene (g)	1.935		

Gas or vapor	*Approximate vapor or gas density, air = 1*
Hydrogen	0.069
Carbon monoxide	0.96
Alcohol, methyl	1.1
Alcohol, ethyl	1.6
Acetone	2.0
Ether, ethyl	2.6
Carbon disulfide	2.6
Benzene	2.7
Gasoline	3.5*
Kerosene	4.5*

* NOTE: These are mean values for gasoline and kerosene found in common use.

Hot Ashes and Coals (direct ignition with flaming or glowing materials or sparks). These causes of fire are readily apparent, as they involve one source, that of direct ignition from flaming or glowing material. Ashes or coal removed from furnaces contain sufficient heat and in many cases visible fire to ignite material with which they come in contact.* Placing them in wooden receptacles or in contact with combustible material is inviting disaster as is also placing them or permitting them to rest on metal-covered wood.

Ignition of Hot Greases, Tar, Wax, Asphalt (open lights, sparks, flames, and static electricity). Flammable materials, as has been stated, vary in their flash and ignition points, and fires resulting from the use of such materials are to be expected if they are subjected to heat or flame which will ignite them. The above-mentioned materials are in addition subject to ebullition and frothing, which cause them to overflow their containers, whether a frying pan or a pot used in industrial establishments. Flames or open lights are the usual source of ignition, and fire results from contact with these owing in most cases to lack of proper supervision and care, unless a defective vessel should break, permitting the contents to fall into the flame of the heating medium. This latter disaster is avoided only by frequent inspection.

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* Ashes from a furnace piled on a concrete floor against a brick wall 12 in. thick caused a fire owing to the radiation of heat through the wall to a wooden guide track of an elevator on the other side.

Hot greases, tar, wax, and asphalt materials are flammable. Their flash points and ignition points are higher than petroleum oil or gasoline, but they will ignite at their own proper temperature. This should be remembered in order to safeguard them from any and all flame. Their use is very broad, so it is hard to lay down any set rules for avoiding fire except the one above. If they are being heated for any purpose, careful watching is necessary to avoid overheating. This is very essential if an open flame is used. Containers should also be carefully examined for cracks or defects.

Gas, Natural and Artificial (see Flammable Gases). (Open flame, sparks, pressure, lightning and static electricity.) Since the fire record encompasses the whole country, the fires from gas as a cooking and illuminating medium would naturally include natural and artificial gas. For the purpose in hand, it is not necessary to consider the technical fire hazards of gases, as these are well understood by those who know about the manufacture of artificial gas or the control of natural gas. To those who use these gases for industrial or household purposes, however, a word of warning seems not untimely, as the record shows that greater care is necessary if fires are to be avoided from this source.

Natural and artificial gas are explosive, and the greatest need for care is in avoiding leaks or the accumulation of these gases in low spots or enclosures. Fires result from these gases being ignited by flame or sparks. Open lights, matches used in locating leaks, defective electric wiring, or friction from machinery will ignite them causing fire or explosion. Danger from these gases is present when they come in contact with an ignition medium, and although the cause of ignition and consequent fire is given as gas, it is in reality the open flame or spark, lightning, or static electricity. Caution should be exercised in looking for leaks. (See Table 5-56.)

FIRE PROTECTION

The extinguishment of fire in industrial plants is an essential responsibility of plant management. It will not be a success unless recognized by the executive management as an important permanent addition to the plant operations. Regardless of the size of the plant or establishment, the personnel and workmen should be organized for the purpose of preventing and extinguishing fire. In some plants the plant manager will be the fire-prevention engineer, whose duties are confined to the analysis and correction of fire hazards. The extinguishment and fire fighting will be delegated to a fire chief and private fire brigade. In other plants the analysis and correction of fire hazards as well as the fire fighting are delegated to the fire chief and the plant fire brigade. In still other plants the safety engineer will include in his duties the analysis and correction of fire hazards, and the fire fighting will be delegated to the fire chief and plant fire brigade. In setting up "the defense against fire" within the plant, it is necessary for whoever is responsible for the defense to have a knowledge of those things which comprise the defense. These are

1. A ground plan of the property and all buildings thereon
2. A plan of all available water supplies (public and private), including water mains, reservoirs, sumps, and tanks, with capacities and pressures available at ground level
3. A diagram of all controlling valves, check valves, pumps, hydrants, and hose houses
4. First aid equipment, fire extinguishers
5. Standpipes
6. Yard hydrants and hose streams
7. Automatic sprinkler systems, foam and waterspray or fog
8. Alarm systems

9. **Plant fire brigade**
10. **Attack**
11. **Stock flow method**

Ground Plan. The ground plan should include the boundary lines of the property and plans of all buildings thereon, using the map symbols, as given on previous page, for types of buildings, heights, stairways and elevators, shafts, and window protection.

Plan of Available Water Supplies. The plan of available water supplies should show the water mains by sizes (and open-butt flow tests at ground level). It should also show standpipes and their gauge pressures at top outlets; sprinkler risers and their gauge pressures at top outlets; and reservoirs, sumps, and tanks, with their capacities and pressures at ground level, if elevated.

Diagram of Valves. The diagram of valves should show all controlling valves and check valves (above- and belowground) and post-indicator valves (OS and Y valves on sprinkler system). If single-floor control, it should be so indicated. The diagram should show pumps, indicating the type, capacity, type of power, and location; hydrants, their location, number of outlets, and pressures; and hose houses (if over hydrant) with the quantity of hose in each and their size.

First-aid Equipment.[29] The primary purpose of first-aid equipment is its immediate application to an incipient fire. To accomplish this, it is essential that those who are charged with its application be thoroughly familiar with the characteristics of the portable equipment. It is recommended that all employees and members of the fire brigade be given an opportunity to handle and operate the equipment at drills and at time of recharging so that they will know how they operate, their ingredients, their weight, and the purpose for which they are best suited. This drill will tend to make their attitude one of confidence and familiarity with the instrument.

Fire Extinguishers. The action of portable extinguishing mediums consists in cooling the burning substance to below its ignition temperature, excluding the air supply, or a combination of both, and in excluding arcs in electrical equipment Portable fire extinguishers should be purchased only if they bear the approval of some authoritative testing laboratory such as Underwriters' Laboratories, Inc , or the Factory Mutual Laboratories, not only to ensure the performance reliability and suitability of the device but to ensure safety, as the pressures developed require sufficient strength in the materials used. The quenching, cooling, and blanketing effects are set forth in the plan developed by Underwriters' Laboratories as follows:

Classification of Fires

Class A fires—Fires in ordinary combustible materials, when the quenching and cooling effects of quantities of water or solutions containing large percentages of water are of first importance

Class B fires—Fires in flammable liquids, greases, etc., where a blanketing effect is essential

Class C fires—Fires in electrical equipment, where the use of a nonconducting extinguishing agent is of first importance

Extinguishers are further classified in units of first-aid fire protection, depending on their relative extinguishing capacity under average conditions of use.

As a guide in determining the number of units that should be installed under various conditions, the following recommendations are given:

Class 1—Light-hazard occupancies. Offices, schools, public buildings, etc. Units should be so located that a person will not have to travel more than 100 ft from any point to reach the nearest unit. At least one unit should be required for each 5,000 sq ft of floor area.

Class 2—Ordinary combustible occupancies. Department and dry-goods stores, warehouses, manufacturers, of average hazard. Units should be so located that a person will not have to travel more than 50 ft from any point to reach the nearest unit. At least one unit shall be required for each 2,500 sq ft of floor area.

Class 3—Extra-hazardous occupancies. Woodworkers, paint spraying, dipping, etc. Units shall be so located that a person will not have to travel more than 50 ft from any point to reach the nearest unit. At least one unit shall be required for each 2,500 sq ft of floor area *and any additional units as the severity of the hazard may require.*

Wheeled extinguishers, provided for larger hazards, shall be located within 200 ft of all ordinary hazards or within 75 ft of extra hazards. Classification or designation (in standards) A-2 indicates that the extinguisher is suitable for class A fires and that two extinguishers are required to make a unit.

Fire pails—five 12-qt pails equivalent to one 2½-gal soda-acid extinguisher

Bucket tanks—six 10-qt pails, 40 gal equivalent to one 2½-gal soda-acid

Sand pails—Same as water pails

The manufacturers' instructions as to filling and maintenance should be followed out carefully. All extinguishers should be inspected once a year and tagged.

Use of Extinguishers on Electrical Equipment. Streams directed on a metal plate at 550 volts, alternating current to ground, gave results as shown in Table 5-57.

Table 5-57

	Distance current was conducted
Calcium chloride	6 ft
Soda acid	4½ ft
Foam	5 in.
Water	2 in.
Carbon tetrachloride; carbon dioxide, CO_2; Dry chemical;	Nonconductors at all distances

Streams directed at 550-v switchboard panels having live bus bars and cutouts showed that foam and calcium chloride caused sufficient arcing on the switchboard to blow the fuses.

Table 5-58. Weights of Typical Fire Extinguishers Charged and Ready for Use

Type	*Approx. weight, lb*
1½ gal soda acid, foam or water	20-25
2½ gal soda acid, foam or water	35
2½ gal pump tank	40
5 gal pump tank	65
5 gal foam	70
2½ gal calcium chloride or loaded stream	40–45
1 qt-vaporizing-liquid carbon-tetrachloride	7
1 gal vaporizing-liquid carbon-tetrachloride	25–35
3 gal vaporizing-liquid carbon-tetrachloride	75
7½ lb CO_2	35
20 lb CO_2	55–75
7½ lb dry chemical	40
20 lb dry chemical	55

Soda-acid Extinguisher for Class A Fires. Soda-acid extinguishers are made in two principal sizes—2½ gal liquid capacity and 1¼ or 1½ gal liquid capacity. The materials used are a dry chemical, bicarbonate of soda, designed to be dissolved in water, and a liquid chemical sulfuric acid (H_2SO_4). Reaction of the acid and the bicarbonate of soda produces pressure which expells the liquid from the extinguisher a horizontal distance of 30 to 40 ft at a rate of 2½ gal in 1 min. Protection against freezing consists of a suitable heated cabinet or felt-lined boxing. Antifreezing solutions should not be used in the extinguisher, as they interfere with the operation.

Table 5-59. Antifreezing Solutions, to Make 2½ Gal

Approx. freezing temperature, deg F	Water	Calcium chloride	Specific gravity
10	2 gal, 1 qt	5 lb	1.138
0	2 gal, 1 pt	6¼ lb	1.175
−10	2 gal	7 lb 6 oz	1.205
−20	2 gal	8 lb 6 oz	1.228
−30	2 gal	9 lb 2 oz	1.245
−40	2 gal	10 lb	1.263

Foam Extinguisher for Class A and B Fires. Foam extinguishers are made in two sizes—1½ gal liquid capacity and 2½ gal liquid capacity. The materials used are a dry chemical, bicarbonate of soda, a foam stabilizing agent (soap bark) dissolved in water for the outer compartment, and aluminum sulfate dissolved in water for the inner cylinder. Reaction between the aluminum sulfate and the bicarbonate of soda produces carbon dioxide gas in the form of bubbles with a light precipitate of aluminum hydroxide in the bubble film, which will expel the foam a horizontal distance of 30 to 40 ft at a rate of 2½ gal in 1 min. A 2½-gal extinguisher will produce 18 to 20 gal of foam at ordinary room temperature. Protection against freezing is the same as for soda-acid extinguishers.

Loaded-stream Extinguisher for Class A and B Fires. Loaded-stream extinguishers are made in three sizes—1 gal liquid capacity, 1¾ gal liquid capacity, and 2½ gal liquid capacity. There are also 17- and 33-gal liquid capacity extinguishers in wheeled unit. The material used is a special solution of alkali metal salts, which is a trade secret. *It is essentially an antifreeze solution.* Pressure to expel the liquid is produced by either a CO_2 cartridge or by chemical action. The stream reaches a distance of 30 to 40 ft at a rate of 2½ gal in 1 min. The alkali metal salt solution has an effect on fire that is unlike any other extinguishing agent. On class A fires, the flame is extinguished rather suddenly, and there is a pronounced fireproofing effect in that flash back is materially retarded. The effect on small class B fires has not been explained. There is no smothering vapor to blanket the fire, but there is a chemical action tending to inhibit oxidation. This extinguisher must be recharged annually and after use.

Plain Water Extinguisher for Class A Fires. Plain water extinguishers are made in 2½ gal liquid capacity. The material used is plain water, and a CO_2 cartridge is used to provide the pressure. The cartridge should be weighed and tested for leaks. The stream reaches a distance of 30 to 40 ft at a rate of 2½ gal in 1 min.

Calcium Chloride Extinguisher for Class A Fires. Calcium chloride extinguishers are made in two sizes—2½ gal liquid capacity and 5 gal liquid capacity. Material used is plain water or an antifreezing solution of granulated or flake calcium chloride and water. The pressure to expel the liquid is produced either by a slow-burning fuse cartridge, a CO_2 cartridge, or by chemical action. The stream reaches a distance of 30 to 40 ft at a rate of 2½ gal in 1 min.

Pump Tank Extinguisher for Class A Fires. Pump tank extinguishers are made in two sizes—2½ gal liquid capacity and 5 gal liquid capacity. The material used is the same as in the calcium chloride extinguisher. Pressure to expel liquid is by a hand pump installed in tank.

Vaporizing Liquid Extinguisher for Class B and C Fires Vaporizing liquid extinguishers (carbon tetrachloride base) are made in four sizes—1, 1¼, 1½, and 2 qt liquid capacity. The material used is a specially treated nonconducting liquid having a carbon tetrachloride base, with components added to depress the freezing point to −50°F and to avoid corrosion. The liquid is heavier than air and will not support combustion. Carbon tetrachloride vapor has an anaesthetic effect and if subjected to high temperature decomposes to some extent, forming toxic gases such as hydrochloric acid and phosgene ($CO + CL_2 = COCL_2$). When it is used, caution should be taken to avoid breathing the vapors or gases produced. The stream reaches a distance of 20 to 30 ft at the rate of 1 qt in 1 min The stream of liquid rapidly evaporates to form a smothering vapor. The quantity of vapor from 1 lb of liquid is approximately 2½ cu ft at room temperature. These extinguishers are made in 1-, 2-, and 3-gal capacity for special use, in pump-type and stored-pressure-type extinguishers, using either air or CO_2 gas.

Carbon Dioxide Extinguisher for Class B and C Fires. Carbon dioxide extinguishers are made in six sizes—2, 4, 7½, 10, 15, and 20 lb. The material used is CO_2 under pressure in a liquid condition. The stream reaches a distance of 3 ft at a rate of 2 lb in 11 sec. Carbon dioxide at ordinary temperature and atmospheric pressure is a gas, 1½ times as heavy as air. It is inert and will not support combustion. The pressure in the container is the vapor pressure of CO_2, which varies with the temperature. It is reduced to a liquid by pressure if the temperature is below 88°F, the critical temperature of CO_2. At ordinary room temperature (68°F) the pressure is between 800 and 900 psi. One pound of liquid CO_2 converted to a gas will occupy approximately 8.6 cu ft. Carbon dioxide gas, like any inert gas, will produce suffocation owing to oxygen deficiency. The extinguisher should be weighed annually for leaks. A loss of 10 per cent is normal.

Dry-compound Extinguisher for Class B and C Fires. Dry-compound extinguishers are made in three sizes—7½, 12, and 20 lb. The material used is bicarbonate of soda which has been chemically processed to make it waterproof and free flowing. Pressure is generated by a CO_2 gas cartridge. The stream reaches a distance of 14 ft at a rate of 7½ lb in 17 sec. Dry-compound wheeled engines are available in 150- and 350-lb sizes, with pressure furnished by nitrogen gas. The streams from these reach a distance of 35 ft at a rate of 150 lb in 1 min and 350 lb in 3 min, respectively. Dry-compound extinguishers should be inspected every 6 months.

Fire Pails for Class A Fire. The 12-qt pail is standard.

Sand Pails for Class B Fire. A 12-qt, fine, dry, clean, sand scoop should be provided (see Antifreezing Solutions for unheated areas).

Soda and Sawdust for Class B Fires. Bicarbonate of soda and sawdust in bins with scoop is the usual equipment.

Glass-bulb and Bottle Extinguisher for Class A and B Fires. Carbon tetrachloride is used in bulbs or bottles. This type of extinguisher is used in confined spaces or applied directly on incipient fires (see Vaporizing Liquid Extinguisher for physical properties).[30]

Standpipes. A well-designed, -equipped, and -maintained standpipe system constitutes one of the best means for the extinguishment of fire in buildings as it is the only reliable means of obtaining effective fire streams at the upper stories of high buildings in a short space of time.

Standpipe systems are designed for two types of service—for small hose streams for use of occupants of building and for large hose streams for use of public fire departments or plant brigades.

For small hose streams for use of occupants, 1½-in. unlined linen hose, with ⅜- or ½-in. nozzle, with not more than 100 ft at each outlet should be provided. The hose should be coupled to a control valve, either straightaway or globe and carried in an approved rack. Drip connections should be provided as hose is liable to rot at the coupling, gaskets should also be replaced at intervals. Where the pressure exceeds 50 psi, an adjustable reducing valve should be provided. For large hose streams for public fire department or plant brigade 2½-in. unlined linen hose with 1- or 1⅛-in. nozzle should be used. In large systems, the hose may not be attached, as the fire department may prefer to bring in its own hose. Hose can also be stored at various floor levels for use of public departments. Reducers can be provided if both small and large hose are to be used on the same system. The smaller size hose is sometimes attached to automatic sprinkler systems.

The size of standpipes is governed by the size and number of fire streams needed and by the distance of the outlets from the source of water supply. Standpipes for small hose streams should not be less than 2 in for buildings 55 ft in height; 4 in. for buildings 55 to 75 ft in height, 6 in. for buildings 75 ft or over in height; and 2½ in. for theaters on both sides of stage.

The number of standpipes should be arranged so that all parts of every floor can be reached within 30 ft, from a nozzle attached to 100 ft of hose connected to a standpipe, *e.g.*, an area covered by interlocking circles having a diameter of 130 ft. Outlets for occupant use should be located in floor area. Standpipes should be located in fire-resistive stairways, or near fire escapes or fire towers so as to be available for public fire department's use.

Water Supply for Standpipes. The value of the water supply for standpipes should be considered in the following order: City water-works system, where the domestic pressure is adequate, should be not less than 250 gpm for one riser or 500 gpm for two or more risers. The supply should deliver 1 hr pressure at 40 to 50 lb at topmost 2½-in. outlet, discharging through 50 ft of 2½-in. CRL hose with a 1⅛-in nozzle. Twenty pounds at topmost outlet flowing pressure is the minimum acceptable It is a practice to add 25 lb to pressure requirements for all standpipes.

Where fire pumps are required, for buildings up to 150 ft high, the sizes should be 500 gpm for 4-in. standpipe; 750 gpm for 6-in. standpipe or two 4-in. pipes; and 1,000 gpm for two or more 6 in. pipes.

At least one fire-department connection should be provided for each standpipe.

Water pressure should be maintained at all times in standpipes. If building is unheated, water should be controlled by manually operated remote-control valves or automatic dry valves. Monitor nozzles and hose houses can be placed on roofs.

Yard Hydrants and Hose Houses.[31] Where hose is kept connected to hydrants in hose houses, a fire line can be laid and water turned on in approximately one-half minute, as compared with 2 or 3 min where a hose cart must be run up, the hose coupled to the hydrant, run out, and the nozzle attached, before the water is turned on.

The standard mill-yard hose house embodies the features of protection against the weather and ventilation, the ventilation being secured through slatted floors and shelves and vent spaces under the overhanging roof, so arranged so that rain cannot drive into the building. At least 100 to 150 ft of 2½-in. cotton-rubber-lined hose should be attached to one outlet. One shelf in a standard hose house, will hold 150 to 200 ft of single-jacket cotton-rubber-lined hose, folded forward and back, so placed that it can be readily inspected. To determine the length of hose required in a hose house, a line should be laid in easy curves from the hydrant to the building, through the door, up the stairs, to the top story of the building, and then a stream played from a far window. A hose should also be taken up an extension ladder to the roof, allowing one length (50 ft) on roof. Make sure that hoisting ropes and hose straps are kept in hose house. There should also be sufficient hose available to reach to the far side of a building, remote from the hydrant, and a safe and effective distance from the building. If it is necessary to lock the hose house, a special lock with brittle shackle should be used or a latch placed behind a glass plate which can be broken easily.

Accessory Equipment for Hose Houses. Accessory equipment for hose houses should consist of the following:

Standard play pipe (one for each hydrant outlet) 30 in. long with swivel handles and 1⅛-in. smooth-bore tip
2 play-pipe holders
4 to 6 tabor spanners
1 fire axe weighing 6 lb
1 1¼-in. by 5-ft crowbar
2 hose straps
1 extra hydrant wrench
1 heavy mill lantern kept filled and ready for use
4 to 6 extra washers for hose couplings
1 hand line of ½-in. manila rope, long enough to reach roof of highest building

Public and Private Water Supplies. The following data may be helpful to private plant brigades:

The capacity of any public water system is determined by the total amount of water it must furnish. This is the sum of

1. Water required for domestic or industrial use
2. Water required for fire service

In rural districts the water problem has these phases:

1. The population needs, domestic, near centers
2. The population needs, domestic, away from centers
3. The fire-protection requirements which involve small congested areas spreading fires owing to shingle roofs and brush fires

In cities, the problem has three phases:

1. The population needs, domestic
2. The fire service needs, domestic and industrial
3. The fire service needs, conflagration

The general domestic water consumption will range from 40 to 250 gal per capita, varying with the peaks of daily consumption and seasonal variations, including wastage from underground piping and carelessness in private use.

Fire service or fire flow based upon average conditions found in communities of various sizes is shown in Table 5-60.

A fire requiring 10 streams for 5 hr would use 750,000 gal of water. A standard fire stream is taken at 250 gpm, 50 psi, with 1⅛-in. standard play pipe.

To determine the reliability of the water supply, the following should be taken into consideration.

1. Frequency of and duration of drought
2. Condition of intakes
3. Danger from earthquakes or seismic disturbances
4. Physical injury
5. Reservoirs out of service for cleaning
6. The condition, arrangement, and reliability of individual units of plant equipment, such as pumps, engines, generators, electric motors, fuel supply, and electrical-transmission lines
7. Combustible construction of pumping stations
8. Machine shops connected with maintenance

Table 5-60. Required Fire Flow for Average City, Gpm*

Population	*Fire flow, gpm*
1,000	1,000
2,000	1,500
4,000	2,000
6,000	2,500
10,000	3,000
13,000	3,500
17,000	4,000
22,000	4,500
28,000	5,000
40,000	6,000
60,000	7,000
80,000	8,000
100,000	9,000
125,000	10,000
150,000	11,000
200,000	12,000

NOTE. Over 200,000 population, 12,000 gpm with 2,000 to 8,000 additional for a second fire.

* NBFU.

Spacing of Hydrants. Hydrants and pipes are cheaper than hose in the long run, so frequent hydrant spacing is desirable.

The minimum satisfactory pressure at which water can be supplied for fire pumpers is 20 lb at the hydrant. If supply for automatic sprinklers is taken direct from municipal water mains or yard systems, normal pressure should give 15 lb on top line of sprinklers with 250 to 500 gal flowing. For extra hazards, pressures from 50 to 75 lb are desirable.

Table 5-61

Fire flow required, gpm	Average area per hydrant, sq ft	
	Engine streams	Direct hydrant streams
1,000	120,000	100,000
2,000	110,000	85,000
3,000	100,000	70,000
4,000	90,000	55,000
5,000	85,000	40,000
6,000	80,000	40,000
7,000	70,000	40,000
8,000	60,000	40,000
9,000	55,000	40,000
10,000	48,000	40,000
11,000	43,000	40,000
12,000	40,000	40,000

Piping. Piping should be no less than 6 in. in diameter for fire service. Depth of cover to prevent freezing will vary from 2½ ft in the South to 10 ft in the North.

Hydrants. Hydrants should not be distributed more than 300 or 400 ft from buildings to be protected, as hose lines of 400 or 500 ft are difficult to handle. If hose streams are to be used direct from hydrants, the spacing should be 100 ft less. Hydrants should be placed 50 ft from building to prevent damage from falling walls. *Testing hydrants* for freezing: may be accomplished by sound by striking the hand over open outlet; water or ice shortens the length of sound tube and raises the pitch. The stem should be tried; if frozen solidly, it will not turn. Tapping on hydrant nut may release it if it is not frozen solidly. If a weight is lowered on a string into hydrant, it may strike ice or water, shown if the weight is wet. Steam should be used to thaw out the hydrant.

Valves. Fire-prevention inspectors must be valve-conscious. Many plants have been destroyed, because a valve was closed.

Reservoirs. Reservoirs form a good source of water supply if adequate. A fire flow of 10 hr is adequate for large cities or similar needs. A fire flow of 5 hr is adequate for cities of 2,500 population or with similar needs. Auxiliary sources consist of ponds, swimming pools, cooling tanks, rivers, and streams. Streams of 1,000-gpm flow are acceptable, with suitable approach for fire-engine suction. For a fire pump of 1,000-gpm rating, a tank or reservoir for suction should have a 100,000-gal capacity.

Gravity Tanks. Gravity tanks are usually built of wood or steel and supported on steel towers. If there is exposure within 20 ft of tower, steel should be fireproofed.

All brush should be cleared from vicinity of tower. Steel tanks of 50,000- to 100,000-gal capacity should have four-column support. Steel tanks of 100,000- to 300,000-gal capacity should have six-column support. Steel tanks of over 300,000-gal capacity should have eight-column support.

If the tank is directly connected to fire service and is to supply hose lines from hydrants, a minimum capacity of 30,000 gal with the bottom of tank not less than 75 ft above ground is required. For unsprinklered buildings where hose streams are relied upon for protection, the bottom of the tank should be 40 ft above the roof of the highest building.

Supports for tanks should preferably be in the ground. If supported on buildings, they should be located over noncombustible stairways or an elevator tower. Buildings should be examined by competent engineers to determine their ability to sustain the load including trestle or towers. Drops should be protected from freezing by suitable boxing. Tank lumber can be treated against rot by preservatives such as sodium fluoride, creosote, or zinc chloride. Tanks can be heated by steam from boilers at not less than 10 lb of pressure. Pits containing OSandY gates and checks should be heated to 40°F during freezing weather. Water-level indicators are of the float or mercury-gauge type, but when there is doubt, the tank should be flooded. Flat hoops, subject to rust, on wooden tanks are obsolete (see Hurricanes).

Pressure Tanks. Tank capacity of pressure tanks is the total combined, including water and air, not including dish ends. Tank capacity should be 4,500 gal for light occupancy, but 3,000 gal is acceptable. Capacity should not exceed 9,000 gal. It is preferable to use two 4,500-gal tanks. Tanks should be kept two-thirds full of water, and an air pressure of 75 psi maintained. When the bottom of the tank is located below the highest line of sprinklers, the air pressure should be three times the pressure caused by the column of water in the sprinkler system above the tank bottom, plus the 75 lb. Pressure tanks should be tied into system 40 ft below the bottom of pressure tank and gravity tank to prevent water lock. They should be installed in incombustible buildings, kept painted, and given a hydrostatic test every 2 years.

Air Compressor. An air compressor should be capable of delivering 16 cu ft of free air per minute for tanks of 7,500-gal total capacity; 20 cu ft of free air for larger sizes. Steam or electrically driven compressors should be located in tank houses

Fire Pumps (specified by Underwriters' Laboratories, Inc., or Associated Factory Mutual Laboratories). Fire pumps should be located in rooms or buildings so constructed as to prevent damage from falling walls. floors, or machinery and from smoke, fire, and water which may prevent their operation. Troubles with pumps can be traced to defects in suction lines. Suction pipes should be tight and should be laid to true grade. Air traps should be avoided. The greater the lift, the less the length of the suction should be. Foot valves should be provided for all centrifugal pumps and for rotary and steam pumps having a suction lift of over 10 ft or suction pipe over 100 ft long with any lift. Screens should be provided, where needed, with an approximate area of ten times the area of the suction pipe.

Pump Capacities (feeding Automatic Sprinklers only). For light hazards, the pump capacity should be 250 gpm, for ordinary hazards, 500 gpm; and for heavy hazards, 1,000 gpm. Occupancy, construction, size, and value of property should be considered.

Steam Pumps. Steam pumps should conform to the specifications for Underwriters steam-fire pumps published by the Factory Mutual Laboratories. See also the NBFU Standards.

To test for condition of steam pumps, all valves should be closed, the pop valve should be set at 100, if piston moves, the travel should be measured (140 fpm is usual travel).

Centrifugal Pump. The centrifugal pump is a casing in which a disk or impeller carrying vanes rotates at high speed. By increasing the speed of an impeller, by increasing its diameter, or by arranging two or more impellers in series (multistage), higher pressures may be obtained. The pump can be driven by an electric motor, gasoline engine, or steam turbine, since the speed of the pump and prime mover may be the same, thus permitting direct connection.

Table 5-62. Standard Steam-pump Sizes for 1,000 Gpm

Pump sizes, in.			Ratio	Capacity at 100 lb at pump	Boiler power required, hp steam 150 pres. at pump, lb	Full speed, rpm	Piston travel, fpm
Steam	Water	Stoke					
18	10	12	3:1	1,000 $\frac{999}{1{,}050}$	45	70	140

Pumps are made in six sizes—500, 750, 1,000, 1,500, 2,000, and 2,500 gpm.

Maximum speeds for centrifugal fire pumps are usually 1,800 rpm, but in some cases, speeds are as high as 3,600 rpm. The minimum-size fire pump recognized is 500 gpm for supplying standpipes and sprinklers; 750 gpm, for supplying both sprinklers and standpipes; and 750 gpm, for supplying hydrants.

In light hazards pumps with a capacity of 250-gpm for sprinklers only and pumps of 500-gpm capacity should supply sprinklers and standpipes.

Pumps should be primed by a 250-gal tank or other reliable source.

Rotary Pumps. The rotary pump is simple in construction and inexpensive, but wasteful of power. It is generally of the double-shaft (or twin) type with cams or buckets which intermesh, the pump being direct-connected by a spur gear or gear clutch or by a friction gear. The cams are keyed to the shafts as are the gears. The gears and cams are so fitted that when the gears are properly in mesh, the cams just touch each other. Cams cannot be fitted absolutely tight, and there is an unavoidable slip or passing of water back into the suction. Bronze fittings have eliminated wear and corrosion. The rotary pump takes water under a slight head. Pulsation is reduced by having air chambers attached. Discharge outlets should not be less than 6 in. for 500-gpm pumps; 8 in for 750- to 1,000-gpm pumps; and 10 in. for 15,000-gpm pumps. Stuffing boxes should be checked for leaks. Bearings should be lubricated.

Booster Pump. Booster pumps with a capacity of from 30 to 100 gpm are used in conjunction with a water tank of 80-gal capacity. An 80- or 100-gal tank takes the same space as a 40-gal chemical tank. Rural districts use tanks of 300 gal or larger. Some fire departments carry lines of $1\frac{1}{2}$-in. hose with $\frac{1}{2}$- or $\frac{5}{8}$-in. nozzles connected to large booster tanks for use in dwelling and brush fires. The booster pumps supplied by OCD are centrifugal gasoline-driven pumps on trailers or tied onto front of motor trucks.

Automatic Sprinkler Systems. An automatic sprinkler system is the extension of the public water system or private-supply installations to every part of a building by a network of piping of reducing sizes, with automatic spray valves placed at intervals on this piping system, which, when activated by the convected or radiated heat of a fire, permit water to be sprayed upon a predetermined area. The design of the system is based upon the nature of the occupancy; the possible fire intensity, which includes the character of the raw stock, the process, and the finished product; the area exposed, the internal exposure to fire spread, and the external exposure to fire attack; and the character of the building that houses the occupancy.

Types of systems designed to meet these requirements are automatic wet system, automatic dry system, automatic deluge system, thermostatic preaction system, open sprinkler system, and nonfreezing solution system.

The auxiliary water supplies, which enter the design of the systems, are taken from gravity tanks and pressure tanks and from underground sumps. To augment the water pressure from public mains and private water supplies, the design may include pumps, either steam or electrically driven, of the piston, cam, or vane type, known as positive displacement, rotary, or centrifugal, respectively.

The standard requirement for automatic wet systems is that water shall be delivered to the spray valve or sprinkler head, located on the highest pipe lines, at not less than 15 psi with a flow of not less than 100 gpm per head for all heads in the largest single fire area. These requirements are minimum for ordinary occupancies, but where the heat intensity, owing to character or quantity of the combustible material present, warrants, they are augmented to effect the control of fire.

The design of the system must also take into consideration the type of building construction, particularly the floor spaces, bays, and ceiling contours, vertical and horizontal openings, roof type (peak or saw tooth), skylights (monitor, Texas), and roof structures. The shape of the floor area is also important as this determines the layout of the piping network.

The internal temperatures which result from the heat intensity of the occupancy and all the equipment it entails in the effective control of fire are based upon the speed of operation of the sprinkler head activated by these temperatures. The external temperatures (weather) are important factors in the design, as the freezing of water in any part of the system will impair its effectiveness.

The above requirements may be termed the basic factors of design, and upon these the system is laid out to protect the single structure, series of structures, or multiple fire divisions in a single structure.

The foremost intent of the installation of an automatic sprinkler system, however, is "the extinguishment of fire in its incipient stage," and the design of the system should be such as to accomplish this purpose.

Table 5-63. Sprinkler Pipe Sizes*

Size of pipe, in	Light	Ordinary	Extra hazard
¾	Eliminated		
1	2	2	1
1¼	3	3	2
1½	5	5	5
2	10	10	8
2½	40	20	15
3	No limit	40	27
3½	No limit	65	40
4	No limit	100	55
5	No limit	160	90
6	No limit	250	150
8	No limit		

* NFPA 1940 Standard.

The details of the design of the system must take into account the adequacy and control of water supply; fire areas; test devices for water flow due to fire or leakage; alarms and detectors; air supply for dry-pipe systems; accelerators and exhausters; thermostats for preaction systems; valves, check valves, dry valves, heaters for tanks, meters on public supplies, post-indicator valves for the control of sections; protection against freezing of tank drops; corrosion and pollution, hose attachments, building-construction details; tightness of floors; curtain boards; scuppers; steam pressure for pumps; electrical equipment; supervisory systems such as ADT to call outside aid or fire department; yard hydrant system for outside protection; hose houses; maintenance and supplies, additional heads; and necessary tools to make repairs.

The 1940 standard provides for three classes of occupancy.

A—Light-hazard occupancies. This includes apartment houses, asylums, clubhouses, colleges, churches, dormitories, dwellings, hospitals, hotels, libraries, museums, office buildings, schools, and tenements.

B—Ordinary-hazard occupancies. This includes mercantiles, warehouses, manufacturing, and occupancies not covered by A or C.

Table 5-64

Flowing pressure, psi	*Discharge, gpm*
5	12 80
10	18 10
20	25 80
30	32 80
40	37 20
50	42 20

C—Extra-hazard occupancies. This includes only those buildings or portions of buildings where the hazard is severe as determined by the inspection department having jurisdiction.

It will be seen by the foregoing that in the 1940 standard consideration was given to the theory of heat intensity and quantity of combustible material as determined by the occupancy. In terms of immediate extinguishment, this will call for wider or closer distribution of water but will not affect the quantity of water flowing from the sprinkler head other than the quantity that would flow under a given pressure. The Underwriters' Laboratories' test in 1926 indicated the quantity of water per head as shown in Table 5-64.

Formulas for the discharge from orifices, according to Factory Mutual Laboratories, are as follows:

$$\text{Gpm} = 29.83CD^2\sqrt{P}$$

where C = coefficient of discharge
D = diameter of outlet, in.
P = pitot-tube pressure, psi

Coefficient = 0.90 for smooth and well-rounded outlet
0.80 for square and sharp outlet
0.70 for square outlet and outlet projecting into barrel

For ordinary use $29.7D^2\sqrt{P}$ can be used. For sprinkler heads *only*,

$$\text{Gpm} = 5.6\sqrt{P}.$$

For required quantities on top lines gpm = $\frac{1}{2}P + 15$ lb. Average flow in sprinkler fires = 5,000 gal, which would fill a tank 12 by 12 by 5 ft. Standard sprinkler heads have $\frac{1}{2}$-in. ring nozzles or $\frac{7}{16}$-in tapered nozzle.

Sprinkler Valve or Head. There are approximately six types of sprinkler valves or heads

1. Link and lever
2. Frangible (fragile) bulb, nonsolder
3. Spring, lever, and strut
4. Duckbill
5. Chemical, nonsolder
6. Triangular strut

The frangible (fragile) bulb nonsolder-type head is a quartz bulb nearly filled with a liquid (tetrachloride of carbon) which expands when heated, shattering the bulb and releasing the valve cap The chemical type (Saveall) contained in the strut is a small cylinder filled with a low-fusing chemical This chemical liquefies at the rated temperature and allows the strut to collapse, releasing the valve cap. The pendant is a special type which prevents clogging Sidewall sprinkler heads have flares attached to deflect the water from the wall about 15 ft. They should only be used in special places. Picker-trunk sprinkler heads have a smooth deflector attached to avoid collection of dust Chloride of calcium sprinkler head is a special design for this type of equipment.

These are furnished in various patterns—plain, ornamented, wax-coated, lead-coated, wax- and lead-coated, calcium chloride, lacquer- and wax-coated, upright, pendant, picker trunk, and sidewall. Most of the heads used are of the solder type (a metal strut and fusible solder).

Tests of sprinkler heads should be made at about 15-year intervals. Tests in the field can be made by cutting the solder if fragile or flaky, head should be removed. An extra supply of heads should be kept on hand at all times and kept at a temperature not exceeding 100°F. The operating parts of a sprinkler head should never be painted.

Table 5-65

Max temp. at ceiling, deg F	Type rated	Operating temperature, deg F		Color
		Nonsolder	Solder	
100	Ordinary	135 150	155 165	Bronze
150	Intermediate	175	212	White
225	Hard	250	286	Blue
300	Extra hard	325	360	Red
375	Extra hard	400	360	Green
475	Extra hard	500	300	Orange

Temperature ratings of a sprinkler head approximate the temperature at which the sprinkler head operates in a hot-water bath at a slowly rising temperature.

Where room temperatures exceed 100°F or in any space subject to the heat of the sun's rays, it is better practice to use the next highest test head for margin of safety.

Network or Piping. If reference is made to differentiate a system such as 1-2-3, 1-2-4, or 2-3-5, this refers to the number of sprinkler heads on a given size of pipe (branch lines).

1 head on a ¾-in. pipe or 2 heads on a 1-in. pipe
2 heads on a 1-in. pipe or 3 heads on a 1¼-in. pipe
3 heads on a 1¼-in. pipe or 5 heads on a 1½-in. pipe

The piping increases as it reaches the riser. The pipe sizes have been figured to allow sufficient water to reach the sprinkler head without undue loss by friction. The design of the network of pipes is governed by the floor construction, as all sprinkler heads should be in an upright position wherever possible. The deflector or spray shield should not be more than 10 in. or less than 4 in. from the ceiling or bottom of beams or joists (in fire-resistive construction, 14 in.) (see Fig. 5-7). The sprinkler head should be so placed that all parts of the ceiling are sprayed by the water.

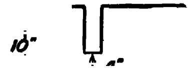

FIG. 5-7. Clearance between sprinkler and ceiling.

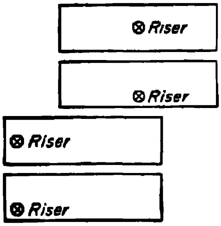

FIG. 5-8. Automatic sprinkler riser positions to accommodate construction types.

The spacing of sprinkler heads is generally staggered so that all intervening spaces are covered. End sprinkler head on alternate lines should not be more than 2 ft from wall; alternate line, 4 ft. Network designs follow four general patterns (see Fig. 5-8).

The center central feed is the most efficient network design as the others favor friction loss.

Table 5-66. Spacing and Area

Cons. of ceiling	Max. sq ft per sprinkler		Max. spacing bet. lines and bays, ft	
	Ord. haz.	Light haz.	Ord. haz.	Light haz.
Fire resistive	100	196	12	14
Mill	100	168	12	14
Semimill	90	144	10	14
Open joist	80	80	8 right angle to joist	8 parallel to joist

Risers. The size of the riser depends upon the number of sprinkler heads on any one floor or section fed by the riser (see Table 5-63). For 400 heads on an 8-in. pipe, it is preferable to use two or more 4- or 6-in. risers than one 8-in. riser where there are more than 250 heads on a floor.

EXAMPLE: Floor or section area is 50 by 100 ft = 5,000 sq ft, divided by 80 sq ft = 63 heads.

$$20 \text{ psi on head-gpm} = \tfrac{1}{2}P + 15 = 25$$
$$25 \times 63 = 1{,}575 \text{ gpm for } 20 \text{ min} = 31{,}500 \text{ gal}$$

All floors or sections should be equipped with scuppers, two to each 250 gal flowing. The water flow from the riser to the network of piping is controlled at each floor or section by OSandY valves (outside screw and yoke). These are sealed open with a dated tag (to prevent tampering). Check valves are placed in the piping to maintain pressures and control from yard main, tanks, etc. PIV (post-indicator valves) are installed to control water supplies to buildings and sections and to cut out valves for repairs. Certain types of systems have a single-control OSandY valve at the base of the riser for a single building or section. This necessitates shutting down the entire water supply to the network in case of fire. The control by floors or single sections shuts off only that area where the fire occurs and is confined, thus making it possible to save water damage and to restore service promptly.

Hydrostatic Test. The entire system when completely installed should be subjected to a hydrostatic test of 200 psi for 2 hr or 50 lb above the normal pressure when the normal pressure is in excess of 150 lb. The general practice is to test at 200 psi plus 50 psi for 2 hr. All tests should be made by the installing company with an inspector (from rating bureau) present, and a written statement should be furnished that the work has been completed and tested in accordance with the approved specifications and plans.

Details of Automatic Sprinkler System. The *base of the riser* where it connects to the main water supply should be carried to a point below the frost line and from this point to the floor of the building. It should be protected against freezing by suitable boxing or brick walls filled with solid earth.

CONTROL VALVES. The control valves should be OSandY, inside buildings, in pits, or aboveground. Underground valves, controlling main lines or sections, should be of the post-indicator type PIV.

CHECK VALVES. Check valves should be of the straightaway pattern, placed in a horizontal position. If double checks are used, they should be arranged so that inspection is made easy.

FITTINGS. Long-turn fittings are preferable. Extra-heavy fittings are required if normal pressure exceeds 175 lb.

HANGERS. For piping, see listing by Underwriters' Laboratories.

CORROSION. Nonferrous metal is used for sprinkler parts, but special coatings are needed when parts are exposed to corrosion, chemicals, or vapors, such as wax, lead, paper, or glass globes.

TEST PIPES. On all 4-in. or over risers, a 2-in test pipe is required which should discharge to the outside of the building. A 1-in. test pipe should be installed on wet systems near the top of the system or riser for use of inspector. A 1-in. test pipe should be installed on dry systems on the end of the most distant branch lines. A test gauge should be installed at the test pipe to note the drop of pressure, if any.

ALARMS. All sprinkler systems should be equipped with alarm devices to notify those responsible for its care and also the authorities, such as fire department or special services These alarms are attached to the system at or near the base of the riser to indicate the slightest flow of water. There are many types of alarms, including the vane or paddle type, with make-and-break electric circuits, vertical checks with electric attachment; retard-chamber types which prevent false alarms, water-flow gongs, and electrical water-flow gongs. All valves, checks, and alarms should be clearly marked.

SUPERVISORY SYSTEMS. Supervisory systems are those which are under supervision of a special service of American District Telegraph which responds when the alarm is given by sending a messenger to the plant and also to the fire department.

PROPRIETARY SYSTEMS. Proprietary systems are those within the plant to notify the employees and local fire defense.

OBSTRUCTION. A clear space of at least 24 in. should be maintained under all sprinklers.

Maintenance. Automatic sprinkler systems should always be maintained at the highest efficiency. A record of 21 years shows unsatisfactory conditions as given in Table 5-67.

Sprinkler Leakage. Sprinkler leakage is accidental leakage or discharge of water from the sprinkler system, which includes piping, heads, tanks, and other water supplies. Systems are rugged and are subjected to heavy pressure. Sprinkler heads will not break unless injured. Fifty per cent of the leakage cases are due to accidents to sprinkler heads.

Freezing. Freezing of sprinklers is generally caused by lack of heat in exposed places such as roofs and skylights and end pipes near windows; concealed spaces; windows left open; incorrect pitch of lines; and poor drainage.

Undesirable Conditions. Some undesirable conditions are change of occupancy, which may call for change of heat temperature rating, lack of heat, improper mechanical conditions with risers not properly supported, and strains due to settling of building or ground.

Table 5-67

	Cases
Water shut off	1,370
Unsprinklered portions	26,669
Defective water supplies	2,607
Freezing	5,049
Dry systems not maintained	2,564
Wrong temperature rating of sprinklers	2,260
Obstruction to distribution	11,001
Alarm valves and gongs out of order	9,197
Gate valves, repair needed	7,805
Gravity tanks, repair needed	4,450
Pressure tanks, repair needed	1,596
Fire pump, repair needed	184
Outside sprinklers with plugged heads	208

Table 5-68. Effects of Sprinklers

Effect	Fires, 1897 to 1939, inc.	
	No.	Per cent
Practically or entirely extinguished	44,256	70
Held in check	16,500	26.1
Total	60,756	96.1
Unsatisfactory	2,490	3.9

Water Supplies. Water supplies are treated under Public and Private Water Supplies (page 248) and consist of fire-department connections, city water, gravity tanks (from 5,000 to 100,000 gal), pressure tanks and fire pumps and underground sumps.

Outside Sprinklers. (See Exposure.) For pipe sizes and supplies see NBFU Standards on Automatic Sprinklers.

Table 5-69. Pipe Sizes for Six Stories (Counting Down)

	Orifice, in.	Riser size, in.	Branch lines, number of heads		
			⅜ in.	5/16 in.	¼ in.
Top line	⅜	1½	6	6	6
5	⅜	2	10	10	10
4	5/16	2½	20	20	20
3	5/16	3	36	36	36
2	¼	3½	55	55	55
1	¼	4	72	72	72

Not more than six heads should be used on a branch line. Cornices should be treated the same as windows. For ridgepoles ⅜-in. orifices, 6 ft apart, should be used.

Deluge System. The purpose of the deluge system is to wet down an entire area all at once by admitting water to open sprinklers. These systems can be operated manually or by thermostatic or rate-of-rise control. Deluge systems are used on extreme hazards, films, lacquer, spray coating, cotton nitrating, and airplane hangars. These systems have from 75 to 150 heads, and the usual requirements are doubled.[32]

Preaction Systems. Preaction systems are designed to eliminate the delay in the discharge of water which occurs in the operation of dry-pipe systems. An air pressure of 1 or 2 lb is maintained on the system. If this pressure is relieved, an alarm is given, and a deluge or hydraulic valve is opened. This also prevents the danger of premature or accidental operation of automatic sprinklers.

Automatic Rate-of-rise Systems. A heat-activated device is fastened to the ceiling and connected by a copper tubing containing air. The impulse caused by increase of air pressure operates an "automatic release." This device gives an alarm, closes fire doors, opens explosion vents, releases water to the sprinkler system, and stops fan motors, etc. The automatic "Supertex" system is a development of the preaction system. It operates on $1\frac{1}{2}$ lb of air.

Dry-pipe Systems. The piping system for dry systems is the same as for wet systems, the difference being that the dry system is filled with air under pressure which is controlled by a dry valve. When the dry valve operates, the system acts like a wet system.

Dry Valves. Dry valves are generally of two types—the differential type and the mechanical type.

The differential type of dry valve has two clappers of unequal size. The tripping point of one clapper to the other is determined by the ratio of area. The ordinary ratio of water to air is 6:1, *e.g.*, water pressure of 50 to 100 lb will take an air pressure of 15 lb. As a factor of safety, 30 to 40 lb of air may be carried.

In the mechanical type of dry valve the clappers may or may not be of the same area. The air clapper holds the water clapper shut by a series of levers and a stress screw. This type is obsolete but will be found in many plants.

Not more than 600 sprinklers should be on a single dry valve. Dry-pipe valves controlling more than 300 sprinklers should be equipped with quick-opening devices. Dry valves should be located in accessible places as near to the sprinkler system they supply as possible and protected from mechanical injury and freezing.

Air Pressure. The automatic sprinkler system should not lose more than 10 lb of air per week. It is permissible to carry 20 lb over the tripping ratio. The air compressor should be able to supply the normal air pressure times 25 per cent in 30 min. Relief valves should be set between compressor and controlling valve (on line) to 5 lb excess over maximum required. Water supply should enter below dry valve. A 2-in test pipe should be located between control valve and dry valve. A 1-in. test pipe should be provided also. All sprinklers should be in an upright position. All piping should be arranged to drain properly and be protected from freezing.

Records show that twice as many sprinklers open on dry systems as wet systems. The fire record shows the following for a 5-year period with a total of 2,130 fires: For 1,961 fires, sprinkler operation was satisfactory (92.1 per cent); for 169 fires, sprinkler operation was unsatisfactory (7.9 per cent). The setting up of the dry valve should be studied in order to understand its technique.

Inspection of Dry System. The main gate should be checked. It should be open and fastened. The air pressure and water pressure should be noted. The 2-in. pipe should be tested, and the air compressor should be checked. The dry valve should be tested for water above the valve (at air test pipe). If water is present, it should be drawn off slowly if quick-opening devices are attached.

Nonfreezing Solutions. Calcium chloride or sodium chloride should be used in cold-storage warehouses as well as glycol and glycerine.

Quick-opening Devices. On systems of over 300 heads, quick-opening devices should be employed.

The accelerator type of quick-opening device operates as follows: When sprinklers open and air pressure drops 1 or 2 lb, the diaphragm becomes unbalanced. This forces open a valve which permits the air to enter the intermediate chamber of the differential valve, and the valve trips.

The exhaustor type of quick-opening device operates as follows: The diaphragm causes a larger valve to open, and the air discharges by a 2-in. outlet to the outer air. The exhaustor takes a little longer time, but the water reaches the sprinklers in about the same time.

Alarms. The alarms on dry systems should be the same as those used on wet systems.

Cost of installation of automatic sprinkler systems can be given only after a complete analysis is made of the plant. For a rough estimate of approximate initial cost, however, the following items can be considered: entire floor-plan area, including basement, attics, and roof structures and all areas in concealed spaces, decks, and tables. This entire square-foot area divided by 80 will indicate the number of heads needed. The price per head area will range from \$20 to \$30, or an average of \$25.

Additional charges for dry-pipe systems may increase the cost 15 to 20 per cent. The city meter charge varies greatly as to whether the public water supply is municipal or private. The cost of gravity tanks, pressure tanks, outside sprinklers, and rate-of-rise systems has to be ascertained. This estimate in no wise fixes the price of the entire installation as this must be given by the installing company.

Special Types of Extinguishing Systems.[33] *Foam Systems.* It has been demonstrated that the extinguishment of fires in tanks of flammable liquids by smothering them through the use of foam on the surface of the liquid can be successfully accomplished. The use of this form of protection is particularly indicated where the usual means of extinguishment involving the use of water may be undesirable. Foam has the quality of flowing over and floating upon most flammable liquids, and it also adheres to most surfaces, thereby protecting them against fire by displacing the oxygen and cutting off the further formation of burnable gases. The foam consists of minute bubbles of carbonic acid gas and has a smothering action on the fire.

The variety of hazards which may be protected by foam extinguishment systems is great, and in applying the rules care should be taken to determine first under which class the hazard to be protected falls, so that the particular rules for that class may be applied. There are two types of foam:

Type 1—Suitable for use on flammable liquids not miscible with water, principally petroleum and its products

Type 2—Suitable for use on flammable liquids that are miscible with water, such as alcohols, esters, and acetones, and may be used on petroleum products

At present there is approved apparatus covering four general methods of applying foam:

1. Self-contained apparatus consisting of certain approved types of containers with devices to operate them automatically, the foam-producing materials being contained within the device.
2. Closed apparatus containing the chemicals necessary for producing foam but having provision for the admission of water to these materials with the necessary automatic or nonautomatic apparatus to produce proper operation.
3. Generators requiring a water supply and into which dry foam-producing chemicals are added, the generators being designed to mix the chemicals and water and produce a discharge of foam solution. There are two types of these generators. One type uses a single foam-producing dry chemical, and the other uses two dry chemicals constituting each of the two principal ingredients, the two solutions formed in the latter type of generator being kept separate until mixed in or beyond the generator.
4. Stored-solution systems. The two foam-producing liquids are stored separately and handled by pump pressure or gravity tanks to provide the necessary mixing and discharge to produce foam. One pound of combined chemicals and 1 gal of water are necessary to produce 1 cu ft of foam. Where two solutions are used, $\frac{1}{2}$ gal of each solution to make 1 gal combined having an expansion of eight at ordinary temperature, may be assumed to produce 1 cu ft of foam.

Water-spray Fog Systems.[34] The value of water applied in finely divided form by means of a special type of spray and so-called "fog nozzles" in extinguishing or controlling certain types of fires has been recognized. There are two methods of applying water spray:

1. By the use of hose lines (see *a* below)
2. By the use of fixed systems of piping (see *b* below)

Extinguishment of fire by the application of water in a finely divided form is probably accomplished by a combination of a smothering effect produced by steam, displacement of air, or a maintained film of water; by cooling; and in the case of some oils, by forming a smothering foam or emulsion over the surface of the burning liquid. Water thus used evaporates (absorbs heat) rapidly and so has a marked cooling effect.

a. For manual use, spray nozzles and play pipes may be used on outside and inside hose lines. When using spray nozzles on hose, a distance of at least 10 ft should be kept from electrically charged apparatus to protect operators.

b. Where fixed-pipe systems are installed, the construction standards for automatic sprinkler systems shall apply. All piping should be galvanized or of noncorrodable material.

Carbon Dioxide (CO_2) *Systems.*[35] The extinguishment of fires through the use of carbon dioxide, producing a smothering effect, can be successfully accomplished where there are flammable liquids in open and enclosed containers, rooms or enclosures containing flammable liquids, ovens, dryers, electrical and other special machinery and apparatus, and processes involving the use of flammable liquids, vapors, or dusts, fur storage, lumber dry kilns, coal bins, loose textile stocks, grain-handling machinery, and in other enclosures containing stocks through which gas may permeate and where protection by water or other means may be ineffective or undesirable; also in vaults, library stacks, organs, and other such places where fires may be extinguished by carbon dioxide gas with less loss than if water is used.

HIGH-PRESSURE SYSTEMS. Carbon dioxide is applied from cylinders in which it is stored at a pressure of about 850 psi at 70°F. It is required that the quantity of carbon dioxide in the cylinders be determined either by weighing or by other means at least once a year.

LOW-PRESSURE SYSTEMS. Carbon dioxide gas may be stored in tanks in liquid form in any capacity. Low-pressure tanks are kept refrigerated so that gas pressure may not exceed the designed working pressure of the vessel (usually around 325 psi). So far as safety to life is concerned, the gas, if not breathed in excessive amounts, is not dangerous, but in quantities it is suffocating.

Alarm Systems.[36] The primary intent of all alarm systems is to notify someone that a fire has broken out. This can be done by audible alarm or by direct connection to public fire-fighting forces. All alarm systems must work 24 hr a day. Therefore, care and maintenance must be maintained every hour of the day and night. Alarm systems have entered the broad field and are used as detecting devices on all types of machinery operated automatically to maintain water levels, steam pressures, atmospheric conditions, gate valves, and fire doors to ensure their immediate operation Some type of fire-alarm system should be installed in every plant because the immediate detection of a fire is the most important factor in its extinguishment.

Clock and Dial System. The watchman carries a clock which contains a paper dial. Keys are placed at predetermined locations throughout the plant (usually by the rating bureau), and the watchman on his hourly rounds "punches the clock" with the key, which makes a perforation in the dial. The dial is printed as an hour and minute graph, and the perforations complete the graph as to the watchman's rounds.

Key and Dial System. The paper dial is set up in a central clock in the office or any suitable location. The watchman carries the key and sends an impulse from the boxes or stations placed throughout the plant. The mechanism of the clock makes a perforation on the dial, which also acts as a graph.

Magneto and Dial System. The magneto and dial system is the same as the key and dial with one difference. The watchman rotates the magneto at each station to send the impulse to the clock.

Water-flow Type of System. The water-flow type of system is an electrical alarm activated by the slightest movement of water in pipes on flaps, vanes, or wheels, which close a circuit and thus send in an alarm.

Rate-of-rise System (air expansion by heat). The rate-of-rise system is operated by the expansion of air in small copper tubing run on ceilings strategically. This expanded air exerts an impulse on a selenoid cell, which in turn either opens or closes a circuit to give an alarm or activates a diaphragm for the same purpose. This system can be extended to operate fire doors, etc.

Thermostat System (metal expansion). The thermostat system is made up of a circuit in which is inserted a broken coil or ring. This coil when heated closes the gap and thereby closes the circuit to send in an alarm. There is also a system that uses the properties of two metals, bismuth and antimony, for making an electrical contact when heated (Ruchel rate of rise).

Supervisory or Central-station Protective Signaling System. The supervisory or central-station protective signaling system is an automatic electrically operated system, using detecting devices. It operates on the principle of rendering a service in addition to transmitting alarms. It checks the watchman on his rounds and maintains runner or armed-guard service which can be sent to the plant in case of trouble.

Proprietary Systems. Proprietary systems are operated on the same principle as that used in the electrical circuits but are confined to the plant and give only an audible local alarm.

Auxiliary Alarm Systems. Auxiliary alarm systems have circuit-closing switches throughout the plant tied into the municipal circuit and are of the shunt or series types. The shunt type is connected to the main circuit and is equipped with a tripping magnet which shunts the current until one of the circuits is closed. The series type has a separate source of current which operates an electric relay controlling the current to the tripping magnet, which is separate from the main circuit. When a switch is closed, the tripping magnet is energized and operates the municipal fire-alarm box. The shunt type is acceptable to most fire departments. The automatic McCullougher circuit utilizes a ground return for the current by inserting a code wheel, which makes and breaks a connection between the line and the ground.

Plant Fire Brigade. Regardless of the size of the plant or establishment, the personnel and workmen should be organized for the purpose of preventing and extinguishing fires. A plant fire brigade, to be a success, must be recognized by the management as an important permanent addition to the plant operation. The fire brigade should be organized under a constitution and by-laws and hold regular meetings for discussion and drill. Each officer should be provided with a badge denoting rank, and incentives should be given for membership and distinction and privileges awarded. The chief of the brigade should be selected for his ability to lead, analyze, and plan; he should be responsible and report to the highest executive in the plant.

The master mechanic makes a good fire chief if he has the ability. The chief must have full charge at a fire unless superseded by an officer in charge of the public fire departments. The assistant chief should have qualifications similar to those of the chief. Captains should be selected from heads of departments. The plant engineer and electrician should be members of the brigade. The watchman should be brought into contact with the brigade to become acquainted with the proper procedure in case of fire. The strength of a company should be 10 men, but the division of units may call for a smaller number. To be efficient a fire brigade should observe three essentials: rigid discipline, provision and maintenance of necessary apparatus, and adequate training under discipline with the apparatus. The plant fire brigade should invite the public fire department to visit and inspect the plant and should cooperate with it. The entire strength of the brigade, amount of equipment, and frequency of fire drills will depend upon the size of the plant and the nature of the hazards involved. Fire drills should be held at least twice a month, and the location of the drills changed so that the men can become familiar with all parts and conditions of the plant.

Drills should consist of making hose connections with hydrants and standpipes, laying hose, coupling and uncoupling, carrying hose up ladders, over roofs, etc., and in holding play pipe because of kickback. When plant is in proximity to natural water supply which is available for fire use, landings should be maintained for pumping engines to take suction. Roads should be kept clear, and if snow impedes access, hydrants should be marked with suitable signs. Crossings at railroads and tracks should be kept free. In sprinklered plants, officers should delegate men in case of fire to make sure the sprinkler valves are open, to see if sprinklers are operating, and that the fire pump is started. Use of hose streams should not endanger the supply of water to the sprinkler system. No one should close a sprinkler valve unless ordered to do so by the fire chief. After the fire all used sprinkler heads should be replaced, valves should be checked to see that none is closed, and the system restored as quickly as possible.

Watchman.[37] An alert, intelligent watchman should be employed. He should know the plant thoroughly, should know how to send in an alarm, and how to shut off water in case of sprinkler leakage. His rounds should be made on time, but time should be allowed to make a thorough inspection of premises as he proceeds.

Stock-flow Method. The stock-flow method is the proper approach to the study of fire prevention in the industrial plant. A diagram of the plant operations and processes, including raw stock (storage), stock flow, machinery and mechanical conversions, finishing processes and finished stock (handling and storage) should be made and studied to ascertain

1. Concentration of critical material
2. Concentration of values
3. Susceptibility of materials (raw, in process, and finished) to water and smoke damage and flaming and the degree of burnability
4. Operational processes of a single machine or group concentration that may develop a fire hazard
5. The area that may be affected by a single fire, including horizontal and vertical area
6. The rapidity of spread of fire due to the flaming nature of the material
7. The possible damage to equipment and stock and the time replacement of such equipment and stock
8. The value of fire barriers or cutoffs to confine a fire to a given area, both horizontal and vertical
9. The standard treatment for mitigating or removing possibilities of fire in any process or machine operation
10. The exposure of hazardous operations to high values, critical material, and equipment

Fire Attack. A map should be made of the entire plant and surrounding exposures. A good-sized map can be made on a scale of 50 ft to 1 in. The map or plan should be divided into squares of 50 sq ft and again into squares of 200 sq ft. The stock flow, concentration of values, hazards, susceptibility of stock and equipment to water and smoke, gases and acids, and discoloration and contamination should be studied as well as floor openings and cutoffs. It should be remembered that fire and smoke go upward, and water goes downward. The necessary first-aid equipment should be provided; precautions to be taken and the necessary action to be taken should be posted. Workmen and employees should be instructed to notify plant fire chief at once, no matter how small the fire. The public fire department should be called at once.

Self-inspection Blank. The following blank should be used as a constant check on the plant, and this report should be perused by the plant manager every month. The inspection should be made by a competent man who is familiar with the details of the blank and who can interpret the defects in an understandable manner.

SUGGESTED FORM
OF
SELF-INSPECTION BLANK
FOR
INDUSTRIAL PLANTS

Prepared by the National Board of Fire Underwriters, 85 John Street, New York City 7, as a Daily Report for Plant Owners to Have Printed in Quantity for Their Individual Use.

NOTE: The following general rules should be observed by the foreman or other trusted employee selected to make these reports.

Give attention to *every* question on this blank. If the question does not apply to this plant, draw a line through the space left for the answer. In this way you can be sure that you have not overlooked anything.

Some questions are to be answered by Yes or No, but others must be answered more fully.

When you find some defect, explain its cause and the steps that you are taking to correct it in the space headed Corrections and Remarks, on the last page of the blank. In doing this, *give the number of the question.*

If it ever becomes necessary to shut off the water from the sprinkler system, notify the office of the plant *at once* (even in advance, if possible).

This is important to enable us to notify the insurance companies or association.

If there is anything that you do not understand, consult with the manager at once.

Turn in the report blanks promptly to the manager of this plant.

Remember that these inspections are very important. The safety of this plant from fire is largely up to *you.*

GENERAL CONDITIONS

Cleanliness and Order

1. Was your inspection complete, covering all parts of the premises, including looking under benches, into closets, behind radiators, elevator pits, etc?........

2. Where did you find dirt or litter?

3. Were all clothes lockers clean and in good repair?..............................

4. Where did you find oily waste or any other greasy material outside of approved waste cans?..............................

5. Are any waste cans not emptied daily after closing hours?..............................

6. Is there any failure thoroughly to clean up shipping and packing rooms at closing time?..............................

7. Is there more packing material brought in than is needed for one day's use?..............................

8. What rooms or departments were not as clean as they should be?..............

9. Was yard free from combustible material?..............................

10. Was basement clean?....................

Maintenance

11. Is there any part of the plant which the watchman fails to visit?............

12. Are any watch-clock records unsatisfactory?..............................

13. Where were machinery, belts, or shafting in bad condition?........................

14. Where were bearings dirty or poorly oiled?..............................

15. Where did you find broken window panes, plastering, partitions, flooring, or other defects?..............................

16. Were any aisles obstructed in stockroom?..............................

17. Where do piles of stock or other obstructions interfere with entering any part of building?..............................

HAZARDS

Heating

18. Where was woodwork or other combustible material too near to smokestacks, flues, furnaces, boilers, steampipes, etc.? ....

....

19. Where was anything placed to dry on boiler or steampipes? ....

20. Was fuel supply safely arranged?

21. Can heated irons be set upon combustible material? ....

Lighting and Electrical Equipment

22. Where were there any open flame lights near combustible material? ....

23. Where were there any broken fixtures or loosened wires? ....

24. Where were ordinary electric cords looped over nails or found in contact with anything? ....

25. Where were any fuses replaced by wire or otherwise defective? ....

26. Where did any panel boards or switch and fuse cabinets need cleaning?

27. Where did any motors need cleaning, outside or inside? ....

28. Where was there insufficient oil in any motor bearings? ....

29. In what manner was there any violation of our rules for storing or handling oils, gasoline, or other inflammable liquids? ....

30. Did you find dry room disorderly or with dirty steampipes? ....

31. Where were any fan bearings inaccessible or poorly oiled?

32. Where were any screens or dampers in air ducts out of order?

33. Where did you find any other than safety matches? ....

....

34. Where did you find any violation of our rules on smoking? ....

PROTECTION

Fire Doors, Traps, Shutters, and Escapes

35. Where were fire doors wedged open, obstructed, or out of order? ....

....

36. Where were the automatic attachments of fire doors out of order? ....

37. Where were traps or doors on openings through floors out of order? ....

38. Are they ever left open when not in use? ....

39. Are fire doors or shutters ever left open at night or on Sundays or holidays? ....

40. Where were fire escapes obstructed, broken, or out of order? ....

....

Fire Apparatus

(Note: Each hose house, watchman station, room, or department should be known by a separate number. If you find *any* defect in any items, place the station number opposite the item. This will aid in keeping records. At each of these points there should be a card stating the quantity of the various kinds of apparatus which are required at that point.)

Inside

41. At what points are there *less* than the required number of

- *a. Full* water casks ....
- *b. Full* fire pails ....
- *c. Full* sand pails ....
- *d.* Feet of hose ....
- *e.* Nozzles ....
- *f.* Spanners ....
- *g.* Chemical extinguishers ....

(*Should be recharged at least once a year*)

- *h.* Ladders ....

Outside

42. Where were post-indicator valves poorly oiled? ....

43. Where did indicator valves fail to show open? ....

44. Which yard hydrants are hard to operate? ....

(*Do not open or test hydrants in the wintertime*)

....

45. Which ones do not drain properly? ....

....

46. Which ones have you found obstructed with snow or ice? ....

47. Which ones are otherwise obstructed or hard to reach? ....

48. Where have you found that hose, nozzles, or spanners were missing or not ready for use? ....

....

Fire Pumps

(Note: Pumps must be started at least once a week, discharging water through relief valve, in order to make certain that they are in working order. They must be given a *thorough test* with rated number of hose streams every spring and fall.)

49 Give make, style, and size of all fire pumps in the plant

	Make	*Style and size*
A		
B		
C		

50 Give results of any tests made since last report

	A	B	C
a Time required to obtain water?			
b What pressure was maintained			
c Was action smooth?			
d Is there an automatic regulator?			
e Was priming tank full?			
f Was priming gate valve closed?			
g Were steam drips open?			
h Did steam trap operate			
i Was there enough oil			
j Was suction clear of leaves or other obstruction			
k Was pump reservoir full			
l Were valves in steam connection from boilers to pump wide open?			
m How many rpm at 100 lb water pressure with all outlets closed			

For electric pumps add	A	B	C
o Has current been interrupted since last report			
p What is condition of contact points on switches circuit breakers controllers etc			

Gravity Supply from Tanks and Reservoirs

51 Give location and capacity of each tank or reservoir

	Location	*Capacity gal*
A		
B		
C		

	A	B	C
52 What was water level			
53 Was water frozen?			
54 Was ladder out of order?			
55 Were hoops and support defective?			
56 Was telltale out of order?			

For pressure tanks add	A	B	C
57 What was air pressure and was water level correct?			
58 What was tank room temperature?			

City Water Connection

59 Was valve (on our premises) in connection to city water main wide open?

60 What was pressure on gauge? lb

61 Has there been any interruption of city supply since last report?

62 Give results of any tests since last report

a Number of hose streams
b Diameter of nozzles in
c Pressure at nozzles lb

Sprinkler Supply Valves

(Note All gate valves must be secured open with leather straps or other approved method fastened with sealed or riveted padlocks the keys being held by responsible persons In inspection each valve must be given one full turn to make sure that it is wide open and in working order Drip valves must be strapped shut in similar manner Valves under approved supervisory system however need not be secured)

Mark every sprinkler supply valve plainly with a number This will aid in keeping record

63 Give the number of any valves found closed

64 Give the number of any valves found not strapped

65 Give the number of any drip valves found open

66 Give the number of any valves to which access was obstructed

Dry Valves

(Note Dry valves should be tested for water column and condition of spring at least once each 3 months they should be tripped at least once a year)

	Location		*Location*
A		C	
B		D	

	A	B	C	D
67. What was room temperature?				
68. What was air pressure?				
69. What was water pressure?				
70. Was alarm out of order?				

Alarm Valves

Location	*Location*
A	C
B	D

	A	B	C	D
71. How was valve tested?				
72. Did any bells fails to ring?				
73. Was valve left in order?				

Automatic Sprinklers

74. Where is stock piled within 24 in. of sprinkler heads?

75. Have any sprinklers operated since last inspection?

76. Where are any sprinklers coated or corroded?

77. Where are any sprinklers obstructed by partitions, joists, etc?

78. Where are any sprinklers bent?

79. Where are any sprinklers leaking?

80. Where are any sprinklers exposed to freezing?

81. Where are any sprinklers missing?

82. Where are any sprinklers disconnected?

83. How many sprinkler heads are kept in reserve?

CORRECTIONS AND REMARKS

(NOTE: Wherever defects are found, steps must be taken *at once* to have them corrected. If the corrections have been made when this report is turned in, *this fact* must be stated below. If the correction has not yet been made, an *explanation* must be given below. Always refer to the question by number. *For example*, "*15—The broken windows in the shipping room were caused by boys throwing stones. I have reported the matter to the police and have had the broken panes replaced.*")
I have made a careful inspection and to the best of my knowledge and belief the foregoing statement are all correct.

(SIGNED)

Date

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SECTION 6

FUNDAMENTALS OF GOOD PLANT MAINTENANCE

BY HENRY W. SHOCKLEY

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FUNDAMENTALS OF GOOD PLANT MAINTENANCE

BY HENRY W. SHOCKLEY

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Maintenance of a modern industrial plant, precedented on low cost and continuous operation, cannot tolerate a status where emergencies and excessive repairs are a regular condition. Instead, it must function according to definite rules of orderly procedure and be subjected continuously to scientific study and change in order to meet continuous changing conditions in industry. Simultaneously, as the methods of manufacturing the product are being continuously investigated for improvement of process, manufacture, etc., to meet the ever-increasing effect of competition, so plant maintenance must be continuously investigated for better materials, improved designs, better equipment, tools, facilities, etc.

Over the past quarter of a century, there have been almost complete changes in the design of industrial plants, the handling and transportation of product, the use of automotive equipment, the divisions of labor, etc., owing in no small part to the effects of mass production in which large volumes and low costs are predominant. In the future these will be further emphasized to an even greater extent owing to the necessity of expanding the fields of industry to meet the needs of greater employment, increased wages, and higher standards of living.

In addition, another change has been added which may have an even greater effect on maintenance. This condition is one in which the individual operator is becoming more interested in the automatic operation of manufacturing equipment, where physical effort is less strenuous, and where the machine performs the major part of the work. This change will require equipment that

1. Is more automatic in operation.
2. Is more complicated in design.
3. Requires every safety feature.
4. Introduces the use of all measuring devices; both mechanical and electrical.

These changes will require closer attention by specially trained mechanics and engineering supervision. They will also necessitate the development of training programs covering the requirements for the special skills needed. The application and maintenance costs should be the basis for a continuous program of advanced development in equipment design and better materials of construction.

This enlargement in scope of a maintenance department indicates that the maintenance engineer can no longer be satisfied in maintaining a plant but must have the additional capacity and knowledge to use those factors which will prevent repairs or interference with operations and the best engineering methods available to improve the conditions of operation, equipment, or plant. This does not mean that the maintenance department will assume the responsibilities of those services now segregated under experiment and development. These, while more essential than ever to a plant today, are separate because of the type of problem and time required for investigation which makes it feasible for them to be segregated unto themselves. However, sufficient engineering knowledge and skill should be available in a maintenance department to recognize and understand the principles of engineering in their application to plant improvement. This requires the creating and availability of basic records and control to point out poor conditions and assist in the study of operating and maintenance costs, as well as the experience in and knowledge of the use of actual skills, tools, facilities, and equipment properly to maintain a plant.

To meet these conditions successfully it will be necessary that the maintenance organization and methods employed be equivalent in both quantity and quality to the size of plant and type of equipment. Otherwise, results will be unsatisfactory and maintenance costs will be excessive. It should be understood that a low maintenance cost may be the difference between an operating profit or loss, particularly since maintenance costs have increased to such a great extent during the past few years, with a greater proportion in labor costs than in material.

The essential need for any plant for best maintenance control is a plan to meet all these conditions. The application and use of such a plan is covered in this treatise.

The plan revolves around the following six major controls, *viz:*

1. Organization—supervisory and wage roll
2. Work and labor control
3. Training
4. Preventive maintenance
5. Corrective maintenance
6. Cost controls

It will be quite obvious as these various controls and subheadings under each are explained that this is a far-reaching plan; that results will be obtained based on effort expended; and that it may be necessary to change methods, viewpoints, and procedures to gain best results. It is believed that the plan is founded on a sound basis of facts; that there are sufficient checks and balances to offset variations in plant conditions; and that it will not only establish a standard of maintenance at all times but will furnish all necessary information to determine where improvements or changes should be made. The plan is described according to the six major controls.

ORGANIZATION (SUPERVISORY)

The importance of the proper organization structure to meet the conditions outlined in this plan cannot be overemphasized. Not only must there be supervision to carry out the duties and responsibilities for all services involving the proper maintenance of a plant at all levels of production and with a minimum of down time of operating equipment, but also the supervisory personnel itself must measure up to corresponding conditions of selectivity. Too often the selection has been based on general and not specific requirements or has been to meet the personnel available in the plant.

In many instances also the actual duties and responsibilities of a maintenance organization are not definitely defined or understood with the result that supervision is not applying itself to necessary requirements. This results in overlapping of duties, failure to understand conditions, and consequent poor results. Many times the knowledge of their responsibilities and a training program covering them would assist materially in improving present personnel, but without both knowledge and training only mediocre results can be expected.

Typical functional organization charts for both large and small industrial plants are shown in Figs. 6-1 and 6-2. In addition, Fig. 6-3 shows the responsibilities of the various types of supervision including clerical assistance. In establishing general standards of selection covering the various types of supervision and clerks, the recommendations would be as follows:

1. All engineers:
   a. Graduate engineers or equivalent
   b. From 5 to 10 years of plant experience
   c. Adapted to the responsibilities as shown in Fig 6-3
2. Foremen and clerks:
   a. Foremen and clerks to have at least a high-school education or equivalent
   b. Foremen to have learned a trade or its equivalent
   c. Foremen to have had from 5 to 10 years of practical experience in their trade or the plant

The organization charts (Figs. 6-1 and 6-2) indicate that maintenance of a plant should be considered under three headings, each of which determines the kind of organization required.

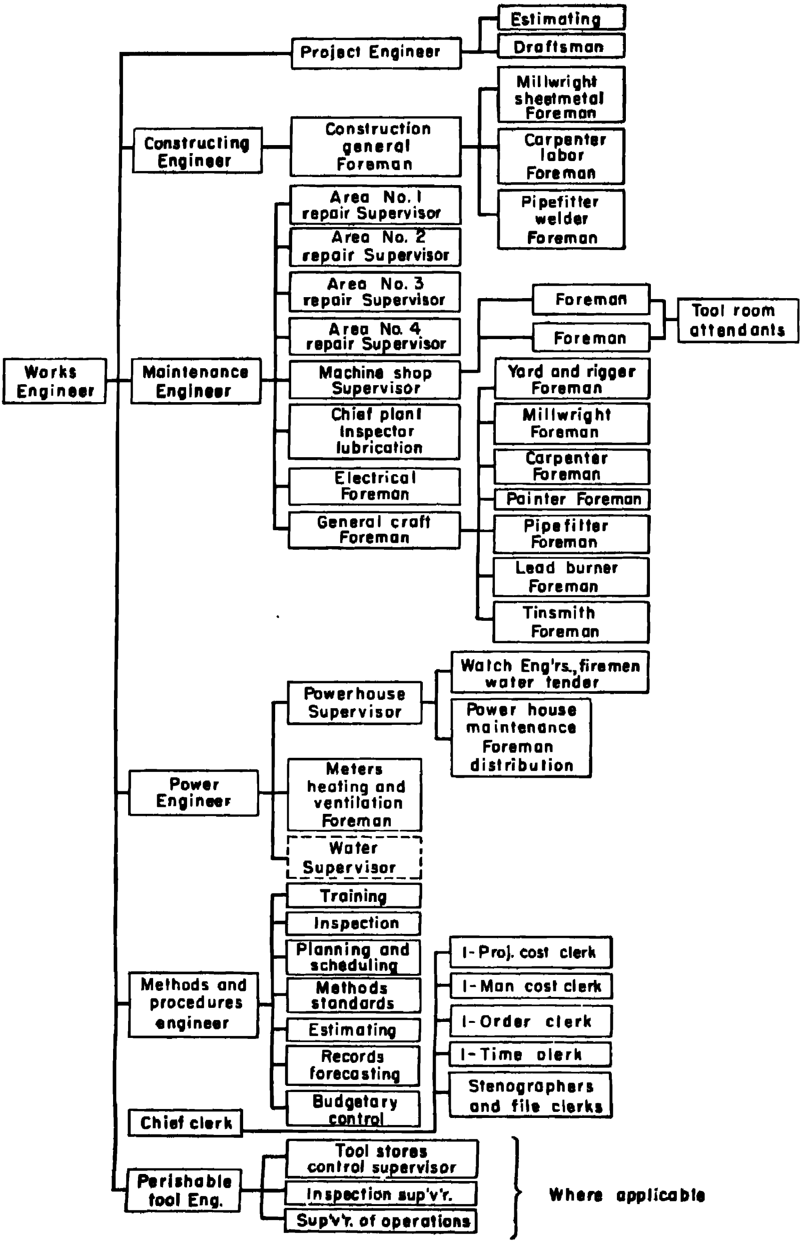


Fig. 6-1. Typical functional organization chart, power and mechanical department (department personnel above 200).

1. Shop and craft work, or common facilities
2. Work special to the particular area
3. Minor construction, which may affect both 1 and 2

In all plants the amount and kind of work done over a period of time indicates that which is common to an area or department and that which represents the use of common facilities both as to personnel and equipment requirements. It is therefore very necessary that an analysis of actual work requirements be known before the proper type of organization can be established and the foreman-mechanic ratio be determined correctly.

Although proof is not too definite, experience has indicated that the foreman-mechanic ratio can be 1:15 for outside crafts and up to 1:25 for shops or more centrally controlled groups. The more definite the responsibilities and duties of supervision are, the better this ratio can be measured. Experience also indicates that emergencies and confusion are evident where responsibilities and duties are not known.

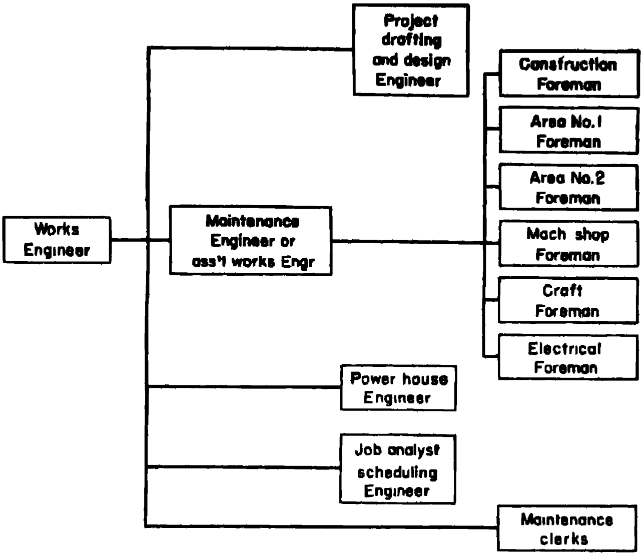


Fig. 6-2. Typical functional organization chart, power and mechanical department (department personnel above 150).

ORGANIZATION (WAGE ROLL)

Just as the administrative and supervisory functions of a works engineering organization must be definite and understood, so also must the work and crew requirements of all crafts and groups be definitely established as to skills, experience, tool requirements, etc., in order that they may do their work safely and effectively. This is also necessary for job-evaluation purposes or for determining wage scales to meet job requirements.

The general requirements common to all maintenance crafts and groups are shown in Fig. 6-4, and the specific job requirements for one craft, millwrighting, are shown in Fig. 6-5.

In order to meet plant requirements, it is necessary that the general requirements be understood and carried out 100 per cent by each craft or group according to its class and that 75 per cent of the job requirements be met in order to meet the rate of the particular class.

In addition, the requirements represent a standard for measuring personnel and in assisting them to do a better job.

WORKS ENGINEERING ORGANIZATION

Responsibilities	Works engr	Const engr	Maint engr	Project engr	Planning and scheduling engr	Const gen for	Maint gen for	Area engr	Craft foreman	Area foreman	Maint clerks
1 Organization	x	x	x	x	x	x	x	x			
2 Repair order control and time distribution						x	x		x	x	x
3 Control of maintenance and construction labor requirements						x	x		x	x	x
4 Job analysis						x	x		x	x	
5 Planning and scheduling maint and minor const work						x	x		x	x	x
6 Maintenance cost—direct, indirect, general	x		x				x	x	x	x	x
7 Equipment cost card procedure	x				x			x			x
8 Maintenance inspection		x				x			x	x	x
9 Job standardization	x				x	x	x	x			x
10 Budgetary control								x	x	x	x
11 Work and crew classification	x		x		x	x	x				
12 Shops layout, tool and equipment requirements	x	x	x		x						
13 Centralized prefabrication and assembly					x	x	x	x			
14 Train of repair order control and scheduling people					x						
15 Train of job analyst and scheduling engr	x				x						
16 Inventory control of materials			x								x
17 Job training supvy	x	x	x		x			x			

18 Job training (wage roll)					x		x		x	x	
19 Forecast major repair			x					x		x	
20 Safety	x	x	x			x	x		x	x	
21 Labor policies	x		x				x		x	x	
22 Emergency job investigation					x			x	x	x	
23 Reports	x	x	x		x						x
24 Quality							x		x	x	
25 Tool and gauges											
26 Costs	x	x	x					x			
27 Controlling waste						x	x		x	x	
28 Transportation and handling						x					
29 Plant rearrangement changes	x			x							
30 Drafting and design		x		x							
31 Estimating											
32 Project preparation				x							
33 Bills of material				x				x			
34 Material procurement and dispatching		x		x					x	x	x
35 Project cost control		x		x							x
36 Project scheduling		x									x
37 Powerhouse operation	x										

Fig 6-3

PROPOSED CLASSIFICATION OF CRAFTS—GENERAL REQUIREMENTS

Although not a requirement, it is very desirable that class 1 and 2 craftsmen be capable of doing an acceptable job in a lower classification in some other closely related craft. Class 3 men of less experience will probably not have sufficient skill in other crafts, although such ability will make them more desirable employees.

To advance to a higher classification, a man must have worked sufficiently long under proper supervision at the advanced classification to prove his capability.

Craftsmen of all classes must have sufficient knowledge to use information given in standard reference and handbooks of their trade. Craftsmen of all classes should cooperate with the foreman in the education of apprentices.

Throughout the various classifications, Supervision will be referred to as Close, Average, and Minimum. The definitions of each are as follows:

Close Supervision	Average Supervision	Minimum Supervision
Close supervision means that the job must not only be planned and scheduled properly, but it must be analyzed in more detail relative to tools and materials required; described fully; and the person executing the job be contacted frequently and given detailed instructions.	*Average supervision* means that the job must be planned and scheduled properly to eliminate lost time in its execution, and contacting between jobs, but that the patrol of the job is infrequent and fewer instructions are required.	*Minimum supervision* means after the job has been properly planned and scheduled that a minimum of contact and instructions are required.

The following general requirements must be met by workmen of all crafts:

Class 1 Craftsmen	Class 2 Craftsmen	Class 3 Craftsmen
1. Class 1 refers to the highest trained man in the craft who shall have had the proper training and experience in his craft according to plant requirements. The time required to meet such requirements will be specified by the plant.	1. Class 2 refers to men who are so trained in their craft that they are able to do as good a job as a class 1 man in certain types of work but will require more supervision and instruction.	1. Class 3 refers to men who have just completed their trade or those in the probationary period, who lack experience, skill, and seasoning. They require closer supervision and cannot be expected to accept responsibility.
2. He must be able to work from any print or clear sketch and be able to take his own field measurements and make a readable sketch or diagram when necessary.	2. He must be able to work from most prints but will require some aid on the difficult ones. He should be able to take his own field measurements when necessary and to make a readable sketch or diagram.	2. He will not be expected to work from drawings, take field measurements, or make sketches unaided, although ability to do so will make him a more valuable employee.
3. He must be able to work effectively from verbal instructions, drawing on knowledge accumulated during his broad experience for determining proper procedures for economical execution of the work.	3. He must be able to work effectively from verbal instructions; however, more detailed instructions will be necessary.	3. He should be expected to work from verbal orders only if given explicit instructions covering all details.
4. He must be reliable, resourceful, conscientious, and steadfast; be able to gain and hold the confidence of his fellow workers; and be capable of directing the work of other men in number according to plant policy. However, the number of men directed should be definitely stated.	4. Same as class 1, but he should be capable of directing the work of mechanics of lower classification or helpers according to policy of the plant.	4. He will not be expected to direct the work of others in any type of work under his classification.
5. He must know and practice all plant safety and housekeeping rules.	5. He must know and practice all plant safety and housekeeping rules.	5. He must know and practice all plant safety and housekeeping rules.

Fig. 6-4.

WORK AND CREW CLASSIFICATION
MILLWRIGHTS

Class 1 Millwrights	Class 2 Millwrights	Class 3 Millwrights
1. To be able to read and interpret mechanical drawings of standard machines or machinery on the plant and make intelligent sketches of the machine parts.	1. To be able to read and interpret mechanical drawings of standard machines or machinery on the plant and make simple sketches or layouts of mechanical parts.	1. To be able to read, follow, and perform work from simple drawings.
2. To check working load on various ropes, chains, and cables with the aid of handbooks. To have a working knowledge of materials which will stand up under the chemical conditions on the plant.	2. To be able to check strength of materials with the aid of a handbook, under close supervision, and to have general knowledge of materials that are standard for use on various machines.	2. Same as class 1 and 2, but under close supervision of foreman or senior craftsman.
3. To do all common methods of rigging or handling of heavy machines and materials in a safe and economical way.	3. To do all common methods of rigging and handling of heavy machines and materials in a safe and economical way under average supervision.	3. To do elementary rigging where no unusual problems are involved.
4. To be able to dismantle and assemble any standard machines and powerhouse equipment except turbine generators requiring outside supervision.	4. To take apart, put together, install, or replace any parts on common machines.	4. To dismantle or assemble any mechanical equipment either under close supervision or assisting a mechanic in a higher classification.
5. To recognize by feel, sight, or hearing, mechanical faults in machines in common use on the plant and repair same. This to be acquired by long and varied experience in the trade and on the plant.	5. To recognize by feel, sight, or hearing mechanical faults in machines in common use on the plant and make general repairs.	5. To have the mechanical ability to recognize common faults on common machines and make simple repairs.
6. To be able to run equipment in his own shop and simple machine-shop equipment, so as to be able to get out simple repair parts without help when necessary.	6. To be able to run drill presses, hacksaws, bolt machines, and simple machines which are in his craft shop.	6. To be able to run equipment in his own shop under supervision.
7. Ability to use the tools of his craft such as calipers, feelers, micrometers, etc., in order to secure tolerances of 0.001 in. on equipment where required, in order to give accurate dimensions and check finished machine work before installation. Also, to be familiar with all tolerances of running, shrink or press fits on shafts, bushings, keys.	7. Ability to use the tools of his craft, such as calipers, feelers, micrometers, etc., necessary to complete repairs on machines in order to secure tolerances of 0.005 in. on equipment where required.	7. To be able to use the tools of his craft, such as calipers, feelers, micrometers, etc., under average supervision.
8. To be able to use a complete set of millwright tools.	8. To be able to use the tools that will allow him to perform the above duties with minimum amount of trouble and expedite the job assigned to him with average supervision.	8. To be able to use such tools that will enable him to step into the next higher classification when the opportunity presents itself.
9. Be able to ascertain difference between materials and their application as related to plant operation.	9. Be able to ascertain difference between such common materials as cast iron, cast steel, steel, tool steel, ordinary babbitt, high-speed babbitt, and their applications. Have the ability to reline bearings.	9. Should be able to ascertain difference between common materials of his craft, such as cast iron, steel, babbitt, etc.

FIG. 6-5.

The training of mechanics relative to both their general and job requirements is essential for a proper understanding of what their responsibilities should be in relation to their jobs, particularly as a majority of mechanics in many plants have not had the benefits of an apprenticeship training program.

It is also believed that a definite scheduled program for training both supervisory and wage-roll personnel will have added advantages in directing their thinking and giving such personnel a status it has not previously had.

Of equal importance is the availability of locker space, showers, washrooms, etc., for the employees, close to their work stations in order to reduce the possibility of colds due to overheating, lack of clothing, etc. It must be understood that personnel is the most important factor in the proper functioning of an industrial plant.

WORK AND LABOR CONTROL

There are a number of maintenance services under work and labor control, each of which affects the maintenance of a plant to a large degree, both in work requirements and in having a minimum of personnel available to meet them. Each has a definite relationship to the other.

Repair-order Control. The first or key service is that of repair-order control, which is the means whereby every type of maintenance job done, whether minor or major in scope, is definitely covered by a standard written form which shows that the work is needed, is adequately described, is properly approved, is issued by proper authority, and provides a record of all this information.

The purpose of a repair order is to enable a maintenance foreman to analyze and schedule his work; to provide a means of scrutinizing jobs as to cost, necessity, size, etc., both before and after they are done; to assist the operating supervision in determining the necessity of the job; and to give maintenance supervision the means to estimate and schedule the job according to best plant requirements.

Reference to Fig. 6-3 indicates that repair-order control and time distribution are direct responsibilities of both general area and craft foremen, as this type of supervision is in direct contact with the work itself and the method of doing it.

It is also very important that operating supervision understand the work requirements, particularly their effect on operating schedules, facilities, etc. Because of their familiarity with process conditions, they are also in an excellent position to know the necessity for the expenditure.

Figure 6-6 shows the front of a normal order form, which describes the work to be performed, assignment, cost preparation and distribution, signatures, etc.

Figure 6-7 shows the back of this form, allowing space for additional instructions, materials required, special tools, number of men, safety precautions, etc.

Figure 6-8 shows a patrol-order form used for small nonrepetitive jobs up to 4 man-hours of labor and a maximum of $10 in material.

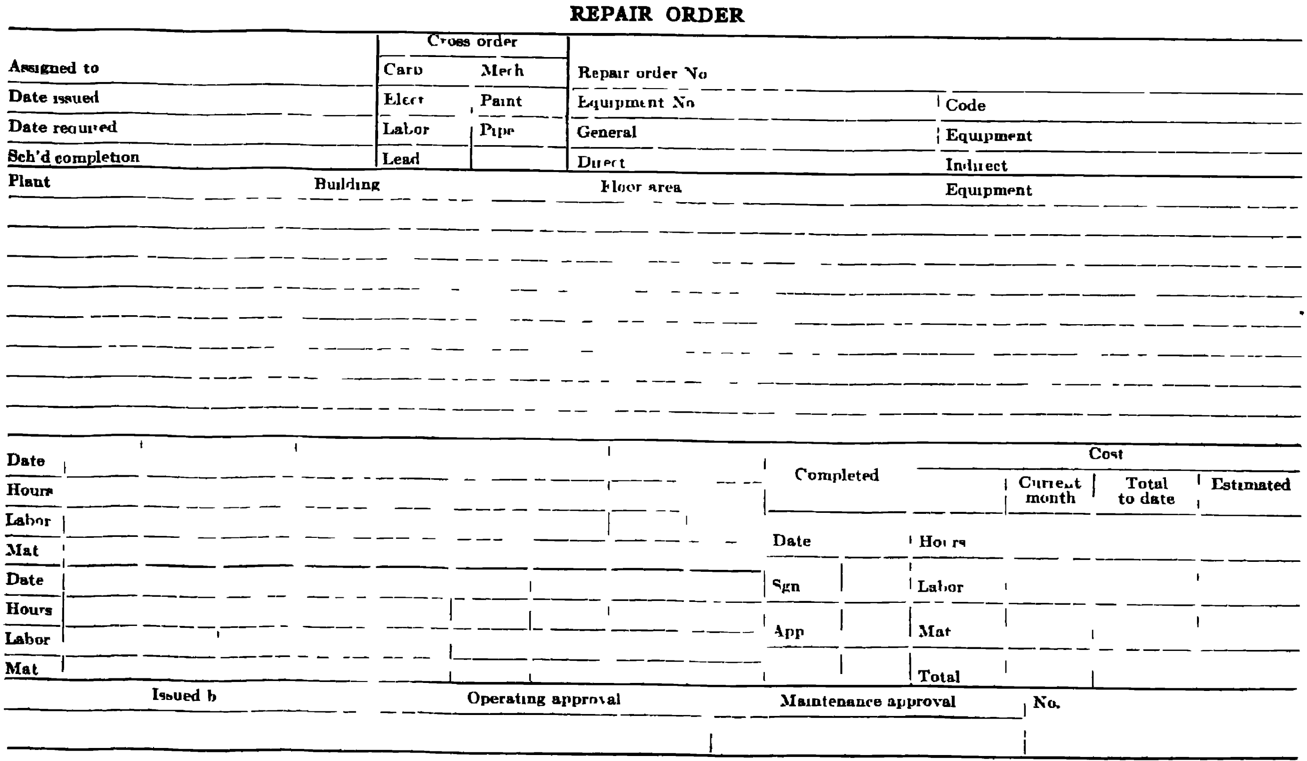
Repair-order control is a control of work requirements. It therefore becomes necessary that similar control be exercised over personnel requirements to do this work plus some measurement of the rate of receipt and completion of work

Control of Maintenance Labor Requirements. Control of maintenance labor requirements is a method used by maintenance supervision to compare estimated hours of work load with actual man-hours available.

The purpose of this control is to maintain minimum personnel requirements covering crafts, shops, areas, or minor construction groups according to optimum conditions of maintenance work and to meet production requirements.

Figure 6-9 shows a labor control or backlog chart covering all maintenance forces whether shop or craft, area, or minor construction groups. They are also segregated according to individual foremen and areas.

The required man-hours of work are estimated by the various foremen responsible according to repair or patrol orders received, and these are compared with man-hours available on a weekly basis. The result is known as "backlog of work." Periodic checks are made for accuracy of estimating and to assist each foreman in improving his estimating effectiveness by comparing actual hours of jobs completed with estimated hours before completion. This is termed the "correction factor."



REPAIR ORDER

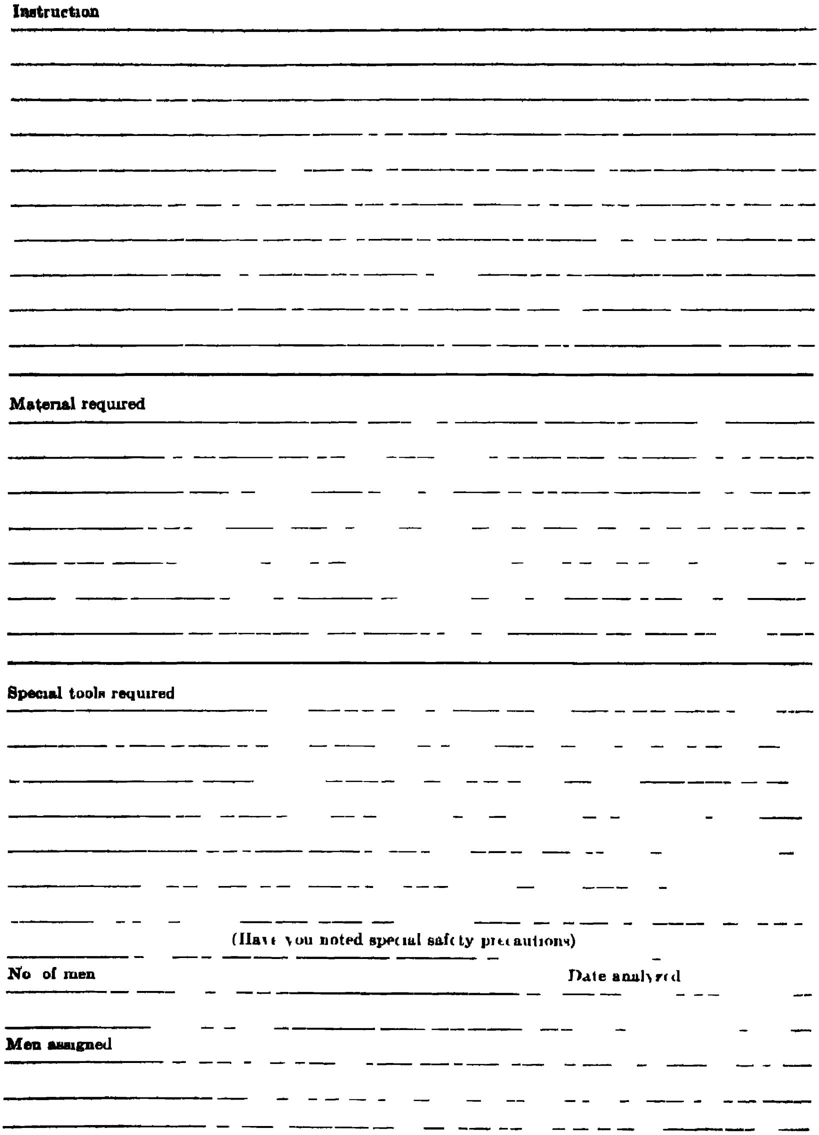
	Cross order			
Assigned to	Carp	Mech	Repair order No	
Date issued	Elect	Paint	Equipment No	Code
Date required	Labor	Pipe	General	Equipment
Sch'd completion	Lead		Direct	Indirect
Plant	Building		Floor area	Equipment

Date				
Hours				
Labor				
Mat				
Date				
Hours				
Labor				
Mat				

Completed			Cost		
			Current month	Total to date	Estimated
Date		Hours			
Sign		Labor			
App		Mat			
		Total			

Issued b	Operating approval	Maintenance approval	No.

Fig. 6-6. Normal repair-order form, front side, description and approvals.



Instruction

Material required

Special tools required

(Have you noted special safety precautions)

No of men

Date analyzed

Men assigned

FIG 6-7 Normal repair-order form, back side, job analysis

Experience has indicated that this procedure gives the following results:

1. It establishes minimum personnel requirements based on available work to be done in order to meet production schedules
2. It determines the rate of increase or decrease for any craft or group in volume of work over a period of time
3. It establishes a means for supervision to measure its own performance

Job Analysis and Scheduling. The third item under work and labor control is job analysis and scheduling of maintenance work. In order to utilize these controls to their best advantage, a method of assisting the foremen is first introduced. This is known as a "scheduling rack."

MAINTENANCE PATROL ORDER

Code number:	Date written:
Equipment number:	Date wanted:
Direct:	Indirect.
Description of job	
Signed	

FIG. 6-8.

MAINTENANCE DEPARTMENT
WORK ORDER PERFORMANCE

PERIOD

DEPTS.	A	B	C	D	E	F	G	H	I	J UPPER	J LOWER	MISC.	TOTAL BACK-LOG OF WORK R.O.	TOTAL HOURS SPENT ON BLANKET & PATROL ORDERS	TOTAL B.L. OF R.O.'s & P.O.'s	NUMBER OF CRAFTSMEN	AVAILABLE MAN DAYS
No OF ORDERS																	
ISSUED	25	8	10	10	6	47	9	8	6		1	27	153				
	168	47	54	58	34	241	34	102	23		17	123	901				
COMPLETED	18	3	7	8	8	35	6	4	4		2	19	152				
	201	46	59	45	25	228	31	88	28		13	118	888				
BACK LOG	85	24	29	26	22	97	6	63	25		6	53	446	* *			
HOURS OF BACK LOG																	
MACHINE	718	175	200	255		696	32	345	64		8	24	2605	5798	8403	55	15 1
TIN	250		18	4	10	168		34					468	358	826	8	12 9
BLACKSMITH	108	14	35	24	18	103		72	46			41	451	229	680	5	17.
CARPENTER	24	8	4		140	190	24	24	56		6	338	814	228	1042	6	21.7
PAINT	88	8		20		687		104			8	140	1047	16	1063	4	33.2
MASON	68					96		8				48	220		220	2	13 7
ELECTRICAL	348	38	30	47	168	187	16	215	312		10	378	1747	1314	3061	13	22.4
PIPE FITTING	262	68	68	10	64	420	24	422	92		4	166	1592	832	2424	19	15.9
PIPE COVERING	34	8	18		12	8	8	30	28			34	178		178	2	11.1
TOTAL	1890	319	443	360	404	2547	104	1250	398		36	1189	9122	6775	15897	112	12.7

BREAKDOWN: WORK ASSIGNED TO AREA CRAFTSMEN

	UPPER	LOWER	TOTAL
MACHINE	710	593	1303
TIN	194	77	271
BLACKSMITH	10	123	133
PIPE FITTING	291	464	755
* B&P ORDERS	1316	2473	3788
TOTAL	2520	3730	6250

* All Blanket and Patrol Order figures used in computing Back Log Hours are based on average monthly figures

** Does not include orders which have not been approved and returned to shops, or Purchase Orders.

OVERALL AVERAGE - MAN DAYS OF WORK ON HAND

$$\frac{\text{Total Hours of Work}}{\text{Total Number of Craftsmen} \times 8 \text{ hours}} = \frac{15{,}897}{112 \times 8} = 17.7$$

FIG. 6-9. Backlog chart, control of maintenance labor requirements.

Figure 6-10 shows a scheduling rack. It consists of six pigeonholes each 6 by 6 by 12 in., each marked as follows: jobs to be analyzed, jobs analyzed; jobs to be inspected; jobs waiting on material; jobs waiting on shutdown; jobs over 1 week old. This rack eliminates lost time in handling all repair orders, permits good housekeeping, and reduces clerical effort on the foreman's part in handling this work.

Jobs waiting on material and jobs over 1 week old will be discussed below. The others will follow under Maintenance Job Analysis, Scheduling Maintenance Work, and Inspection.

Jobs Waiting on Material. When a job is analyzed by the foreman, the material requirements are established. The type or kind of material is determined by the foreman as stores stock or special. If special, it must be purchased from outside vendors, which delays the execution of the work until the material is available in the plant. If the material is part of stores stock, it is assumed that the work can be done as soon as it can be scheduled.

Jobs over One Week Old. A certain backlog is necessary to plan and schedule maintenance work properly. Too great a backlog may result in emergencies, loss of production, down time, etc., while too small a backlog may result in lost time, poor scheduling, etc. Experience indicates that over 2 weeks' backlog in shops, exclusive of construction, results in the first condition given above, while less than 2 weeks in crafts results in the second condition. Segregation of jobs over 1 week old allows a means for closer checking of work and makes it available for scheduling.

Maintenance Job Analysis. Maintenance job analysis is the analysis of each maintenance job as to craft, material, tools, and methods and the preparation of a workable estimate of time and man-power requirements.

1 Jobs to be analyzed	2 Jobs analyzed	3 Jobs to be inspected
4 Jobs waiting on material	5 Jobs waiting on shutdown	6 Jobs over one week old

FIG. 6-10. Foreman's scheduling rack for repair orders.

The purpose of job analysis is to train maintenance supervision in the various factors that affect proper job execution, to have available for personnel doing the work the means of working effectively, and to have available for computing backlog the estimated man-hours for completion of all approved jobs.

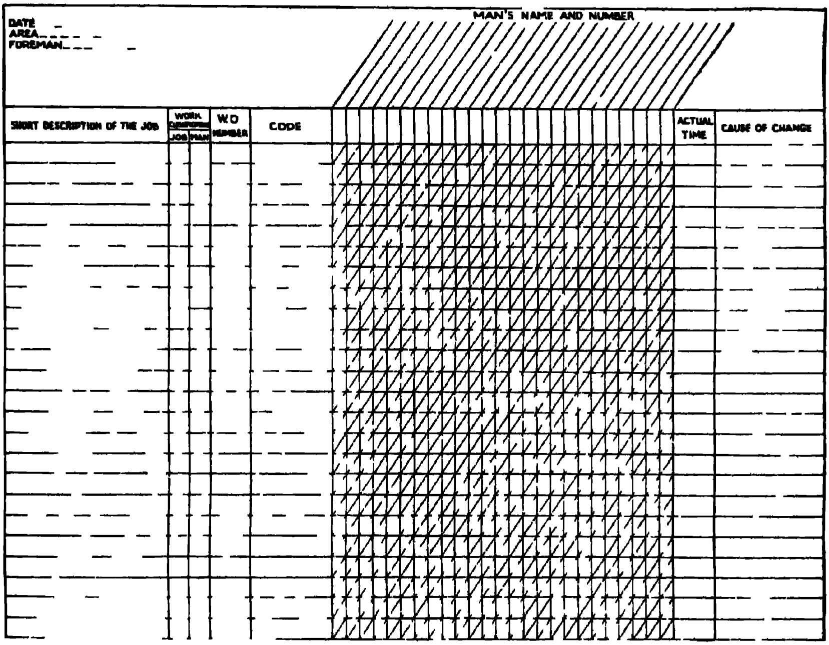
A maintenance foreman receiving a normal repair order (Fig. 6-6) visits the location of the proposed work. He analyzes the job as to man-hours required by his own craft or group and enters this on the face of the order, noting also the work to be done by the other crafts and groups, to each of which he issues a cross order. In addition, he notes on the back of normal repair order (Fig. 6-7) the following: any additional descriptions that will assist the person or persons doing the job; the general materials necessary; tools, either regular or special, that are not a part of a standard tool kit; and any safety precautions to be taken. If the material required to do his part of the job is not stores stock, it is his responsibility to see that it is ordered. Cross orders are treated in the same way as normal orders by the foremen receiving them. Patrol orders, because of the type of work involved, require little analysis by the foreman. The job is also estimated at the time it is analyzed.

In discussing job analysis, one adverse argument is the inability to determine the extent or need of repair without a certain amount of inspection or disassembly. This is definitely recognized and only emphasizes the need for analysis to determine this condition and make ready for it and the necessity of being sure that the repair order describes the real repair as a basis for effective execution and proper record for corrective maintenance.

It also seems obvious that this is a method whereby a foreman should investigate each job to determine the above conditions if he is interested in securing effective work and lower costs. The use of the scheduling rack and the knowledge of job analysis enable the foreman to schedule his maintenance work in a formal manner, but it is also essential that materials be available before any work is formally scheduled

One of the most recurrent criticisms of job analysis in its relation to scheduling and patrolling of work by the maintenance foreman is the time it requires Even though actual study has indicated that this criticism is fictitious, it persists To assist the foreman in a better understanding of what job analysis means in the proper completion of work and in knowing those factors that enter into a thorough analysis, job-training programs have been established, which cover all the tools, procedures, or services that are needed to do a good maintenance job

Scheduling Maintenance Work. Scheduling maintenance work consists in fitting analyzed jobs into a daily planning and scheduling sheet (Fig 6-11) according to operating conditions, production schedules, personnel, and availability of materials.



DATE

AREA

FOREMAN

MAN'S NAME AND NUMBER

SHORT DESCRIPTION OF THE JOB

WORK

JOB

MAN

W.O NUMBER

CODE

ACTUAL TIME

CAUSE OF CHANGE

Fig 6-11 Foreman's daily planning and scheduling sheet

The purpose of scheduling is to have ready for personnel in each craft or group sufficient jobs to cover a day's work, arranged according to priority and analyzed as to labor, material, tools, etc, required and so arranged that on the completion of one job another is available for every person in the craft or group

The scheduling of maintenance work is done by the maintenance foreman assisted by a repair-order-control and scheduling clerk Analyzed jobs are scheduled according to the personnel available and are recorded on a daily planning and scheduling sheet (Fig 6-11) made out in duplicate These sheets are prepared for the following day's work by a fixed time of the previous afternoon They are then approved by the general foreman A red line is drawn under the last job recorded Any job added to this sheet after general approval becomes an emergency job and is investigated as to cause by the planning and scheduling engineer, whose duties in this respect are shown in Fig 6-3

The schedule sheet is valuable because it is

1 A formal record of actual use of maintenance labor

2. A tool to train maintenance supervision in the proper planning of work
3. A record of real emergency conditions
4. A summation of the day's activities for each foreman

On the basis that each man under each foreman has sufficient duly analyzed jobs, the effectiveness of scheduling will be measured by the number of emergency jobs.

An emergency job is one which has not been scheduled but which must be taken care of during the day the schedule sheet is in effect. In order to place the responsibility for such jobs, a list of emergency classifications has been determined (see Fig. 6-12). At the end of each week typewritten copies of investigation sheets covering all emergency jobs and their classifications are submitted to plant management and to operating supervisors in whose areas the jobs occurred. The investigation of these emergency jobs and the recording of results (Fig. 6-12*a*) for corrective purposes are functions of the planning and scheduling engineer working in conjunction with the area engineer and craft or area foreman, as indicated by Fig. 6-3. To enable these records to represent the truest picture, they are approved by the maintenance engineer before distribution. These records should be discussed with the area engineers and operating supervisors, and a definite decision should be made concerning what means are to be taken to prevent recurrence. Such records are very valuable in giving both operations and maintenance a picture of why jobs cannot be planned, what causes the emergency condition, and what means must be taken to prevent a recurrence if down time is to be eliminated and poor maintenance conditions corrected.

EMERGENCY JOB CLASSIFICATION

Plant__ __Area

For week ending__

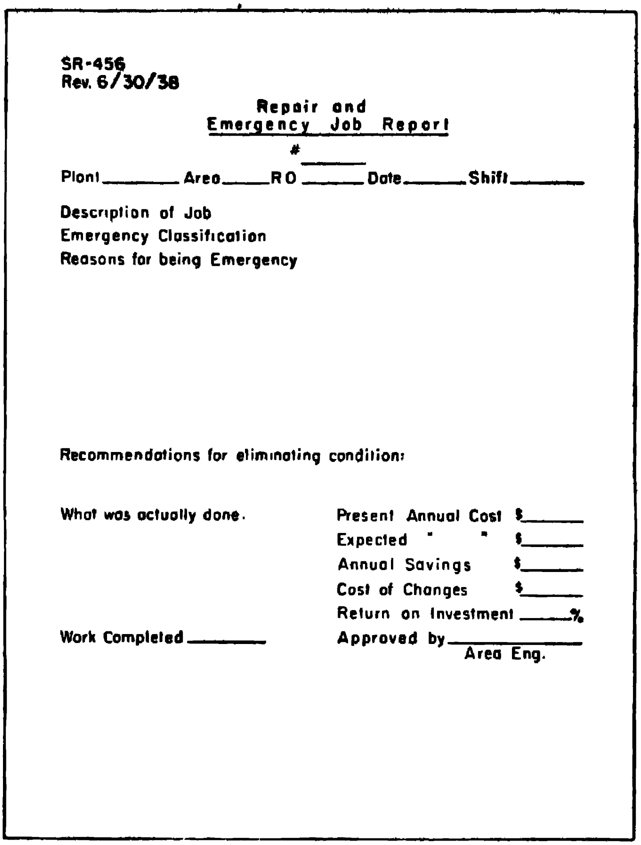
Classification		No. of jobs	Hours	Per cent of total	Remarks
E-1	Faulty electrical inspection				
E-2	Faulty mech. inspection				
E-3	Preventable by inspection				
E-4	Uninspectable				
E-5	Faulty lubrication				
E-6	Faulty equipment				
E-7	Faulty layout				
E-8	Faulty maintenance				
E-9	Faulty operation				
E-10	Faulty process				
E-11	Constr. and rearrangement				
E-12	Skilled service to operations				
E-13	Acts of God				
E-14	Caused by inspection				
E-15	Safety				
Total					

Fig. 6-12.

To reduce maintenance costs by investigating repair or replacement conditions as they occur, whether emergency or not, and by such measures training the foreman relative to the importance of this phase of his responsibility, certain repair orders designated by the maintenance engineer are also being investigated in the same manner as emergency jobs. Selection of such work is usually based on repetitive repair conditions as shown by equipment records.

Forecasting or Longer Term Scheduling. The coordination of plant maintenance and minor construction work with production and other factors requires specific planning. One method just described was daily planning and scheduling. However, there are many jobs requiring more than a day's duration or more than one craft to complete. This method of planning and scheduling is termed "weekly forecasting."

According to this method, man-power requirements by craft or major maintenance and minor construction jobs are detailed into a single master schedule (see Fig. 6-12*b*).



SR-456
Rev. 6/30/38

Repair and
Emergency Job Report
#

Plant________ Area_____ R O ______ Date______ Shift_______

Description of Job
Emergency Classification
Reasons for being Emergency

Recommendations for eliminating condition:

What was actually done.

Present Annual Cost $______
Expected " " $______
Annual Savings $______
Cost of Changes $______
Return on Investment ______%

Work Completed________

Approved by__________
Area Eng.

Fig. 6-12a. Repair and emergency job report.

WEEKLY MASTER SCHEDULE OF MAINTENANCE AND CONSTRUCTION WORK

Works ____________________

Week 2/7 to 14/46

Auth.	Dept.	Description	2/7	8	9	10	11	Job Capt.
MC-1714	Ch. zinc	No. 6 bin fireproof	2 Ms 2 L	2 Ms 2 L	2 Ms 2 L			AD
MC-1740	Sil.	Fuel-oil burner		2 PF 2 Ms 1 L	2 PF			RK
MC-1767	SA-2	Sump tank and pump	2 LB	1 E 2 LB 1 C	2 C 4 L 2 BM	2 LB 1 C 4 L 2 BM	1 E	B
C-214	ZAC	Rocker hold tank	2 BM	2 BM	2 BM			WH
C-228	24	Bldg. repairs		3 C	3 C	1 P 3 C	1 P 3 C	TP

Fig. 6-12b.

Tentative labor assignments are detailed for each individual craft or other groups in terms of man-days for each day of the 1-week schedule period. Minor and routine maintenance work is excluded in order to simplify this schedule. Labor assignments, as represented by the master schedule, are proportioned on the basis of the total labor supply in relation to the anticipated requirements of minor, routine, emergency maintenance, and minor construction work.

Correlation and supervision of the weekly schedule is a function of the planning and scheduling engineer or similarly qualified individual.

The weekly master schedule serves maintenance and minor construction foremen as a firm starting point for establishing daily work schedules. It permits coordination among various crafts, tie-ins with production shutdowns, and less frequent changes in the daily work requirements.

TRAINING

A training program is introduced for all foremen covering the use of the various procedures that affect the work.

Referring to Fig. 6-3, it is noted that job training (supervisory) is a responsibility of all engineers except the project engineer; job training (wage roll) is a responsibility of all foremen; and the planning and scheduling engineer is responsible for both. His responsibility, however, is only to assist in the preparation of training programs and instruct where assistance to supervision is necessary.

Training is usually divided into two parts—orientation and job training. In the former all subjects such as company history, policies, various company plans, etc., are included, while job training refers to procedures or methods that affect the actual maintenance work of the plant.

Training in its broadest sense can be used in various ways and is continuously used by good supervision in teaching those under them a better understanding of their jobs. By this method the basic responsibilities and duties can be both understood and taught in an informal way if all members of an organization are to accept the responsibility of their jobs to their fullest extent.

The persons fundamentally responsible for training programs are the works engineer and the planning and scheduling engineer, the former to be assured of their being used and the latter for their establishment and application. The latter responsibility also applies to all engineers and foremen. It is therefore essential that the planning and scheduling engineer be trained to assist organization in applying the procedures most effectively that will result in the best execution of maintenance and minor construction work.

Maintenance Job Standardization. Maintenance job standardization is the arrangement of the component parts of the job and the selection of correct materials, tools, equipment, supplies, and labor to determine the one best method of performance at any particular time.

The purpose of maintenance job or work standardization is to break down a job into its elemental operations in order to determine the possibilities of improvement relative to use, design, methods, etc., and to utilize the information secured in establishing a standard of performance for practical use.

Jobs must be studied by engineers who understand how such an analysis need be made. A flow chart should be made up of the elemental operations, and this should be studied from the standpoint of use of the equipment, the cause of the repair, the best method to be used, the tools required, etc. A revised chart covering all improvements should be made and discussed with both operating and maintenance supervision to secure the benefits of their experience. Any development work required as a result of this study should be turned over to the experimental or development groups referred to previously. After all corrections or improvements have been made, the process chart is issued as a standard operation sheet covering that maintenance job.

Once job standardization is accomplished, very little job analysis is necessary.

Job standardization assists the maintenance foreman in reducing analysis time since it outlines the one best way of doing the work. In addition, the investigation of job costs and their repetition determines their priority in being standardized. A job-standardization program results in a continuous review of all maintenance work for cost reduction and definitely opens the way for corrective maintenance and the use of both maintenance and development engineers in its activities.

Planning and Scheduling Engineer. The responsibilities of the planning and scheduling engineer are to institute, coordinate, and follow up methods required by a works-engineering organization for optimum conditions of performance and cost and to assist in the training of personnel in the application and use of maintenance procedures such as

1. Repair-order procedure
2. Planning and scheduling
3. Equipment-cost-card procedure
4. Job standardization
5. Budgetary control
6. Organization training
7. Preventive maintenance
8. Corrective maintenance

One of the persons to be trained by the planning and scheduling engineer is the repair-order-control and scheduling clerk in the clerical duties related to

1. Repair-order control, control of maintenance labor requirements, scheduling, maintenance-cost control, maintenance-inspection control, costing, etc., segregating from maintenance and construction supervision the following clerical duties:
   - *a.* Coding and writing repair orders
   - *b.* Mechanics of scheduling by the use of racks, job cards, time clocks, schedule sheets
   - *c.* Posting accurate records of job time to scheduling sheets
   - *d.* Accumulating and maintaining records and backlog charts, cost cards, etc.

Inventory Control of Materials. One of the sources of irritation in the successful planning of maintenance work is the unavailability of materials. This is due to some extent to the necessity of keeping capital investment for such materials to a reasonable figure and having available the many grades or sizes of maintenance materials and spare parts, etc , used in the maintenance of a plant. To this might be added the additional problem of storage space.

Experience over a period of years has indicated that two factors affect the availability of materials more than any others. They are

1. Lack of adequate records
2. Abnormal removals from maintenance stores

Figure 6-13 shows an inventory-control card, which contains all the information necessary for the proper control of any maintenance material, spare part, etc. If the maintenance department places the full responsibility on the stores department for receipt and disbursement of maintenance materials, spare parts, supplies, etc., and assists them in eliminating from stores accounts all slow-moving parts, the material problem will be greatly improved.

Closely related to this is the use of both central-stores and department-stores stock. If materials are not easily available, it is obvious that in plants covering large areas, or even in separate buildings, certain time will be lost in the handling and transportation of such materials. If it has been found to be advantageous to segregate labor by areas or buildings, the same reasoning also should be applied to materials, but such materials should be common to that area and properly controlled for stores purposes. The maintenance department can appreciably help in this particular phase of material control if it assists in both having and keeping these material records up to date.

Closely allied to the successful scheduling of maintenance work are shops, mechanical tools, and facilities. All these represent capital investment which should be made available only as found necessary. Studies should be made of the work and the use of mechanical facilities to the extent possible for greatest economies.

Such a study usually must extend over a considerable period of time as it involves both area- and central-shop requirements, as well as repetitive and nonrepetitive work. The increase in labor-material ratios as applied to maintenance costs over

INVENTORY-CONTROL CARD

Loc.	Description of article	Unit	Capt. No.	Loc.
	Vendor			

	Year	Jan.	Feb.	Mar.	Apr.	May	Jun.	Jul.	Aug.	Sep	Oct.	Nov.	Dec.	At balance of				
	19																	
	19													Change unit price to				
	19																	

Date	Ordered			Receipts	Disbursements		Balance on order	Monthly consumption to date	Balance on hand
	Order No	Requ No.	Quantity		Quantity	Code			

Fig. 6-13.

the past years from an average of 50:50 to approximately 75:25 clearly indicates that a study of these services is necessary in most plants.

Prefabrication also fits into the shops and tool picture. It is obvious that disassembly, fabrication, and reassembly can be done with greater effectiveness in a location where proper equipment and facilities are available. Such conditions are possible to a greater extent either in a central shop or area shop arranged for such work than in the field. It is not only more convenient, but it is safer.

The use of job analysis and job standardization represents a degree of analysis where definite indications should be evident as to the possible extent of prefabrication. Repetition of work and studies in handling and transportation will contribute information that will aid in determining to what extent prefabrication can be used.

Shop layout must also be considered, particularly because of the need for available floor space in order to utilize this service to do the job properly; to have room to place and fabricate parts; and to have handling facilities such as cranes, monorails, chain falls, etc., to reduce fatigue and expedite work. Power-driven hand and machine tools are available in practically every line of craftwork, and each one has its place in cost reduction. The same applies to cutting tools, dies, fixtures, etc.

PREVENTIVE MAINTENANCE

Preventive maintenance consists in establishing the means to reduce maintenance costs through a knowledge of the necessity of repair. Therefore, the real instrument to meet this condition is inspection.

Maintenance inspection is a procedure for determining the necessity for either major or minor repairs. Its purpose is to prevent, from any cause, unscheduled interruptions to operating equipment and undue deterioration to buildings and facilities.

An inspection procedure operates in two ways:

1. Visual or working inspection, or observing equipment in operation to determine potential breakdowns and making adjustments or small repairs
2. Shutdown inspection, or determining the wear and general condition when equipment is not operating

In addition, inspection is necessary for all parts of an industrial plant such as buildings, roofs, lines, etc. This might also be called visual inspection, although it may be done at less frequent intervals. In establishing an inspection procedure, all buildings, equipment, etc., should be listed and analyzed to determine whether they should be inspected on a visual or shutdown basis.

An inspection system must be applied in as simple a way as possible, with the duties of individuals definitely delegated and results maintained if it is to be of value and succeed in its purpose. Therefore, it should be an integral part of actual repair to as large an extent as possible.

The organization of a maintenance department was described as consisting of craft and shop groups and area groups. If there were to be a segregation of these groups as to inspection, based on the kind of work done and the size of the job as well, we would say that the work of the crafts and shops dealt more with shutdown inspection, while that of the area groups dealt with visual or working inspection.

It is therefore necessary that the extent of the use and kind of inspection be based on an actual study of work conditions.

Assuming that all maintenance work is done either by patrol (Fig. 6-8), or normal repair orders (Figs. 6-6 and 6-7), it is obvious that the work covered by each of these must be analyzed over a definite period of time to determine the kind of work involved, amount of each kind, frequency, and individual times required. After this has been done, it becomes a matter of establishing a standard task with definite work and personnel requirements covering all work of a visual-inspection nature. Infrequent small jobs must still be covered by written patrol orders.

The size and frequency of overhaul of the larger jobs will establish their frequency for shutdown inspection. This frequency is very vital to the success of an inspection system, since too long an interval will not describe potential breakdowns in time to apply preventive maintenance, and too short a period results in excessive inspection cost and reduces equipment operating hours

Additional frequency-determining factors for shutdown inspection are

1 Operating time
2 Individual equipment conditions, such as
   a Speed
   b Complexity
   c. Materials of construction
   d Materials processed
   e Surrounding conditions
3 Importance to plant operations

Another important method of determining frequency of inspection consists in dividing annual maintenance costs by investment, *i e*, if the estimated inspection

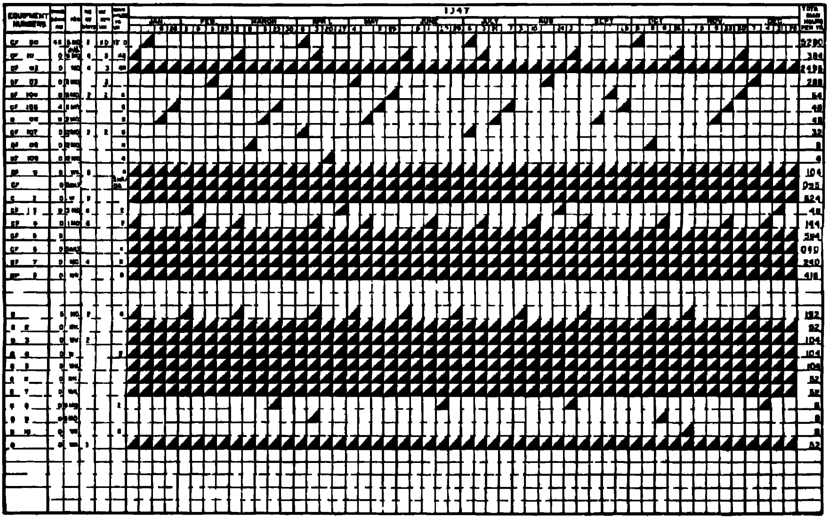

1947
JAN
FEB
MARCH
APRIL
MAY
JUNE
JULY
AUG
SEPT
OCT
NOV
DEC

FIG 6-14 Plant equipment inspection schedule.

frequency is high, the maintenance to investment ratio will usually be found to be high, and vice versa

This means that, if maintenance costs are not kept by equipment, which will be explained under Corrective Maintenance, this information will not be available for establishing inspection frequencies based on work done

Inspection Schedules. After all inspection frequencies have been determined, a schedule (see Fig 6-14) is established covering all equipment in this category (The same can be done for buildings, facilities, etc) The equipment is not scheduled any closer than a week, since arrangements must be made with the operating department to take the equipment out of service at a time convenient to production schedules Space is also allotted on this sheet to show the actual date on which the inspection is completed, since this serves as a follow-up and information for future scheduling

In addition to the inspection schedule, there should also be a record of the condition of each piece of equipment when inspection was made This record will also serve as a basis for repair as well as a continuous master record of the condition of that piece of equipment Such a maintenance inspection operation sheet is shown in Fig 6-15, which is laid out to fit in with the ideas of job standardization as to the proper arrangement of the component parts of the job and the selection of correct materials, tools, etc

MAINTENANCE INSPECTION
STANDARD OPERATION SHEET

Bldg. No ____________ Date ____________
Equipment No ____________ Inspector ____________

Operation	Date conditions found	Date conditions found	Date conditions found	Date conditions found	Date conditions found
Disassembly					
Inspection and adjustment					
Assembly					

Standard tools Standard materials

Fig. 6-15

CORRECTIVE MAINTENANCE

There are five requisites to a corrective maintenance program, *viz*:

1. A Real Desire for Improvement. Too often responsible personnel in a plant tends to become close-minded relative to possible improvements because of a lack of facts or figures to indicate the condition. In order to overcome this last condition, the engineer must accumulate as many measuring factors as is possible to indicate the need for improvement. For instance, the maintenance cost as per cent of investment of a total plant may seem to be quite reasonable, whereas a breakdown by individual or groups of similar equipment may be quite the reverse. Also the total maintenance labor-material ratio might not be indicative of any unusual condition, whereas a similar breakdown by individual equipment might show some very outstanding discrepancies. The collection of such informative data will very definitely hasten a desire for improvement.

2. Engineering Assistance. A review of Fig. 6-3 indicates that while constructive criticism can be given by supervision of the engineering department toward any corrective maintenance program, supervision is not in a position to make actual studies either of jobs or of accumulated information. These are functions of the planning and scheduling or methods engineer.

3. A Practical Inspection Program (discussed under Preventive Maintenance).

4. Job-standardization Program (page 284).

5. An Equipment-cost-card Procedure (see below).

To state that 5 is superfluous is to deny facts that indicate the tremendous possibilities to plant improvement through the use of an equipment-cost-card procedure as well as an active support of 2, 3, and 4. Under Cost Controls will be shown charts of the results of maintenance conditions in two plants where one was receiving active support of improvement programs and the other was not.

Equipment-cost Cards. An equipment-cost-card procedure is a systematic posting of all completed maintenance work according to individual building or equipment.

EQUIPMENT-COST CARD

Equipment:
Specifications.

Code No:
Bldg. No:

Date	Shop order No	Description of work	Labor			Material			Total labor and material
			Const.	Direct	Indirect	Const	Direct	Indirect	
		Brought forward							

Sheet No ______

Equipment No ______

FIG. 6-16.

The purpose is to provide a system of records of plant maintenance and construction work by buildings, equipment, facilities, etc., and to have available a means of easier analysis of jobs for both preventive and corrective maintenance.

The equipment-cost card (Fig. 6-16) provides for the posting of the following information:

1. Repair-order number and date
2. Description of work done and material used according to key method
3. Labor and material costs by direct and indirect
4. A periodic summary of data as a basis for maintenance improvement

Equipment cards should cover

1. Individual buildings
2. Groups of similar equipment
3. Utilities such as the main air, steam, and water lines, roads, outside tanks, etc.
4. Blanket orders as blanket equipment number
5. Burden codes such as shops, supplies, hospital, fire protection, etc., as burden equipment numbers

Postings are made of completed normal orders. Patrol orders are posted only as a summation of monthly costs according to equipment numbers.

In this section reference has been made a number of times to a blanket repair order. This type of order covers a series of regular repetitive scheduled jobs. While it very definitely can be misused and can become a dumping ground for miscellaneous charges, if used correctly it very definitely has a place in a maintenance program. In the equipment-cost-card procedure blanket orders become blanket equipment numbers. It is very necessary that all maintenance costs be covered by the equipment-cost-card procedure.

Some criticisms concerning the use of the equipment-cost cards for keeping maintenance costs are

1. It requires the numbering of all equipment, buildings, etc., and represent additional codes to the plant.
2. It requires clerical assistance to maintain
3. It requires clerical assistance to publish a report semiannually.
4. In some cases the information is not used after collected.

Answers to these criticisms are as follows:

1. Equipment-cost cards do require the numbering of all plant equipment, buildings, etc., but in most cases they must be numbered for inventory purposes anyway. The equipment numbers also serve as departmental codes for cost purposes, and no additional coding is required.

2. They do require clerical assistance to maintain, but (*a*) the repair orders represent the actual records of work done, and this method of recording tends to reduce engineering experience required for analyzing purposes, thus reducing the clerical effort to a minimum. In addition, the need for better description as required for the equipment posting tends to necessitate better analysis of the repair orders by the foreman. Simplicity of posting is secured by the use of the key method which records on the equipment-cost card only the equipment number, part repaired, and total labor and material costs and is usually limited to one line. The equipment-cost card does not take the place of the standard operation sheet used for job standardization or shutdown inspection, as this would only clutter up the card with information that has already been established elsewhere.

(*b*) They give a summary of results over a sufficient period of time to determine trends and allow these results to be analyzed for corrective purposes.

3. As stated previously, it is necessary to want improvement to have it. If a plant or maintenance organization is not interested either in improvement or correct frequencies of inspection for preventive maintenance, it is useless to keep these records.

COST CONTROLS

The objective of the cost-control plan is low maintenance cost, but it is a result rather than a cause, and it is essential that all factors discussed herein be complied with if these costs are to be attained.

Maintenance costs cannot be measured as a lump sum or in relation to the product or products as a whole. Maintenance costs are not that simple to interpret but are representative of various plant conditions which must be understood if these costs are to be of value in indicating actual conditions. The plan's approach to this problem of cost interpretation is to segregate maintenance costs into three classes, which are described as follows:

Cost Classification—Direct, Indirect, and General. Cost classification is a method of segregating maintenance labor and material costs according to three classes of maintenance known as direct, indirect and general.

Direct Maintenance. Direct maintenance covers repairs to operating equipment or auxiliary equipment attached to it. It also includes miscellaneous repairs such as oiling machinery, packing pumps, inspections, etc. Direct maintenance should be proportional to production.

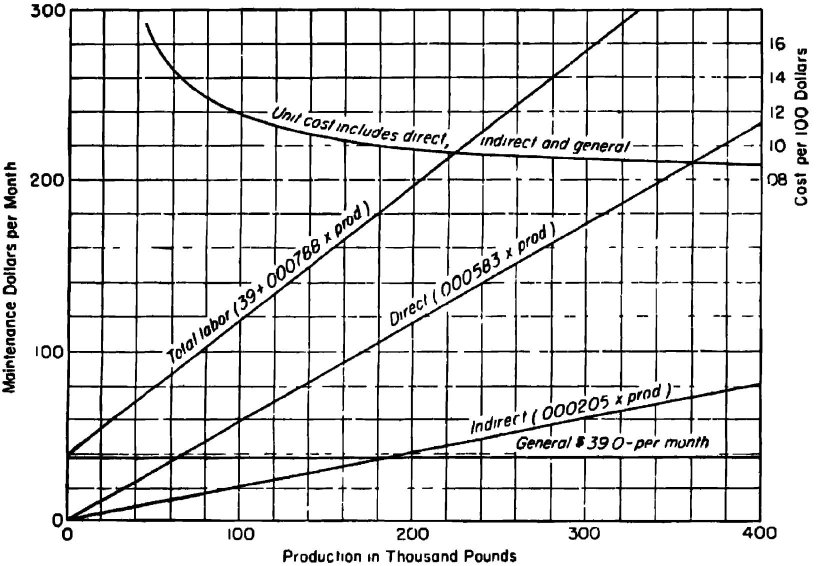


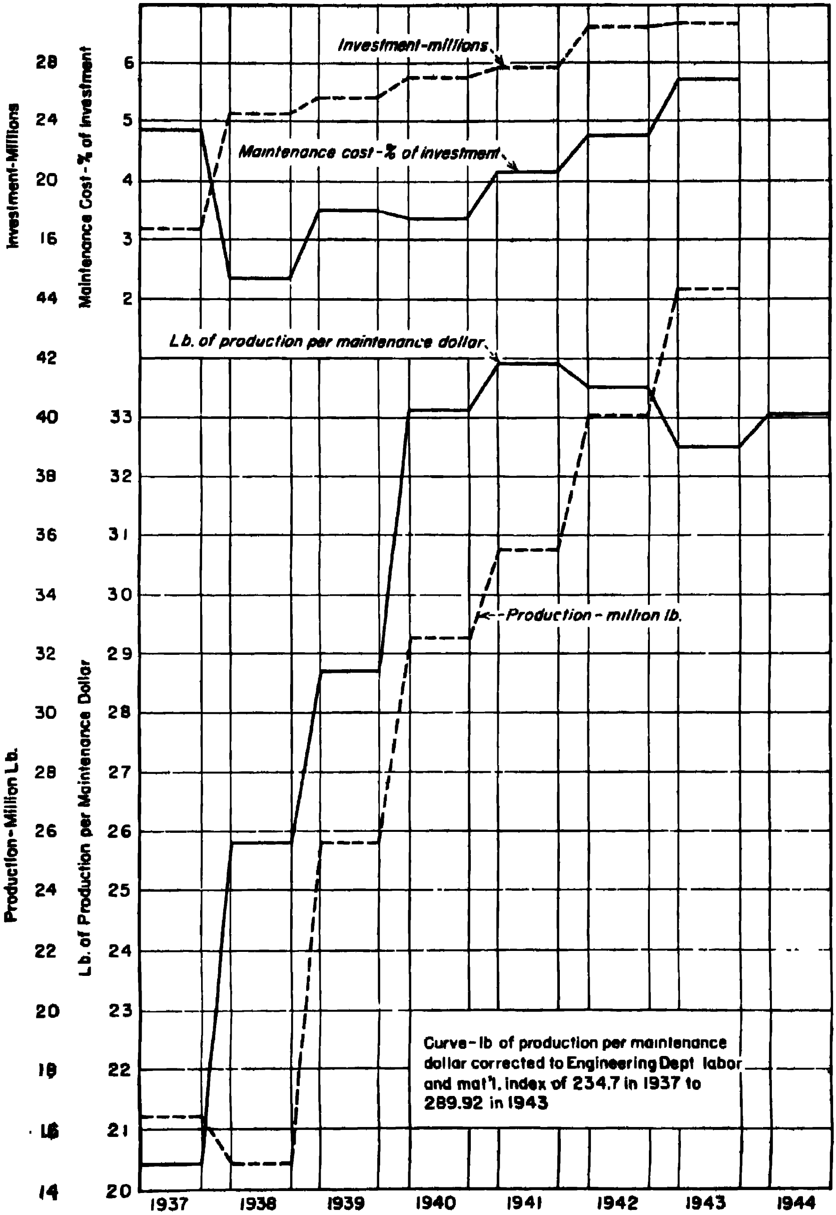
Fig. 6-17. A chart for establishing monthly budgets based on direct, indirect, and general maintenance costs.

Indirect Maintenance. Indirect maintenance covers repairs or replacement of parts, such parts being not entirely worn out, replacing them with parts of improved design, additions (when not capitalized), replacements (when they reflect changes in design), materials and methods only, rearrangements, and service to operations.

General Maintenance. General maintenance covers repairs to buildings, outside lines, storage tanks, roads, main power, steam, air, or process lines, and structural support of buildings, but not that supporting equipment. General maintenance covers any type of repair that does not vary with production.

The purpose of this classification is to segregate the various classes of maintenance as to responsibility, conditions, etc., in order that cause can be investigated and improvements made accordingly; to emphasize what work is or is not affected by production; to establish a control of indirect work and determine its relationship to direct and general classes in similar plants.

In charging all maintenance costs in a lump sum, the variables due to production, development changes, service conditions, etc., are lost sight of, as well as those conditions in which production has little effect such as weather, service, burden, etc. By dividing maintenance costs according to the three classes, a means is provided



Curve-lb of production per maintenance dollar corrected to Engineering Dept labor and mat'l. index of 234.7 in 1937 to 289.92 in 1943

FIG. 6-18. A chart for measuring annual maintenance effectiveness; indicates progress.

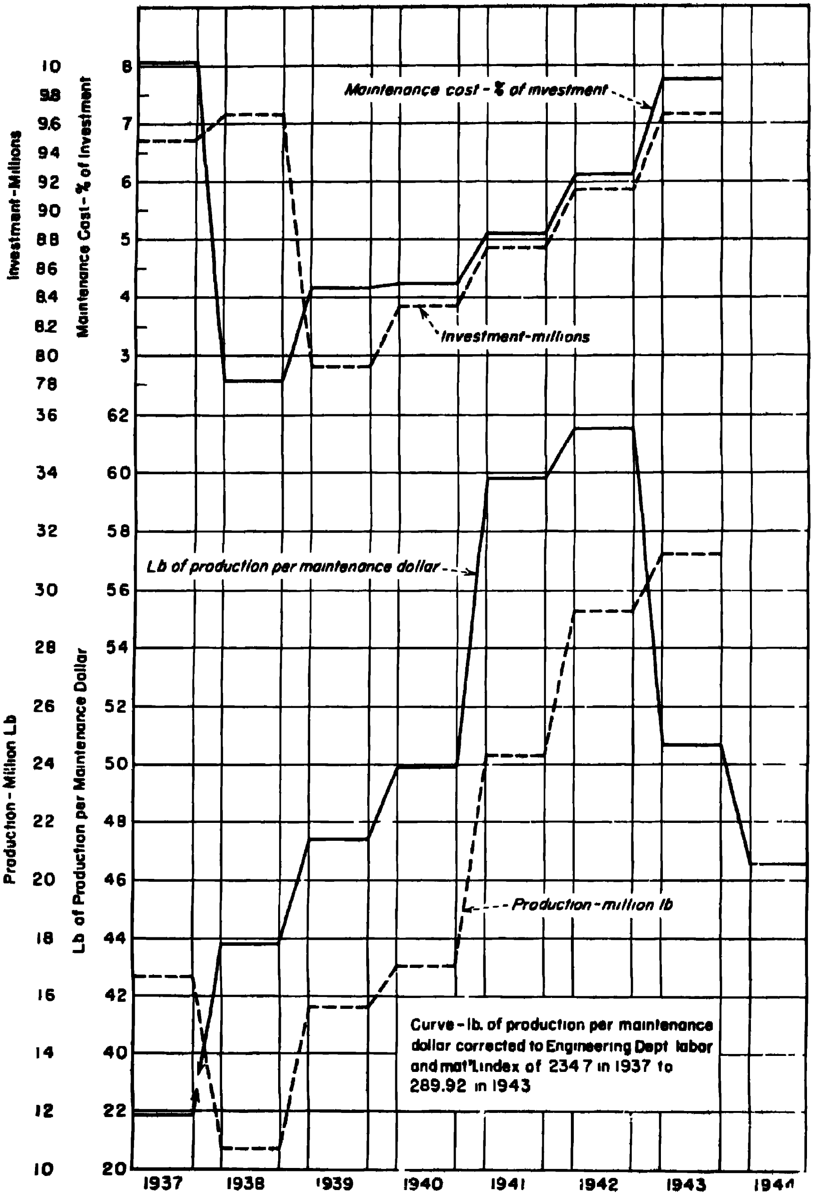


FIG. 6-19. A chart for measuring annual maintenance effectiveness; indicates retrogression.

for definitely segregating the work that is a direct function of a maintenance organization and that which must be justified by the other departments.

Budgets. A monthly budget chart established according to the three classes of maintenance discussed is shown in Fig. 6-17. Figures making up such a chart must be worked up carefully covering a number of years for any particular plant in order to reflect an average condition of maintenance. The chart also must represent maintenance conditions at various levels of production which is so necessary in determining budget figures.

MAINTENANCE DEPARTMENT
MONTHLY REPORT

1. Production costs	Current month	Average 12 months
Net production, lb or unit		
Total maintenance costs (labor and material)		
Unit cost, per 100 lb, ton, etc		
Replacements		

2. Summary of planning and scheduling performance	Current month	Average 12 months
Total man-hours		
Per cent scheduled		
Emergency work, per cent		

3. Mechanical department labor	Current montn	Average 12 months
Blanket work orders	$	$
Normal work orders		
Emergency		
Patrol		

4. Mechanical department labor, per cent	Current month	Average 12 months
Blanket	31	30.7
Normal	46.7	50.5
Emergency	17.2	14.2
Patrol	5.1	4.6

5. Breakdown of work order labor and material	Direct	Indirect	General	Total
Total				
Per cent				
Average, 12 months				
Per cent				

6. Work orders	Current month	Average 12 months
Normal		
On hand first of month	454	374
Issued during month	264	284
Completed during month	256	263
Cancelled during month	13	10
On hand end of month	450	383
Blanket	66	63
Patrol	173	139
Emergency	88	58

7. Organization chart

FIG. 6-20.

On the basis that budgets are made to indicate possible costs, that meeting budgets is only balanced spending, and that improvements will be reflected in the use of budgets, it is obvious that the same rules must be applied to this manner of measuring costs as to any other.

Charts. A method of measuring maintenance results at two plants is shown in Figs. 6-18 and 6-19. The bases of measurement are

1. Maintenance cost as per cent of investment
2. Pounds of production per maintenance dollar

In order to have conditions comparable, all maintenance costs are corrected to an index figure (1) as shown on the lower part of the chart. Maintenance cost as per cent of investment reflects actual costs.

To show improvement it is necessary that the heavy line in the lower part of the chart indicating pounds of production per maintenance dollar continue its upward path with the production or dotted line. Conversely, a trend in the opposite direction indicates lack of improvement.

Maintenance cost as per cent of investment shown on the upper part of the chart by the heavy line indicates a method of measuring results over a period of time and also indicates whether maintenance improvement has been secured by expenditure of capital dollars. This will also be reflected if the maintenance department makes improvements through the application of methods and better procedures without additional capital expenditure.

It was previously stated that charts would be shown, one of which represented a plant where improvement was about continuous, while the other showed a definite backward trend. Figure 6-18 is an example of the first case, while Fig. 6-19 represents the second case.

Monthly Report. Another measure of results is the maintenance monthly report. A proposed form to be used covering the result of the use of some of the procedures outlined in this section is shown in Fig. 6-20. It is believed that not only this report, but all the other measures outlined, are essential for good maintenance.

The results to be gained by using the basic principles outlined in this plan can be summed up as follows: The plan constitutes

1. A method to maintain a plant in good operating condition at a minimum of cost
2. A means to correct and improve
3. A basis for selecting a high type of maintenance supervising personnel

Maintenance Measurement (Actual Time Values). All the foregoing is measurement, and it is believed necessary before actual time measurement is of any real value. In establishing time measurement standards, experience has indicated that their failure in most cases has been due to one or more of the following:

1. Lack of job standardization, or establishing the one best method
2. Insufficient knowledge and training of personnel in establishing the standards
3. Lack of consideration and understanding of the importance of maintaining the standards up to date, taking into consideration both process and technological changes

Although time measurement can be successfully applied to maintenance and/or any other factor of production, it should not be effected until the above or some similar plan has been applied, since only confusion and adverse reaction will occur.

SECTION 7

INDUSTRIAL WATER PURIFICATION AND TREATMENT

BY D. C. CARMICHAEL

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INDUSTRIAL WATER PURIFICATION AND TREATMENT

BY D. C. CARMICHAEL

Chemical Engineer, E. I. du Pont de Nemours & Co.

Water is a fundamental requirement for every industrial plant. It supplies feed water for all steam-power plants; cooling water for internal-combustion engines and steam condensers; means of washing, preparing, and distributing the pulp for paper manufacture; washing requirements for textile mills, dyeing establishments, and laundries; and enters in some manner into all major industries.

The fitness of the water supply for the respective industries has frequently been a determining factor in the location of manufacturing industries, especially in case of textile and dyeing industries. With the industrial development of our country, we have long since passed through the period when the character of the water was a determining factor in the location of an industry. With the advancement made in the art of water purification, it is only the exceptional water supply which cannot be treated or purified to meet the requirements of any of our major industries.

The substances in water that is objectionable for use in various industries are chiefly the mineral solids or gases. The bacteriological growths in water supplies, which must be so carefully controlled in the large municipal plants where water is prepared for drinking purposes or food products, are not commonly of much significance for most of the industrial requirements. This, however, does not always apply, since bacteria in water supplies frequently are the cause of slime deposits objectionable in the paper industry as well as in other processes where water is used for cooling.

It therefore becomes necessary to determine all the various constituents in the water that may be objectionable to any particular process requirement before the best method of water purification can be determined. When the process requirements are understood and the constituents of the water are known, a practical treatment with the necessary equipment can nearly always be prescribed. For procedure on analysis of water, the American Public Health Association's "Standard Methods for the Examination of Water and Sewage" is a standard reference. Table 7-1 is a conversion chart frequently used in water analyses.

NATURAL WATERS AND THEIR IMPURITIES

Water is a tasteless, colorless, and odorless liquid. It is a chemical compound which is expressed by the formula H_2O, indicating the combination of two volumes of hydrogen and one volume of oxygen. Chemically pure water is an extremely rare article and difficult to obtain since it is an almost universal solvent, and practically all substances are soluble in it to some degree. Because of this property, water is readily contaminated by substances with which it comes in contact.

Table 7-2 gives a list of the constituents present at times in some waters and which may be objectionable when present.

Generally speaking, the preparation of water for industrial purposes is accomplished by the utilization of equipment for performing the services of

1. Sedimentation
2. Filtration
3. Softening
4. Deaeration

SEDIMENTATION

Sedimentation, filtration, softening, and deaeration may be used either alone or in combination, depending upon the condition of the raw water. In industrial plants,

Table 7-1. Conversion Chart

	CaO	MgO	Na_2O	HCO_3	CO_2	SO_3	Cl_2	$CaCO_3$	$MgCO_3$	Na_2CO_3	$NaHCO_3$	NaOH	$CaSO_4$	$MgSO_4$	Na_2SO_4	$CaCl_2$	$MgCl_2$	NaCl
CaO		0.7191	1.1056	2.1762	0.7848	1.4279	1.2647	1.7848	1.5039	1.8904	2 9965	1.4270	2.4279	2.1470	2.5335	1.9794	1.6985	2.0850
MgO	1.3906		1.5375	3.0262	1.0914	1.9857	1.7588	2.4820	2.0913	2.6387	4.1670	1.9844	3.3764	2.9857	3.5232	2.7526	2.3619	2.8995
Na_2O	0 9045	0.6504		1.9683	0.7098	1.2913	1.1439	1.6143	1.3601	1.7099	2.7103	1.2907	2.1961	1.9419	2.2916	1.7903	1.5363	1.8859
HCO_3	0.4595	0.3305	0.5081		0.3606	0.6562	0.5812	0.8201	0.6911	0.8687	1.3770	0.6557	1.1157	0.9866	1.1642	0.9096	0.7805	0.9581
CO_2	1.2741	0.9162	1.4087	2.7727		1.8194	1.6115	2.2741	1.9161	2.4086	3.8179	1.8182	3.0936	2.1730	3.2282	2.5220	2.1641	2.6566
SO_3	0 7003	0.5036	0.7743	1.5240	0.5496		0.8857	1.2499	1.0531	1.3239	2.0985	0.9993	1.7003	1.5036	1.7743	1.3862	1.1895	1 4602
Cl_2	0 7907	0.5686	0.8742	1.7208	0.6206	1.1291		1.4112	1.1891	1.4947	2.3692	1.1283	1.9197	1.6976	2.0032	1.5651	1.3429	1.6486
$CaCO_3$	0.5603	0.4029	0.6195	1.2193	0.4397	0.8001	0.7086		0.8426	1.0592	1.6789	0.7995	1.3604	1.2030	1.4196	1.1090	0.9517	1.1682
$MgCO_3$	0.6650	0.4782	0.7352	1.4471	0.5219	0 9495	0 8410	1.1868		1.2570	1.9925	0.9489	1.6145	1.4277	1.6847	1.3162	1.1294	1.3866
Na_2CO_3	0 5290	0.3804	0.5849	1.1512	0.4152	0.7554	0.6691	0.9441	0.7955		1.5850	0.7549	1.2844	1.1358	1.3403	1.0471	0.8964	1.1028
$NaHCO_3$	0.3337	0.2400	0.3690	0.7262	0.2619	0.4765	0.4221	0 5956	0.5019	0.6309		0.4762	0.8103	0.7165	0.8455	0.6606	0.5668	0.6958
NaOH	0.7008	0.5039	0.7748	1.5250	0.5500	1.0007	0.8863	1.2507	1.0538	1.3247	2.0999		1.7015	1.5046	1.7755	1.3871	1.1903	1 4612
$CaSO_4$	0 4119	0.2962	0.4554	0.8963	0.3233	0.5881	0.5209	0.7351	0.6194	0.7786	1.2341	0.5877		0.8843	1.0434	0.8153	0.6836	0.8588
$MgSO_4$	0.4658	0.3349	0.5149	1.0136	0.3656	0.6651	0.5891	0.8313	0 7004	0 8805	1.3956	0 6646	1.1308		1.1800	0.9219	0.7911	0.9711
Na_2SO_4	0.3947	0.2838	0.4364	0.8589	0.3008	0.5636	0 4992	0.7045	0 5936	0.7461	1.1827	0.5632	0.9583	0.8474		0.7813	0.6704	0.8230
$CaCl_2$	0.5052	0.3633	0.5589	1.0994	0.3965	0 7214	0.6390	0.9017	0.7597	0 9550	1 5138	0 7209	1.2266	1 0123	1.2800		0.8581	1.0534
$MgCl_2$	0.5888	0.4234	0.6509	1.2812	0.4620	0 8407	0 7446	1.0509	0.8854	1 1130	1 7642	0.8402	1 4296	1.2641	1.4917	1.1654		1.2276
NaCl	0 4796	0.3449	0.5303	1.0437	0.3764	0 6849	0 6066	0.8560	0.7213	0 9067	1 4371	0.6844	1.1645	1.2964	1 2151	0.9493	0.8146	

where the quality of the water is not an important factor, sedimentation alone may satisfy all requirements Many plants install impounding reservoirs for preliminary settling basins since they serve not only for removing part of the turbidity but also as collecting basins for recovered cooling water Not infrequently, a spray cooling system can be installed at one end of the impounded reservoir for lowering the temperature of recovered water An impounding reservoir is especially suitable where the original source of water is a river from which pumping can be carried on during the off-peak electric load

Table 7-2. Common Impurities in Water

Constituent	Chemical formula	Principal source of contaminating substance	Treatment for removal from water
Suspended solids		Surface drainage trade waste	Plain subsidence coagulation filtration evaporators
Silica	SiO_2	Mineral deposits	Plain subsidence coagulation, filtration
Calcium carbonate	$CaCO_3$	Mineral deposits	Softening by chemical, by base exchange materials
Calcium bicarbonate	$Ca(HCO_3)_2$	Mineral deposits	Softening by heaters by chemical or by base exchange materials
Calcium sulfate	$CaSO_4$	Mineral deposits	Softening by chemical by base exchange materials
Calcium chloride	$CaCl$	Mineral deposits	Softening by chemical by base exchange materials
Magnesium carbonate	$MgCO_3$	Mineral deposits	Softening by chemical by base exchange materials
Magnesium bicarbonate	$Mg(HCO_3)$	Mineral deposits	Softening by chemical by base exchange materials
Magnesium chloride	$MgCl_2$	Mineral deposits	Softening by chemical by base exchange materials
Free acids	HCl H_2SO_4	Mine drainage trade waste	Neutralizing acid followed at times by softening
Sodium chloride	$NaCl$	Sewage trade waste mineral deposits	Evaporator
Sodium carbonate	Na_2CO_3	Mineral deposits	By acid exchange material
Sodium bicarbonate	$NaHCO_3$	Mineral deposits	By acid exchange material
Carbonic acid	H_2CO_3	Absorption from the atmosphere mineral deposits decomposition of organic matter	Deaerator
Oxygen	O_2	Absorption from the atmosphere	Deaerator
Grease and oil		Trade waste	Coagulation filter
Organic matter and sewage		Domestic and trade wastes	Coagulation filter

The determining factors in favor of preliminary settling basins are

1 Reduction in the cost of coagulation and filtration resulting from a smaller amount of chemicals required for these purposes

2 Better clarification in coagulation and secondary basins, resulting in reduced load on filters

3 Increased length of runs between cleaning of filters, with reduction in wash water

Chemical Coagulation with Sedimentation. Sedimentation in its simplest form consists of passing the raw water through specially designed horizontal basins at comparatively low velocities so that the solids are precipitated of themselves or by the aid of a coagulant In cases where waters are extremely high in turbidity, *e g*, Mississippi River water, sedimentation may be divided into two stages (1) plain subsidence, in which a greater portion of the suspended matter is allowed to settle without the aid of artificial means, (2) sedimentation with the aid of a coagulant

In most cases, coagulation is coupled with the process of sedimentation, and the most common chemical used as a coagulant is alum or aluminum sulfate.

Sedimentation-basin Design. Present-day design of sedimentation basins calls for equipment for the proper mixing of the coagulant with the water before it enters the sedimentation basin. The apparatus may vary in design, but the sole purpose is to ensure the uniform distribution of the chemical with the water. After the mixture of the coagulant with the water is made, it travels horizontally through the basin at a velocity usually not exceeding 9 to 12 in. per min. The basins are usually rectangular with a total depth not exceeding 14 to 16 ft.

The retention period may vary from 1 to 4 hr, depending upon the condition of the raw water or the degree of sedimentation desired. Frequently, the flow through the sedimentation basin is interrupted by baffles to ensure freedom from short-circuiting and maximum precipitation in the time available. The basins are constructed so that the precipitated matter, after accumulating over a length of time, can be washed to waste through a drain. This requires taking the basin out of service, and, where justified, designs can be provided for continuous collecting of the sludge at one end of the basin and intermittent removal by manual or automatically controlled blowoff.

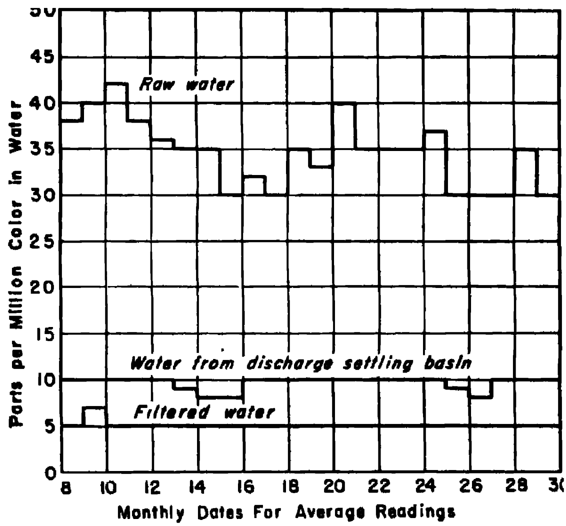


Fig. 7-1. Color content of water in various steps of removal in sedimentation basin.

Color Removal. In addition to the requirements for highly turbid water, sedimentation is necessary for the removal of coloring matter. Coloring matter is prevalent in woodland areas, particularly in swamp districts such as portions of Florida, Wisconsin, and New York State. The color in these waters varies from that which cannot be detected by the eye to that which resembles weak tea. The color is detrimental to many industrial purposes such as papermaking and textiles. While these waters are usually very soft and quite free of suspended matter, removal of the coloring matter presents a problem different from the removal of suspended matter. Aluminum sulfate customarily is used as the coagulant, and in some cases chlorinated copperas has come into use as a highly efficient coagulant for the removal of color. Figures 7-1 and 7-2 show the color content of water in various steps of removal in a sedimentation basin and the effectiveness of chlorinated copperas as compared to alum in removal of color. The chlorine and copperas (ferrous sulfate) form the chlorinated copperas but are fed as separate chemicals.

The removal of color by means of a coagulant is not a chemical process as is the case with suspended matter where the precipitated coagulant, spongelike in nature, mechanically gathers together the finely divided suspended matter. In the case of coloring matter, the reaction is chemical, and the coagulant actually reacts with the coloring matter to form a substance known as a "lake." This substance is more delicate in structure than the aluminum hydrate formed in the coagulation of suspended matter in the absence of color. The delicacy of this precipitate brings about a problem in filtration that is not found where suspended matter only is to be removed.

Iron Removal. Sedimentation has been referred to in connection with the preparation of surface waters. However, it is frequently necessary in the preparation of well waters containing iron. There are certain districts in Illinois, New Jersey, Florida, and the Western states where well waters are known to contain iron having the property of precipitating a brown iron-rust-like sludge after standing. This sludge was originally in solution in the form of ferrous carbonate, and upon becoming exposed to the air it precipitates as ferrous hydrate.

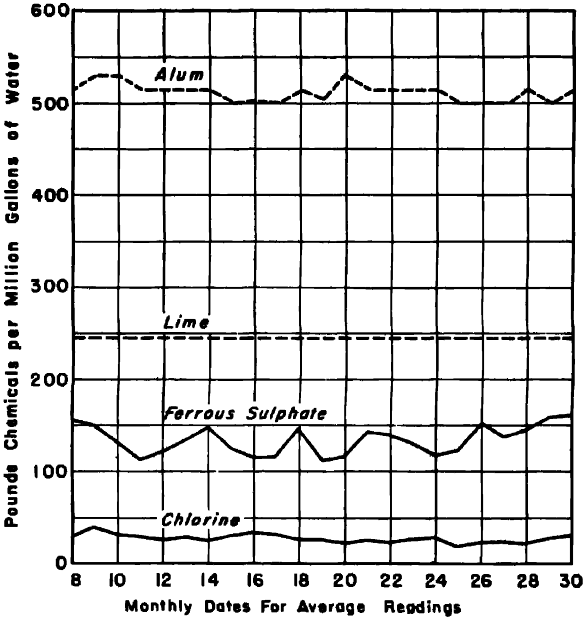


FIG. 7-2. Effectiveness of alum vs. chlorinated copperas in removal of color.

For industries where clear water is essential, it is necessary first to remove the iron from solution before it can be used. The process involves aeration and, usually, sedimentation. Sedimentation, however, is abbreviated compared to that required for the removal of suspended matter and color, and, where the amount of iron is high, less than an hour is required. It is customary in the removal of iron to add either a coagulant or caustic soda in order to bring about better flocculation and precipitation of the iron particles. By a combination of aeration, sedimentation, and filtration, the iron content of the water may be reduced to within limits of 0.2 ppm and in certain cases to 0.01 ppm.

Chemical Control. In the use of coagulants in the process of sedimentation, it is highly desirable to predetermine the most efficient dosages. These coagulants are temperamental and, under slightly varying conditions, may not behave as expected. It is therefore necessary to maintain within delicate limits a certain condition of acidity or alkalinity in the water in order to ensure the most efficient point of coagulation. This point is usually referred to as the "optimum point" and is determined in the laboratory by very delicate indicators for excess alkalinity and acidity as pH or hydrogen-ion values. When the determination is made in the laboratory, it is important to maintain the operating conditions exactly as the laboratory experiments dictate. In many cases, instruments are used where the coagulant is fed not so much in accordance with the flow of raw water as on the basis of the pH or hydrogen-ion value.

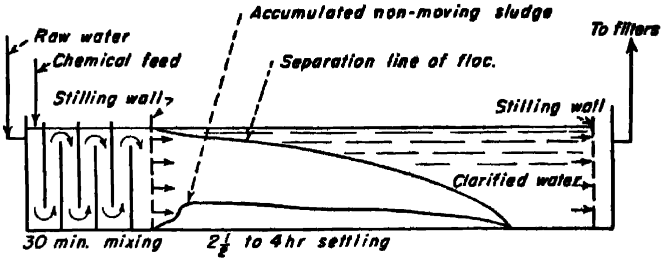


FIG. 7-3. Graphic illustration showing settling in standard mixing chamber and settling basin.

Mechanical Settling. Recent developments in water-purification plants have shown mechanical settling to be superior in many cases to the usual rectangular horizontal basin not only in first cost but also in operating and chemical costs. Mechanical settling, frequently referred to as "rapid sedimentation," is offered by a number of companies, among which are Dorr Company, International Filter Company, and the Permutit Company.

The advantage of mechanical settling over the plain settling basin is that retention time required during mixing and settling is less when compared to the time required in the conventional baffled mixing chamber and settling basin. The required time is 35 to 60 min in the case of mechanical settling and $2\frac{1}{2}$ to 4 hr in the plain settling basin. Mechanical settling by the use of certain principles utilizes sedimentation capacity more efficiently then heretofore. By recycling sludge at high concentrations, coagulation is accelerated, thus reducing the time requirements for mixing. The comparison between the standard mixing and settling basin and two systems of rapid sedimentation is shown graphically in Figs. 7-3, 7-4, and 7-5.

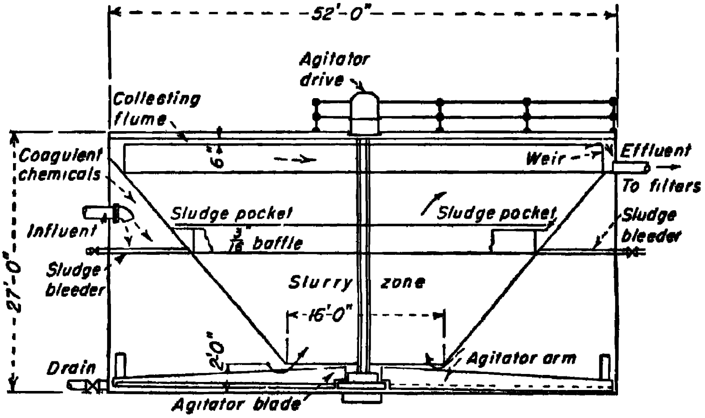


FIG. 7-4. Graphic illustration showing settling in Spaulding precipitator. (*The Permutit Co.*)

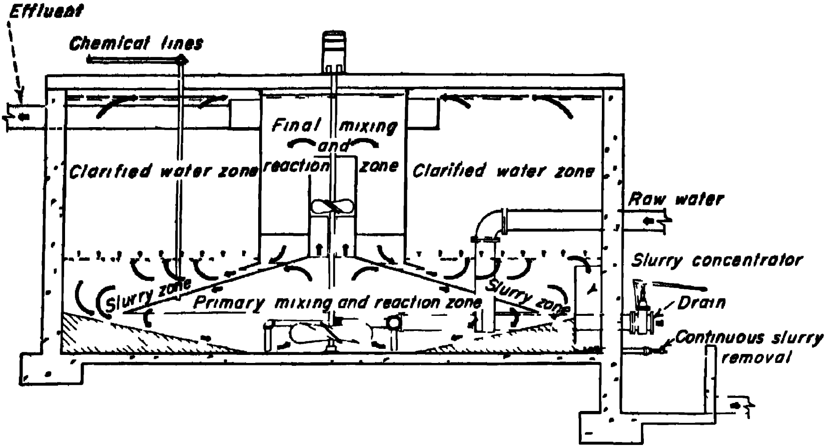


FIG. 7-5. Graphic illustration showing settling in accelator softener. (*Infilco Incorporated.*)

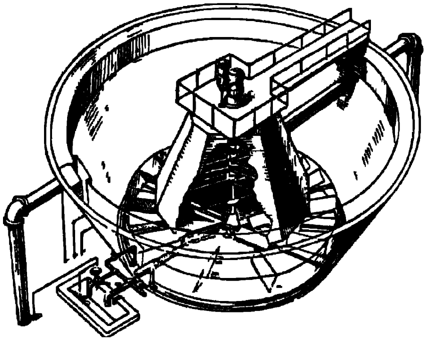


FIG. 7-6. Arrangement of vertical Spaulding precipitator. (*The Permutit Co.*)

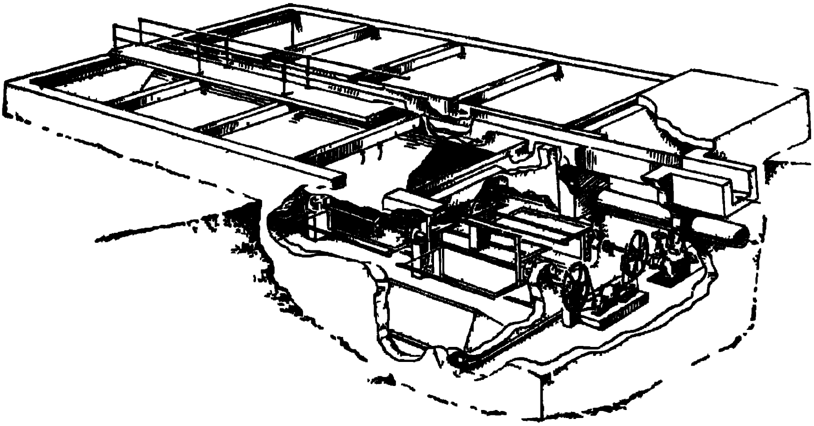


FIG. 7-7. Arrangement of horizontal Spaulding precipitator. (*The Permutit Co.*)

The operation of the standard mixing chamber and sedimentation system as shown in Fig. 7-3 indicates that clarification is accomplished by the precipitation of the individual particles of floc in free settling.

In rapid sedimentation (Figs. 7-4 and 7-5) coagulation is seeded and accelerated by the presence of highly concentrated sludge. No such seeding effect is obtainable in the standard mixing chamber. Settling occurs somewhat in the manner of filtration. The coagulated water is passed upward through a bed of suspended sludge, and the particles of floc are filtered out by the sludge bed. Advantage is taken of the rapid settling rate of the concentrated sludge—hence the high capacities of the rapid sedimentation systems. The sludge level is maintained by continuous or intermittent blowing down, and, where desired, this operation is automatically performed through an electric-program-clock control.

Figure 7-6 is a cutaway view of a vertical Spaulding precipitator, and Fig. 7-7 is a horizontal Spaulding precipitator. The latter design can be employed in converting plain settling basins to mechanical settling.

FILTRATION

The clarification of water by filtration is accomplished by passing the water through a sand bed consisting of grains suitably graded in size, preceded (if required) by coagulation and sometimes by sedimentation. Generally speaking, it is not safe to filter a surface stream without previously passing it through a sedimentation basin. Filtration is essentially a screening and frictional-resistance process. Suspended matter that is too large to pass through the interstices between the sand grains is strained out and retained on the surface of the filter bed, and smaller particles are retained by the bed through frictional resistance. Of the remaining particles, too fine to be removed by straining or frictional resistance, some are colloidal and can be removed after proper coagulation. However, it is well to remember that most filtration processes are preceded by coagulation, with or without sedimentation, in order to make the filtration process more efficient. When water flows through a filter, coagulated solids are retained on the surface of the filter bed, where they in no way interfere with perfect filtration but, on the contrary, cause increased frictional

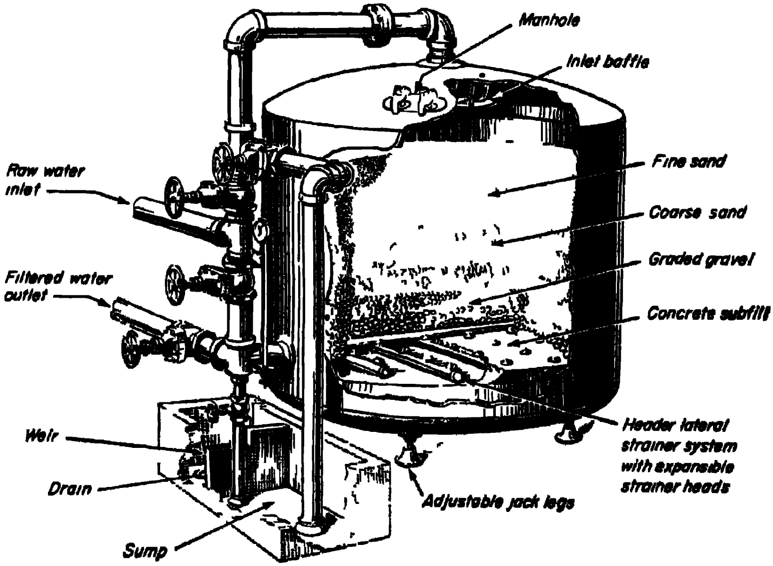


Fig. 7-8. Vertical pressure filter. (*The Permutit Co.*)

resistance and afford a finer screening than when the filter is first placed in service. In fact, a filter is at its best just as its capacity begins to decrease.

When the solids on the surface of the bed accumulate to such an extent that the

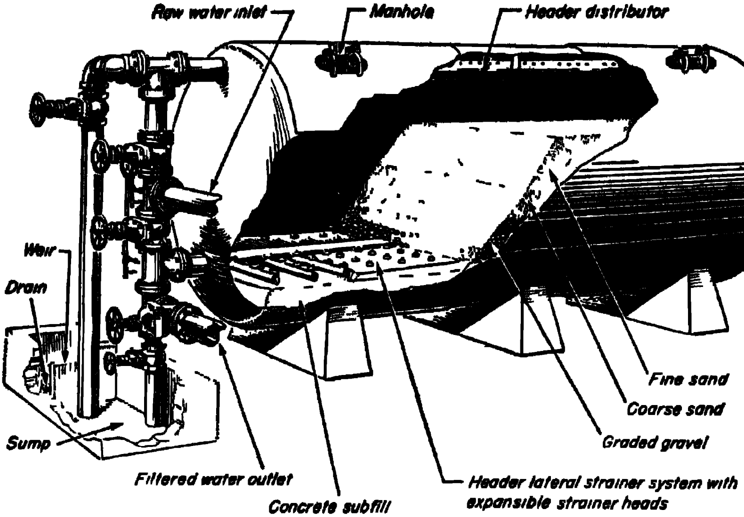


Fig 7-9 Horizontal pressure filter. (*The Permutit Co*)

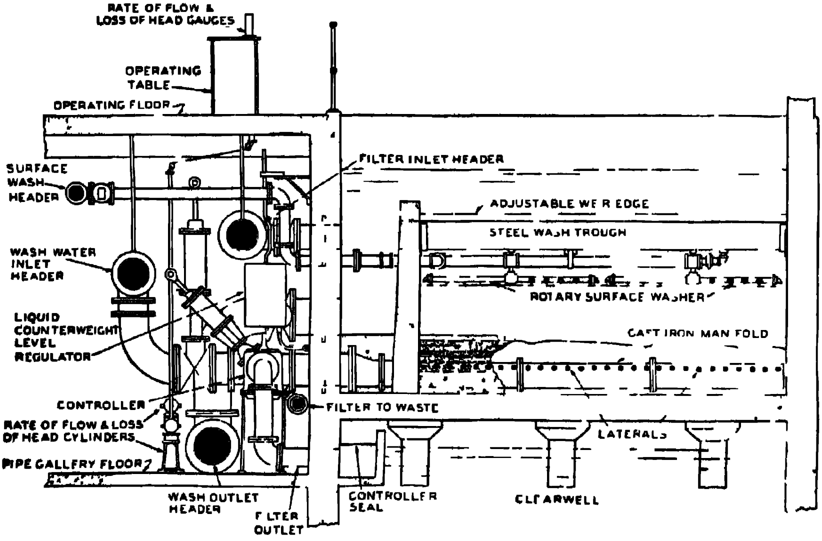


Fig 7-10. Cross section of concrete gravity filter (*The Permutit Co*)

frictional resistance through the bed becomes excessive (say, 5 lb per sq in.), it is generally found desirable to remove the retained solids by backwashing.

Backwashing. In backwashing a filter, the water is passed through the bed in a reversed direction at a rate four to seven times the filtration rate, causing boiling of the sand and carry-off of the suspended matter. In some cases the suspended matter is packed so densely that its removal becomes more easy when backwashing with water is preceded by air agitation. The air bubbling up through the sand bed breaks through the matted surface, which allows easier backwashing with water.

Types of Filters. Filters are of two general designs—the pressure and the open gravity type. For large installations, *e.g.*, textile-plant supplies or municipalities, the gravity type of filter is simpler than a large number of pressure-type filters. The pressure type of filter consists of a tank of either vertical or horizontal construction. Within the tank are placed the essential parts of the filter, such as the inlet manifold, the sand and gravel bed, and the strainer mechanism for collecting the filtered water.

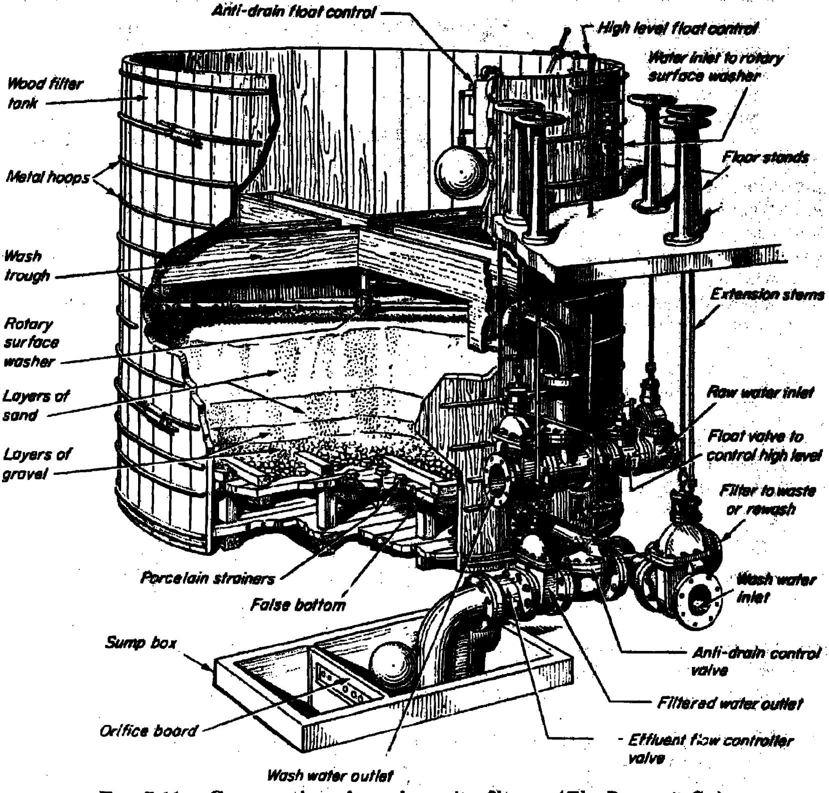


FIG. 7-11. Cross section of wood gravity filter. (*The Permutit Co.*)

The gravity filter covers a greater area than is possible with the pressure type of filter, which is usually rectangular and operated by means of hydraulic valves. One gravity filter may have as much filtering capacity as a dozen pressure filters. It is therefore obvious that for large capacities the control of the gravity installations is very much simplified, whereas for small installations, under certain conditions, the pressure filter offers an advantage. Figures 7-8 and 7-9 show vertical and horizontal pressure filters, and Figs. 7-10 and 7-11 show concrete and wood gravity filters.

The chief distinction between pressure filters and gravity filters is that the former have a closed container designed to withstand the working pressure so that, ordinarily, the use of pressure filters does not require additional pumping. Gravity filters have open containers which make necessary additional pumping unless the water can flow by gravity from a settling basin to the filter, in which case the service pump picks up the filtered water from a clear-well chamber and pumps it to the storage tank or to the various points of use. If it is necessary to pump the water to a filter, it is generally advisable to install a pressure filter, for under these conditions the use of a gravity filter would make it necessary to pump the water twice.

The capacity of a filter is dependent upon the character of the water to be filtered and, in general, lies between 2 and 4 gpm for each square foot of filter-bed area. The more suspended solids in the water, the lower should be the rate of filtration.

Although we consider filtration as meaning a straining action through sand and gravel, recent developments have shown that activated-carbon filters are very efficient in the removal of odors and taste. While this may not be classed strictly as a filtration problem, the units are usually designed along the same lines as filters, including water passage. Therefore, it is pertinent to place this type of equipment in the filter classification.

For the destruction of odor and taste, numerous methods have been employed, such as aeration, excessive chlorination, and introduction of permanganate of potassium.

Where excessive quantities of chlorine are introduced to bring about the destruction of contaminating material, which causes bad taste, the quantity of chlorine is so high that in itself it is objectionable. The removal of this excess chlorine is brought about very efficiently by the passage of the water through a bed of activated carbon. The carbon, of course, acts in an entirely different manner from filtering material, it being an absorbent rather than a strainer, and in the case of chlorine it is believed that the action of chlorine on carbon is purely chemical rather than mechanical. The absorption of other gases, however, may be mechanical, and after the activated carbon becomes saturated, it is necessary to renew the bed.

SOFTENING

In the industrial plant the type of equipment to install presents a problem in itself, depending on the requirements. The industrial softening plant may be divided into several different phases:

1. For industrial processes such as in textile mills
2. For cooling purposes such as engine jackets, spray ponds, etc.
3. For boiler-feed purposes

Textile Mills

In the preparation of water for textile mills, it is essential and highly desirable to remove, as completely as possible, the calcium and magnesium salts. In most textile operations a tremendous amount of soap is used. This soap precipitates any calcium and magnesium and forms a curd which is likely to adhere to the strands of threads and reflect on the quality of the material produced. The absence of calcium and magnesium in such waters not only brings about a better product but also, at the same time, decreases the soap bill considerably. For this purpose, large quantities of cold water are necessary, and the most efficient method of softening under these conditions is the zeolite system. There are many installations now in use where enormous quantities of water are softened by this process.

The most important application of the zeolite softening systems is found in the preparation of industrial waters used in processes where soft water is required, although such systems are used for boiler feed. However, in the selection of a softener for cold industrial water, reduction of hardness and cost of treatment are the main factors to be taken into consideration.

Zeolite Softener. Zeolite softening in its simplest form consists of a tank containing grains of fine bimetal silicate which has the unique property of removing calcium and magnesium from the water. The grains of silicate differ in chemical composition, method of preparation, and amount of softening that may be accomplished by a given volume of the material. The chemical principles involved and the degree of softening obtainable are much the same in all cases. For waters of low hardness it is sometimes desirable to use a natural zeolite of an exchange value of approximately 2,700 grains per cubic foot. For waters very high in hardness and where it is desirable to use small containers, a zeolite of higher exchange value, approximately 7,500 grains per cubic foot, may be used. The degree of softening accomplished by either mineral is essentially the same.

The exchange value of the zeolite and the depth of bed determine the capacity obtainable with any given diameter of container. For convenience, the exchange value of a zeolite is generally based on the cubic foot as a unit. The pound is not so good a unit because the same zeolite may not always contain the same amount of moisture.

The exchange value of a zeolite is measured by the amount of hardness that it will remove between successive regenerations with salt while delivering water of zero hardness by the soap test. This is dependent to some extent upon the rate at which the water is passed through the zeolite bed, although, generally speaking, the rate is not a limiting factor but ordinarily can exceed the actual requirements without lowering the exchange value.

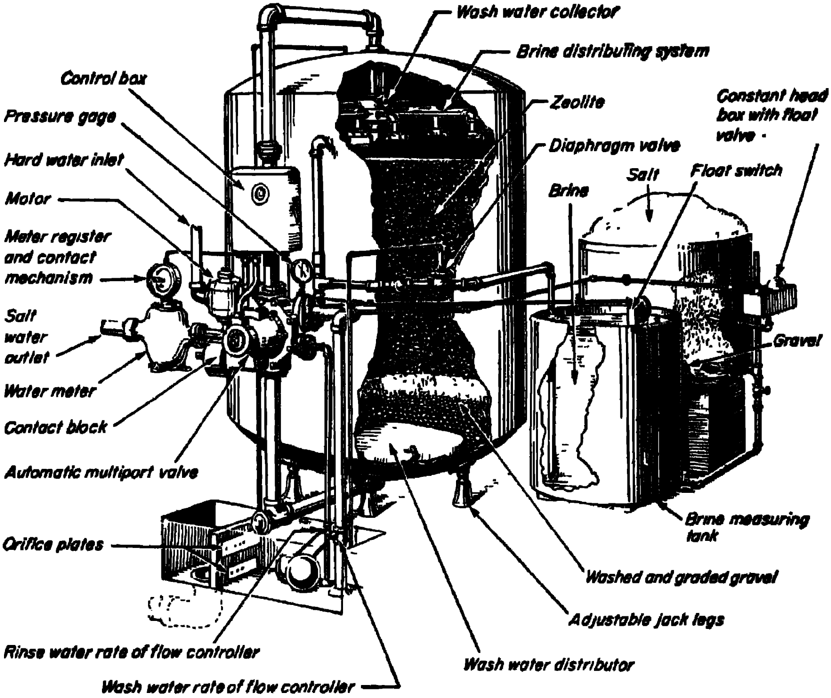


Fig. 7-12. Fully automatic zeolite softener. (*The Permutit Co.*)

Zeolite removes hardness from water much as a sponge absorbs water. Until the pores of the sponge become filled no water will pass through it, but once the pores of the sponge are satisfied any additional water delivered to the sponge passes through or displaces an equal amount of water previously absorbed by the sponge.

Zeolite softeners are offered by a number of companies, among which are Infilco, Cochrane, and Permutit.

A typical automatic zeolite water softener is shown in Fig. 7-12.

Regeneration of Zeolites and Salt Requirements. In passing water through a bed of sodium zeolite, the calcium and magnesium dissolved in the water are absorbed by the zeolite, which gives up a corresponding amount of sodium. After a definite amount of water has passed through the zeolite, so much of the sodium of the bed has been replaced by calcium and magnesium that the zeolite loses its exchange power, and the water is no longer softened. It then becomes necessary to regenerate the bed, *i.e.*, to replace the calcium and magnesium in the zeolite by sodium. This is done by irrigating the zeolite with a strong brine (sodium chloride) solution. A reverse reaction then takes place in that the calcium and magnesium are exchanged for sodium. Whether calcium and magnesium go from the water to the zeolite or from the zeolite to the water depends on the relative proportions of sodium or of calcium and magnesium in the water and in the filter bed. The reaction is, however, so complete that water passing through a regenerated bed of zeolite shows zero hardness as measured by the soap test. As soon as the water leaving the bed contains sufficient hardness to be detected by the soap test, the bed should be regenerated.

The amount of salt required for regeneration is 3½ grains of salt for every grain of hardness in terms of calcium carbonate removed, which is equivalent to ½ lb salt per thousand gallons of water per degree of hardness. To illustrate: if a zeolite softener has the capacity for softening 20,000 gal of 10-deg-hardness water between regenerations, each regeneration will require 100 lb salt.

Since a saturated salt solution contains a definite amount of salt, varying but slightly for ordinary differences in room temperature, and as the water becomes saturated with salt as it passes through the salt in the brine tank, it is necessary only to feed a predetermined depth of salt solution at each regeneration.

Cooling Supplies

The zeolite softener has other applications besides the preparation of process water, such as preparation of water for cooling purposes in the jackets of diesel engines, in the softening of make-up water for cooling ponds, and in the preparation of water for boiler feed and evaporators. In the preparation of water for these latter services, other factors must be taken into consideration besides the degree of softening, for such water, either as leakage or as make-up, finds its way to the boiler, and its preparation must be considered in the same light as would be the preparation of boiler-feed water. The cooling tower and spray pond concentrates water in much the same manner as does a boiler, and wherever condensers are in use with a high-carbonate water, deposits are experienced that greatly interfere with the vacuum to be obtained in the condenser, and, consequently, the operating efficiency of turbo-generator units.

In the treatment of this water the complete softening of the water is not so essential as is the preparation of water for use in textile mills. It is necessary only to reduce the carbonate content to a point sufficiently low so that it does not cause deposits in condensing equipment. For this purpose a number of methods have been proposed. One, as mentioned above, is treatment with a zeolite water softener where the entire hardness must be removed. The second method consists of treating the water with a cold-process water softener, using lime and soda ash as the treating agents. In most cases, however, lime alone is all that is required. This is usually accompanied by a coagulant such as aluminum sulfate, ferrous sulfate, or possibly sodium aluminate. The precipitation of the carbonate hardness to several grains per gallon is all that is required for ample protection. In some cases soda-ash treatment is added to the lime treatment for the precipitation of the permanent hardness, in which case a portion of the water may be used for boiler feed.

Boiler-feed Water

Perhaps the most intricate problem in the selection of water-treating apparatus is in the preparation of water for boiler-feed purposes, where consideration must be taken not only of the degree of softening that may be obtained but of many other factors as well. For the softening of boiler-feed water there is a rather wide selection of treatments and combination of treatments that may be used. These treatments may be divided into two general classes—internal and external. The internal treatments include compounds, soda-ash treatments, colloidal treatments, and various other chemicals that are placed on the market under a definite trade name.

The second method includes lime and soda softeners, cold and hot process, lime and barium softeners, and zeolite softeners.

Internal Treatment. The application of internal treatment without knowledge of the condition of the boiler water, such as the use of a boiler compound, is gambling on whether or not the right dosage is applied. Usually there will be a certain amount of success achieved with this method, depending upon how nearly correct the proper treatment was guessed. Table 7-3, taken from the "Prime Movers Committee Report of the National Electric Light Association," reports 19 boiler compounds of varying compositions. The relative price of soda ash, on the basis of water-softening ability, shows a wide fluctuation as compared to soda ash when purchased on the open market. It is therefore clear why boiler compounds have fallen into disrepute and why a fair success can be achieved with internal treatment by using commercial chemicals such as soda ash, caustic soda, or any of the sodium phosphates.

Internal treatment of water has a certain field, and in some cases, satisfactory results can be obtained. In localities of soft water, where the degree of hardness is low, internal treatment is quite satisfactory. It must be remembered that with internal treatment there is no removal of solids from the raw water supply, and if internal treatment is used on very hard waters where the make-up is high, it is usually followed by a condition of priming or scale formation which requires softening equipment external to the boiler for its correction.

Table 7-3. Boiler Compounds

Boiler compound number	Soda content based on soda ash, per cent	Pounds compound equal to 100 lb soda ash	Costs compared
1	14 0	700	$ 70 00
2	26 6	368	36 80
3	38 8	253	25.30
4	5 4	1,810	181.00
5	26 1	375	37.50
6	35 5	277	27.70
7	Contains no soda		
8	5 3	1,840	184 00
9	13 0	755	75 50
10	1 4	7,000	700 00
11	16 0	610	61 00
12	65 7	150	15 00
13	26 1	375	37 50
14	19 2	510	51 00
15	16 5	610	61 00
16* soda ash	98 0	100†	2 50
17*	99 0	94†	2 35
18*	44 0	455†	11 37
19*	24 6	528†	13 20

* Commercial chemicals.

† Weight based on market value and per cent purity.

External Treatment. The external softening of water for boiler-feed use may be accomplished in a number of different ways, among the most common being

1. Intermittent—using lime and soda ash
2. Continuous softening—either cold or hot process, using lime and soda ash and/or sodium phosphate
3. Zeolite

Intermittent Softeners. Intermittent softeners are operated on the fill-and-draw plan, the system consisting of two or more tanks, one supplying soft water which has been previously treated, while the others are being filled and treated with the softening reagents. The units are operated generally on a definite cycle, which consists of filling the tank with raw water, adding the entire amounts of chemical in proportion to the amount of carbonate and noncarbonate hardness present in the water, agitating the water to ensure thorough admixture and permitting the insoluble salts to precipitate while the water remains quiet. After a sufficient time has elapsed for subsidence of the precipitate, the soft water is drawn from the tank. The same procedure is followed with the other tank or tanks.

The disadvantage of this method of softening is the rate of reaction. Also the settling of the precipitates is much slower in cold water than in warm or hot water, and degree of hardness removal is limited.

Continuous Softeners. Continuous softening, as the term indicates, softens the water while it flows through the apparatus. The essential features of this type of softener are as follows:

1. The mixing device for the preparation of the chemical
2. The chemical-dosing apparatus for proportioning the chemicals used
3. The reaction or mixing chamber

4. The settling compartment
5. The filter

The disadvantages of cold continuous-process softening are much the same as those of the intermittent process, although the space requirements per unit volume of water are much less with the continuous process.

Hot-process Chemical Softeners. Hot-process chemical softeners are essentially the same as cold softeners of the continuous type but are distinct from other precipitation softeners in that the softening process takes place in a hot solution. The apparatus is a combination of an open feed-water heater and softener in which the scale-forming solids are removed in part by the effect of heat and as the result of chemical reaction.

The speed of chemical reactions is more rapid in hot than in cold solution, and the insoluble compounds formed settle more rapidly and are more completely removed than in cold water. The rate of chemical reaction is shown graphically in Fig. 7-13.

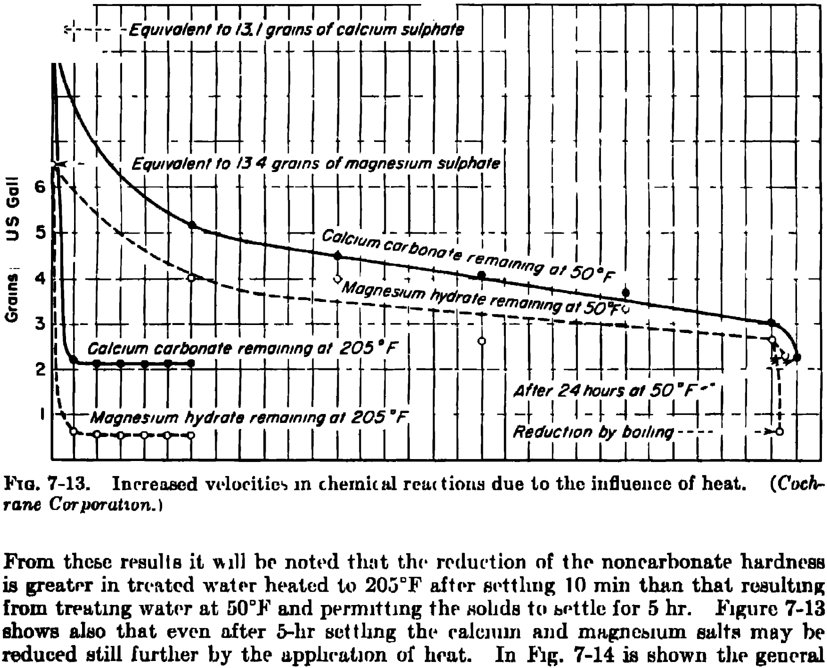


FIG. 7-13. Increased velocities in chemical reactions due to the influence of heat. (*Cochrane Corporation.*)

From these results it will be noted that the reduction of the noncarbonate hardness is greater in treated water heated to 205°F after settling 10 min than that resulting from treating water at 50°F and permitting the solids to settle for 5 hr. Figure 7-13 shows also that even after 5-hr settling the calcium and magnesium salts may be reduced still further by the application of heat. In Fig. 7-14 is shown the general arrangement of a hot-process softener which has been used to treat a large number of feed-water supplies.

A number of hot-process softeners have been placed upon the market within the past few years. They differ from one another primarily in the construction details and in respect to the method of heating the water. Exhaust or live steam may be used, and while exhaust steam is used in most installations the heaters are designed to function under a few pounds of back pressure.

Zeolite. The zeolite softener offers one of the simplest methods for removing hardness from a water supply. Until within the past few years this method of softening was not well adapted for treating all types of water for boiler-feed purposes as none of the solids were removed from the water but were converted to the corresponding soluble sodium salts. The hypothetical reactions which take place in softening water by zeolites may be expressed as follows:

Na_2 zeolite Sodium zeolite	+	$CaCO_3$ Calcium carbonate	Na_2CO_3 Sodium carbonate	+	Ca zeolite Calcium zeolite	(1)
Na_2 zeolite Sodium zeolite	+	$Ca(HCO_3)_2$ Calcium bicarbonate	$NaHCO_3$ Sodium bicarbonate	+	Ca zeolite Calcium zeolite	(2)
Na_2 zeolite Sodium zeolite	+	$CaSO_4$ Calcium sulfate	Na_2SO_4 Sodium sulfate	+	Ca zeolite Calcium zeolite	(3)
Na_2 zeolite Sodium zeolite	+	$Mg(HCO_3)_2$ Magnesium bicarbonate	$NaHCO_3$ Sodium bicarbonate	+	Mg zeolite Magnesium zeolite	(4)
Na_2 zeolite Sodium zeolite	+	$MgSO_4$ Magnesium sulfate	Na_2SO_4 Sodium sulfate	+	Mg zeolite Magnesium zeolite	(5)

Thus it may be seen that all the carbonates and bicarbonates are converted to soluble solids while with the hot-process softener these materials are largely precipitated as was shown in Fig 7-13 If water containing 10 grains per gallon of calcium

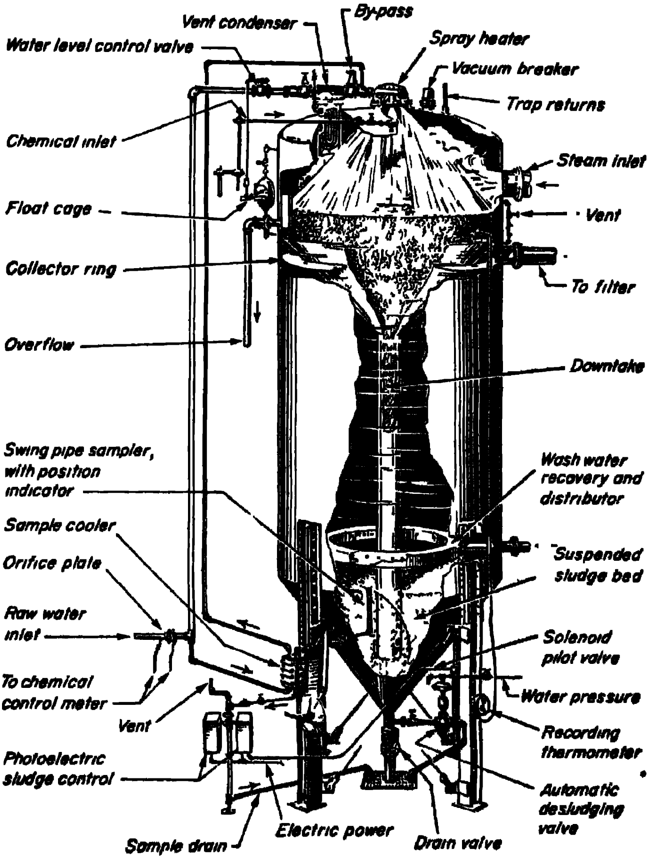


Fig. 7-14. Hot-process chemical softener (*The Permutit Co.*)

carbonate is passed through a zeolite softener, it would be converted to 10.6 grains per gallon of sodium carbonate.

The merit of the zeolite process lies in the fact that the sodium salts do not precipitate to form a scale as would the calcium and magnesium salts. Because of this conversion or exchange process the zeolite softener is not satisfactory for the preparation of all water for boiler feed. Thus the selection of the softening process depends upon the constituents of the raw water.

Degree of Remnant Hardness. The degree of hardness of the treated water is important up to a certain point because it is desirable to keep the boiler water fairly free of suspended matter due to the concentration and consequent precipitation of the hardness remaining in the water after treatment. With any system of precipitating hardness by chemical treatment, such as the cold or hot process, the reduction in hardness depends upon the excess of chemicals used. With a hot-process softener, for example, the use of 1½ grains per gallon excess of sodium carbonate reduced the hardness to 1½ grains per gallon. By increasing the excess of chemical by 1.0 grain, the hardness is reduced by ½ grain. With a cold-process softener, the hardness is not removed so completely, for heat is an accelerator of chemical reactions. While in cold water the hardness is not so completely removed as in hot water, the rule of excess chemical still holds.

With the zeolite, or exchange, type of softener, the reduction in hardness depends upon a number of factors, such as the amount of hardness initially present in the water and the amount of sodium salts initially present.

The hardness of the effluent water may be greater where the raw water is high in sodium salts or hardness than with low-sodium salts or hardness.

Carbonaceous Zeolites. Within the past few years new types of zeolite have become available, among which the most common is carbonaceous zeolite. This material is blackish in color and of granular form, the size of which is so regulated that it will pass a 10-mesh sieve and have definite limits in the amount that will pass a 50- and 60-mesh sieve.

Carbonaceous zeolites can be prepared by any one of a number of treatments of carbonaceous materials such as coal, lignite, or wood. The raw material properly graded is treated for extended periods with any one of a variety of chemicals such as fuming sulfuric acid, sulfur trioxide, chlorosulfonic acid, or zinc chloride, after which it is washed, stabilized, and finally screened to the desired size. Whether or not the raw material has any exchange power, this chemical treatment imparts to it a substantially increased ion-exchange power. In view of this exchange power, this material is classed as a zeolite and, because of its carbonaceous origin, as carbonaceous zeolite.

Carbonaceous zeolite functions in the normal sodium zeolite cycle when it is regenerated with sodium chloride in the usual manner. Because of its unusual resistance to acids, it can also be regenerated with acid solutions such as sulfuric or hydrochloric, in which case it operates in what is termed the hydrogen zeolite cycle, the hydrogen of the acid entering the zeolite and being later replaced by the cations calcium, magnesium, or sodium present in the water to be treated. In addition, the material possesses high operating-exchange value in the order of 5,000 to 12,000 grains of cations (expressed as $CaCO_3$) removed between regenerations per cubic foot of zeolite.

Regeneration. Carbonaceous zeolites when functioning in the sodium (Na_2Z) cycle operate on the same principle as the ordinary sodium zeolite.

For Na_2Z softening

$$(\text{Ca and Mg) salts} + Na_2Z = (\text{Ca and Mg})Z + \text{Na salts}$$

For Na_2Z regeneration

$$(Ca + Mg)Z + 2NaCl = Na_2Z + (\text{Ca and Mg) chlorides}$$

In the hydrogen cycle (H_2Z) the reactions and results differ, as may be noted in the following reaction. Since the cations Ca, Mg, and Na occur in water principally as bicarbonates, sulfates, and chlorides, the various reactions that take place in the hydrogen zeolite cycle are as follows:

For H_2Z softening

$$\left\{\begin{matrix}Ca\\Mg\\Na_2\end{matrix}\right\}(HCO_3)_2 + H_2Z = \left\{\begin{matrix}Ca\\Mg\\Na_2\end{matrix}\right\}Z + 2H_2CO_3$$

$$\left\{\begin{matrix}Ca\\Mg\\Na_2\end{matrix}\right\}SO_4 + H_2Z = \left\{\begin{matrix}Ca\\Mg\\Na_2\end{matrix}\right\}Z + H_2SO_4$$

$$\left\{\begin{matrix}Ca\\Mg\\Na_2\end{matrix}\right\}Cl_2 + H_2Z = \left\{\begin{matrix}Ca\\Mg\\Na_2\end{matrix}\right\}Z + 2HCl$$

For H_2Z regeneration

$$\left\{\begin{matrix}Ca\\Mg\\Na_2\end{matrix}\right\}Z + H_2SO_4 = H_2Z + \left\{\begin{matrix}Ca\\Mg\\Na_2\end{matrix}\right\}SO_4$$

From the above reaction it is noted that in the H_2Z cycle all bicarbonates are converted to H_2CO_3 which decomposes into water and carbon dioxide, thereby reducing the dissolved solids to the extent of the bicarbonate or temporary hardness in water. Where carbonaceous zeolite is used for softening on the hydrogen cycle, the resultant sulfuric or hydrochloric acid is neutralized by mixing water that is treated by the sodium cycle which operates in parallel with the H_2Z units or by the addition of suitable alkali such as caustic soda, soda ash, or phosphates.

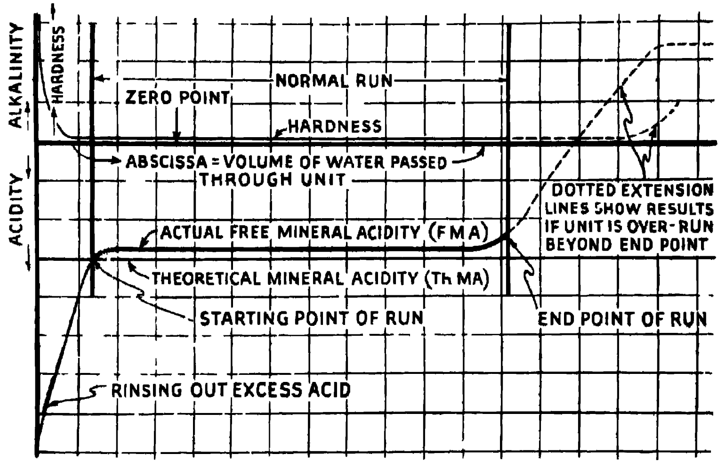


FIG. 7-15. Characteristic curve for carbonaceous-zeolite effluent. (*The Permutit Co.*)

Advantages of Carbonaceous Zeolites. The application of hydrogen zeolite requires study of each water supply to determine the most suitable and economic setup, whether the use of water is for process or boiler feed.

The use of hydrogen zeolite is the only known method, other than distillation, for removing sodium bicarbonate from solution. It likewise will remove hardness to zero and simultaneously reduces alkalinity to any desired value. This is shown in Fig. 7-15. Carbonaceous zeolite has a wide tolerance for acid water and will withstand the attack of acid for long periods.

A typical arrangement of a carbonaceous-zeolite unit operating on the H_2Z cycle is shown in Fig. 7-16.

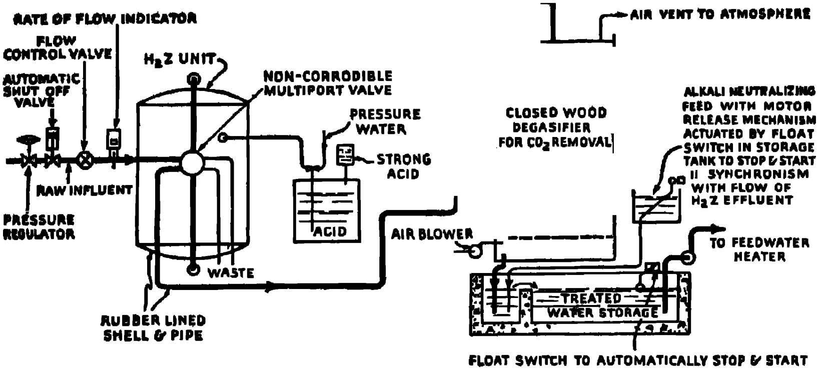
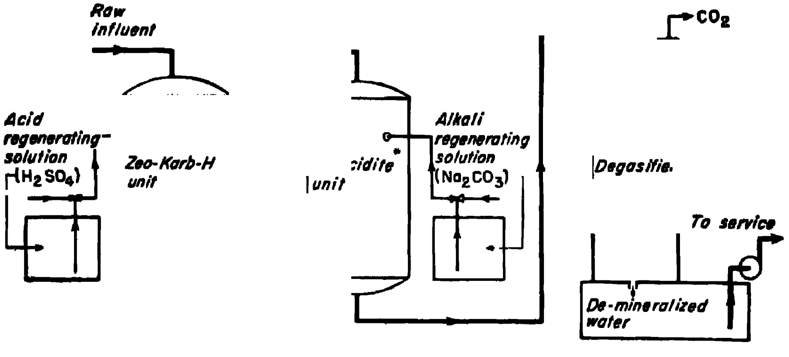

RATE OF FLOW INDICATOR
FLOW CONTROL VALVE
AUTOMATIC SHUT OFF VALVE
RAW INFLUENT
PRESSURE REGULATOR
H_2Z UNIT
NON-CORRODIBLE MULTIPORT VALVE
PRESSURE WATER
STRONG ACID
ACID
WASTE
RUBBER LINED SHELL & PIPE
AIR BLOWER
CLOSED WOOD DEGASIFIER FOR CO_2 REMOVAL
AIR VENT TO ATMOSPHERE
ALKALI NEUTRALIZING FEED WITH MOTOR RELEASE MECHANISM ACTUATED BY FLOAT SWITCH IN STORAGE TANK TO STOP & START II SYNCHRONISM WITH FLOW OF H_2Z EFFLUENT
TO FEEDWATER HEATER
TREATED WATER STORAGE
FLOAT SWITCH TO AUTOMATICALLY STOP & START

FIG. 7-16. Arrangement of carbonaceous-zeolite unit with degasifier and neutralizing feed. (*The Permutit Co.*)

Demineralizing Process for Reducing Dissolved Solids in Water

Many industrial processes require water that is free from the various mineral salts ordinarily present in all natural supplies. This is the case in the processing or manufacture of certain cellulose products, plastics, fine chemicals, pharmaceuticals, photographic materials, and many other materials. In some cases, water free from mineral salts is required throughout all the wet processing. In other cases, it is required only for the final washing, purification, solution, or dilution of the product.



Step No. I
Replacement of metallic cations by hydrogen (H ion) converts salts present in raw water into corresponding acids, viz. HCl, H_2SO_4, and H_2CO_3

Step No. 2
Removal of acids formed in Step No. I. The HCl and H_2SO_4 are absorbed in the De-Acidite unit and the H_2CO_3 and CO_2, this CO_2 escaping to the atmosphere in the degasifier.

*Trademark reg. U.S. Pat. Off.

FIG. 7-17. Flow diagram of demineralizing process. (*The Permutit Co.*)

Until recently, distillation was the only known process for producing water of such a character. Distillation is effective but expensive. Distilled water costs from 60 cents to $8 per 1,000 gal, depending on the number of evaporator effects.

A simple and relatively inexpensive method known as "demineralization" is rapidly obtaining commercial significance. This is a method whereby the dissolved solids content of water may be reduced by chemical means rather than distillation. In this process, the clear, raw water is demineralized at ordinary temperatures by passing it in succession through two tanks of granular ion exchangers. The process is very similar in principle to zeolite softening and differs only in the chemicals required and the character of the exchange materials. This process is not applicable to highly saline waters such as sea water.

Operation of Demineralizing System. The removal of dissolved solids is accomplished in two steps. First, the cations such as calcium, magnesium, sodium, and potassium are exchanged for hydrogen ions by passing the water through a carbonaceous-zeolite unit operating on the hydrogen cycle. The reactions are those shown on page 316. This leaves the water acid to the extent that anions such as chloride, sulfate, and nitrate are present in the raw water. The second step which takes place in the anion-exchange unit removes the acid in the water from the cation unit.

The chemicals required for regeneration of the exchange materials are sulfuric acid for the cation-exchange unit and sodium carbonate for the anion-exchange unit.

Figure 7-17 is a flow diagram of the demineralization process.

Treatment Costs—Demineralization Process. The operating cost per 1,000 gal for demineralization will depend on the dissolved solids in the raw water and the market price of sulfuric acid and soda ash. Typical costs for various waters are shown in Table 7-4.

Table 7-4. Costs of Water

Water analysis	A	B	C	D	E	F	G
Hardness as ppm $CaCO_3$	43	61	116	120	212	261	302
Alkalinity as ppm $CaCO_3$	34	24	58	93	140	188	227
Sulfates as ppm SO_4	19	26	70	38	80	60	60
Chlorides as ppm Cl	4	8	2	33	24	10	25
Total salts as ppm $CaCO_3$	64	68	149	197	274	277	337
Regenerant costs/1,000 gal	$0.03	$0.05	$0.08	$0.10	$0.15	$0.12	$0.16

Composition of Effluents. In general, the composition of degasified effluents from demineralizing plants will be within the ranges shown in Table 7-5.

Table 7-5. Composition of Degasified Effluents

Total hardness as ppm $CaCO_3$	0–2
Methyl orange alkalinity as ppm $CaCO_3$	0–6
Chlorides as ppm $CaCO_3$	0–3
Sulfates as ppm $CaCO_3$	0–3
Free carbon dioxide as ppm CO_2	5–10
Color	Below 5

Analysis of demineralized waters from the plants as shown in Table 7-4 are shown in Table 7-6.

Table 7-6. Analysis of Demineralized Waters

Water analysis	A	B	C	D	E	F	G
Hardness as ppm $CaCO_3$	0	0	0	0	0	0	1
Alkalinity as ppm $CaCO_3$	3	2	3	4	6	2	4
Sulfates as ppm SO_4	0	0	0	0	1	0	1
Chlorides as ppm Cl	0	0	0	0	2	0	0
Total salts as ppm $CaCO_3$	3	2	3	4	9	2	6

AERATORS AND DEGASIFIERS

Aerators and degasifiers are used for the following purposes:

1. Removal of carbon dioxide and increase of pH value
2. Removal of hydrogen sulfide
3. Reduction of taste and odor
4. Oxidation of ferrous iron
5. Oxidation of manganous manganese

In the removal of carbon dioxide, the action is purely physical The introduced air dissolves in the water and, by reducing the partial pressure of the carbon dioxide, forces the gas out of solution

The removal of hydrogen sulfide by aeration is mostly mechanical as, at the temperatures involved, the oxidation of the hydrogen sulfide is so slow as to be almost negligible

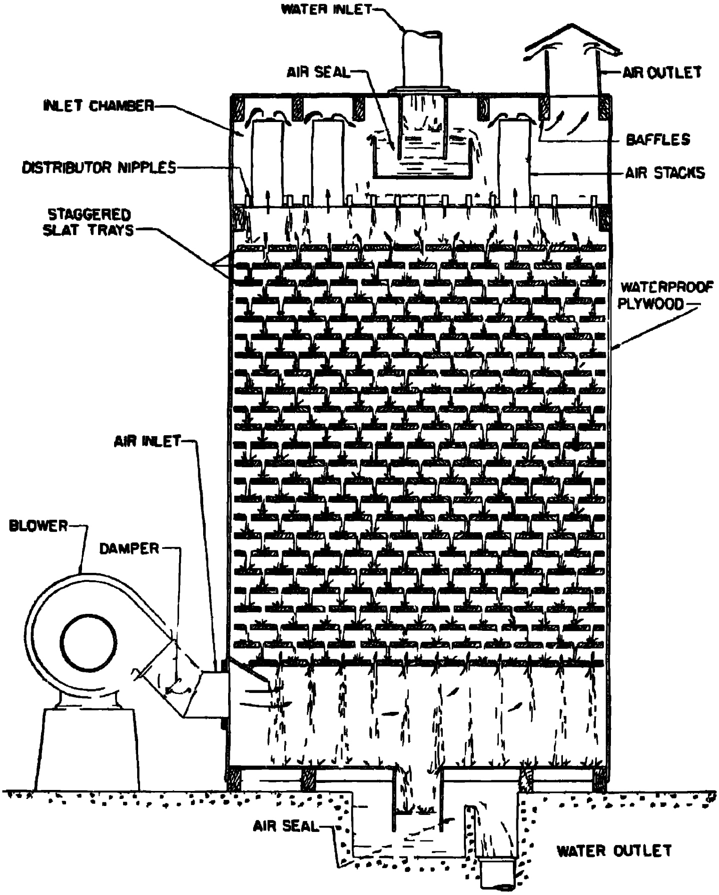


FIG 7-18 Section through degasifier (*The Permutit Co*)

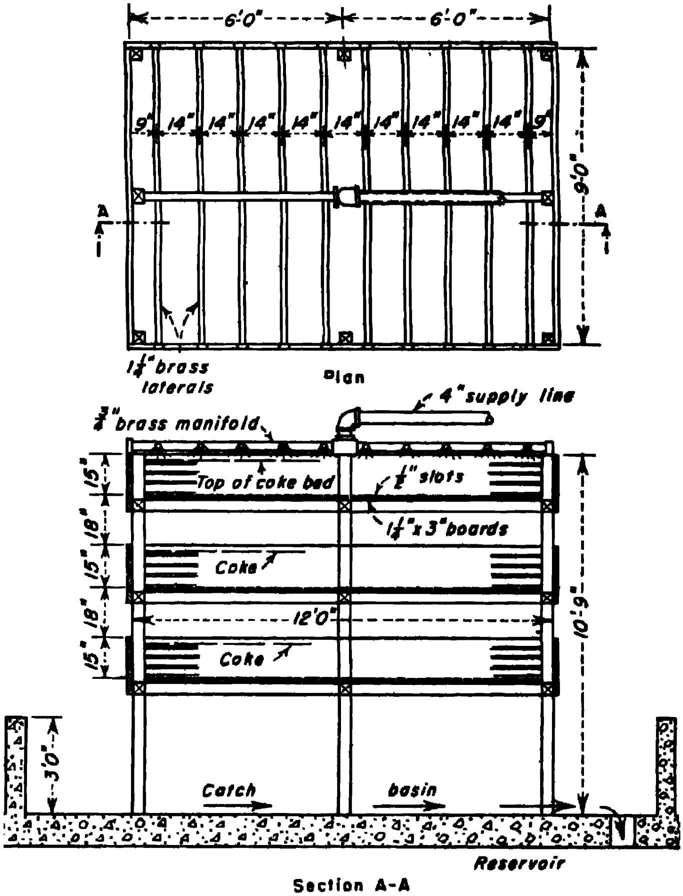
The reduction of taste and odor by aeration is largely mechanical as the objectionable gaseous or volatile bodies are removed in a manner similar to the removal of carbon dioxide, although part of this action may be attributable to oxidation

The oxidation of iron and manganese from soluble ferrous and manganous compounds to practically insoluble higher hydrated oxides is entirely chemical

In aeration, contact of water with air can be accomplished either by (1) introducing the air into the water or (2) introducing the water into the air

Introducing Air into Water. *Open-tank Aeration* Air may be introduced into water by blowing it through water contained in an open tank or by blowing the air into water under pressure in a closed tank which may or may not be vented Blowing the air through water in an open tank may be effected by air grids or porous plates, the source of the air being a blower or compressor. This method is quite often employed where acid is being fed to the water, in which case the air serves the double purpose of mixing the acid with the water and removing the carbon dioxide formed by the reaction.

Pressure Aeration. Pressure aeration is sometimes employed for iron removal where the amount of air to be introduced is not large and where double pumping is undesirable. As the solubility of air in water is directly proportional to the pressure,



Plan

Section A-A

FIG. 7-19. Coke aerator.

saturation of the water with air is often undesirable, since, when pressure is released as at a faucet or valve, a considerable volume of air is evolved from the solution and causes objectionable spattering. This can be avoided either by limiting the quantity of air introduced into all the water or by saturating with air only a portion of the water which is later mixed with the remaining quantity.

Introducing Water into Air. The second method of aeration whereby water is introduced into the air is the method most widely employed. Various devices of various efficiencies have been used to accomplish this. The most satisfactory device is the degasifier (see Fig. 7-18) in which the water rains down over a series of slot trays through a controlled countercurrent of air. The least efficient device is one in which a short, thick stream of water flows through the atmosphere. Other devices range between these extremes.

Coke-tray Aerators. There are two basic designs of equipment for the coke-tray method of aeration—the cylindrical type and the square or rectangular type. Both of these types are louvered structures containing three or four coke trays with air spaces between them. Figure 7-19 is a 300-gpm coke aerator. The entering raw water flows into a distributing tray which causes the water to flow evenly over the entire area and then permits it to drop through an air space to the surface of the first layer of coke. The water percolates through the porous coke and then drops from the bottom of this first tray through an air space to the surface of the second tray. This operation is repeated through the series of trays, and from the bottom of the last tray in the series the water rains down through an air space into a catch basin. Catch basins vary in size depending largely upon the composition of the water and the purpose for which it is being aerated.

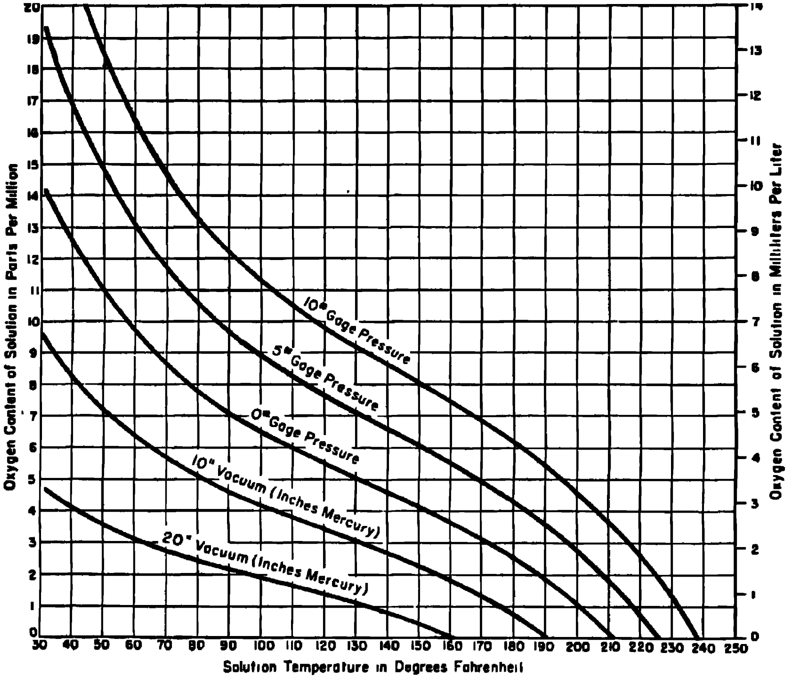


Fig. 7-20. Solubility of oxygen in distilled water.

Spray Aerators. Spray aeration is more frequently used in municipal water-works practice than in industrial-water treatment for removal of carbon dioxide and hydrogen sulfide and reduction of taste and odor. A closed-tank aerator similar to that shown in Fig. 7-18 is more applicable to industrial practice where ground area is limited or where location indoors is desired.

A spray aerator is similar in design to a spray pond, *i.e.*, the water is broken into fine droplets as the water discharges from the spray head and comes in contact with the atmosphere. The loading and space requirements per spray head depend upon the angle of the cone and volume of water handled, the required satisfactory operating conditions being specified by the spray-head manufacturer. A collecting basin is necessary where water is first aerated and then treated with coagulating chemicals, which makes for increased cost over that required with a closed-tank aerator.

Either system of aeration will, for all practical purposes, saturate the water with oxygen at the existing temperature, but in most cases the closed-tank aerator will show the lower first cost.

Corrosion. The utilization of large quantities of raw water in process cooling systems such as through condensers and heat exchangers has brought about a demand for cold-water deaeration to arrest corrosion of these equipments without waste of heat. In many process industries the heat pickup through such equipment by the cooling water is taken to the power plant where this water is a source of make-up. It is found, however, that with the ordinary water supplies, corrosion does occur, and in order to protect these units, cold-water deaeration has become quite prominent the past several years.

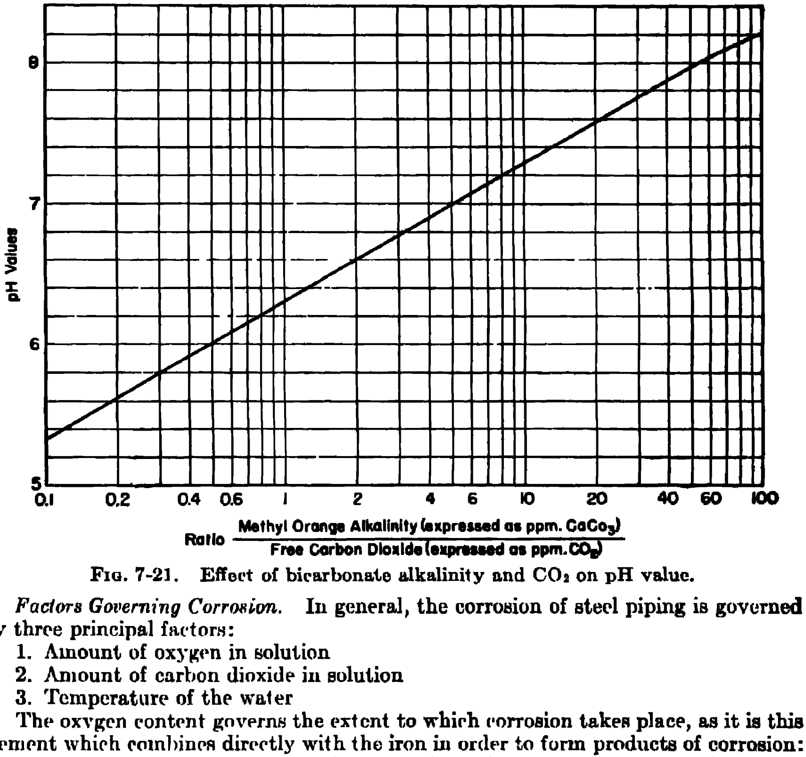


FIG. 7-21. Effect of bicarbonate alkalinity and CO_2 on pH value.

Factors Governing Corrosion. In general, the corrosion of steel piping is governed by three principal factors:

1. Amount of oxygen in solution
2. Amount of carbon dioxide in solution
3. Temperature of the water

The oxygen content governs the extent to which corrosion takes place, as it is this element which combines directly with the iron in order to form products of corrosion: iron rust, such as ferrous oxide, FeO; magnetic oxide, Fe_3O_4; or ferric oxide, Fe_2O_3, the type of product depending upon the concentration of oxygen present at the time rusting takes place.

The solubility of oxygen in distilled water at various temperatures and pressures is shown in Fig. 7-20.

Carbon dioxide in conjunction with the natural alkalinity of the water governs the pH value, which is a means of expressing the active acidity or alkalinity of the water. Figure 7-21 shows the effect of bicarbonate alkalinity and CO_2 on pH value, and Table 7-7 shows the relation of hydrogen-ion concentration and pH value.

The pH value influences the rate at which corrosion proceeds; the lower the pH value, the more acid the water and the greater the intensity of the corrosion; conversely, with higher pH values the corrosion activity becomes less, and in some cases where the temperature is not increased appreciably, corrosion can be arrested merely by adding alkali and raising the pH value.

The temperature of the water is governed by the process requirements and is fixed by the design of heat exchangers and the rate of heat transfer. It therefore becomes impractical to attempt to govern the rate of corrosion by control of the temperature, although it is well known that the corrosion of steel increases rapidly with increase in water temperature when the water contains quantities of oxygen and carbon dioxide in solution.

Table 7-7. Relation of Hydrogen-ion Concentration and pH Value

Hydrogen-ion concentration, per l (1)	Column 1 expressed as a fraction (2)	Reciprocal of Col. 2 (3)	Log of Col. 3 (4)	pH (5)	Acidity or alkalinity of solution relative to a neutral solution (6)	
1.0	1	1	0	0	10,000,000	↑
0.1	$1/10^1$	10^1	1	1	1,000,000	
0.01	$1/10^2$	10^2	2	2	100,000	
0.001	$1/10^3$	10^3	3	3	10,000	Acidity
0.0001	$1/10^4$	10^4	4	4	1,000	
0.00001	$1/10^5$	10^5	5	5	100	
0.000001	$1/10^6$	10^6	6	6	10	
0.0000001	$1/10^7$	10^7	7	7	1 ← Neutral	
0.00000001	$1/10^8$	10^8	8	8	10 (pure water)	
0.000000001	$1/10^9$	10^9	9	9	100	
0.0000000001	$1/10^{10}$	10^{10}	10	10	1,000	Alkalinity
0.00000000001	$1/10^{11}$	10^{11}	11	11	10,000	
0.000000000001	$1/10^{12}$	10^{12}	12	12	100,000	
0.0000000000001	$1/10^{13}$	10^{13}	13	13	1,000,000	
0.00000000000001	$1/10^{14}$	10^{14}	14	14	10,000,000	↓

Cold-water Deaerators. Cold-water deaerators are used to remove oxygen and carbon dioxide from solution and obtain a noncorrosive water, where it is desirable to accomplish this without chemicals and with a minimum amount of mechanical equipment.

The art of deaerating cold water is not new, as one of the first installations was placed in service on a large scale in 1917 for checking corrosion of a 30-in. steel pipe 350 miles long from Mundaring to Coolgardie in Western Australia. This installation was capable of deaerating 6 million gal of water per day and consisted of two units, each capable of treating 3 million gal of water in 24 hr. The water was deaerated by spraying it in a vacuum chamber.

Powell and Burns report on the effectiveness of a vacuum deaerator of the tray type in checking corrosion of a 9-mile pipe line conveying Mississippi River water to the Freeport Sulphur Company mining operations 50 miles south of New Orleans, at Grand Ecaille. This installation brought about 95 per cent reduction in the oxygen, the residual oxygen being 0.33 ppm. In order to have assurance that corrosion would be arrested, the residual oxygen was absorbed chemically by the proportioned introduction of sodium sulfite. It is interesting to note that this installation was very successful, whereas chemical means tried previously were not effective and were likewise costly.

General Design of Equipment. Cold-water deaerators, where heat is not applied, require

1. Maintenance of a pressure within the unit corresponding to the boiling point of the water under treatment
2. Means for dividing the water into small and minute particles in order to expose surface and to facilitate the escape of gases from solution while the optimum atmospheric conditions are maintained within the unit
3. Means for withdrawing the gases from the container after liberation and the maintenance of a constant vacuum condition

While the Coolgardie and Freeport installations were successful in achieving their objectives, the design of such equipment has advanced considerably, and it is now possible to reduce oxygen and carbon dioxide down to a fraction of a part per million, eliminating the necessity of supplementary chemical treatment for absorbing the last traces of oxygen.

In general, the equipment is arranged so that water is introduced into the vacuum chamber through a spray pipe from which it is discharged in fine particles over a column of staggered trays, generally constructed of wood. The water cascades down these trays which serve the purpose of distributing the water and breaking it into fine particles or films, permitting easy escape of the gases.

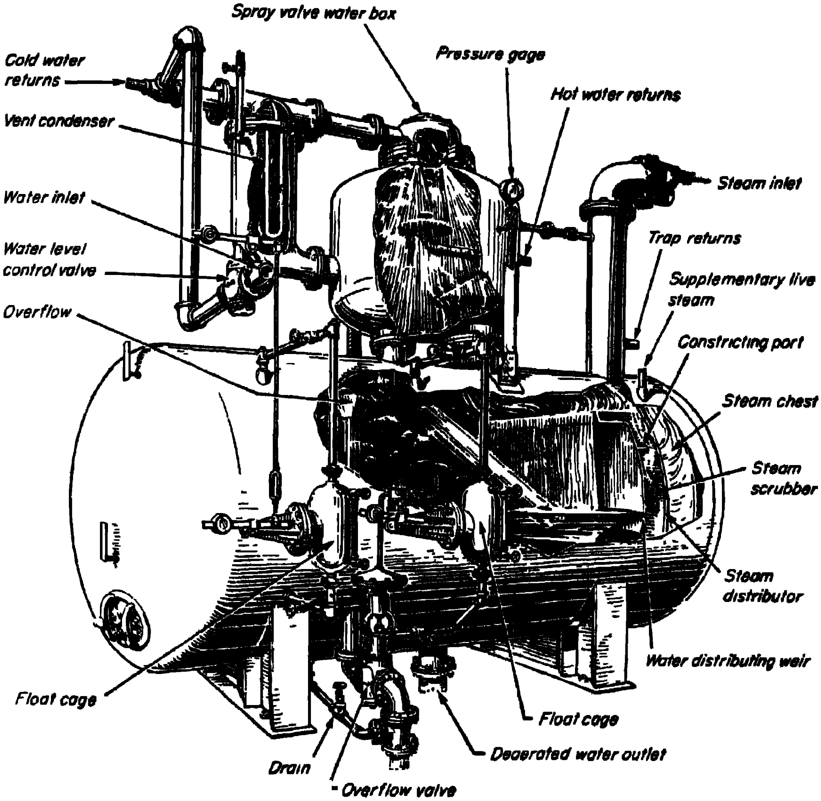


Fig. 7-22. Atomizing deaerating heater. (*The Permutit Co.*)

The noncondensable gases and a portion of the water vapor are withdrawn from the deaerator by means of vacuum-producing equipment which may take the form of steam-jet ejectors or mechanically operated vacuum pumps.

Hot-water Deaerators. In the deaeration of water for boiler-feed purposes or other uses, where hot water of a noncorrosive nature is desired, removal of oxygen and carbon dioxide is accomplished by heating the water to the boiling temperature corresponding to the pressure acting upon it. Various deaeration means for carrying out this heating and removal of the dissolved gases are employed, of which there are three basic methods, *viz.*, reboiling, flowing over trays or pans, and atomization.

In the reboiler deaerator, dissolved gases are removed by causing steam to bubble through the water, this steam being generated either by a closed heater element located in the storage space of the deaerator or by blowing the steam into the water through a multitude of small orifices.

In the second type of deaerator, the tray or pan type, the water is spread out over baffles or trays in thin films in such a manner as to provide intimate contact of the steam with the water, through agitation, and a definite path for the steam so as to remove the gases from the apparatus as rapidly as they are removed from the water.

The third type of deaerator utilizes the principle of atomization. The water is first sprayed into a steam atmosphere where it is heated to within two to three degrees of the steam temperature. This hot and partly deaerated water is then delivered into a high-velocity steam jet created by the flow of the incoming steam through a spring-loaded nozzle.

Figure 7-22 shows a deaerator of the atomizing type.

Scale-forming Waters. Water as used for cooling purposes may be classed as aggressive or nonaggressive in respect to the action on ferrous pipe Very few supplies are neutral, *i.e.*, neither corrosive nor scale forming Table 7-8 gives three water supplies used for cooling: source *A*, which is corrosive to steel, and *B* and *C*, which formed a coating of calcium carbonate in the lines.

Table 7-8. Sources of Water for Cooling

Source	A	B	C
Total solids	48	344	400
Suspended solids	2	9	0
Silica	2	20	10
Sulfates as SO_4	3	24	50
Chlorides as Cl	8	24	40
Carbon dioxide	27	29	30
Calcium as Ca	2		65
Magnesium as Mg	2	.	24
Iron and alumina	1	2	
Soap hardness as $CaCO_3$	12	256	240
Methyl orange alkalinity as $CaCO_3$	5	220	290
pH value	5.9	7.2	7.2

NOTE: All quantities as part per million except pH value.

Problems arise with either type of supply, corrosion with accompanying tuberculation increases the friction drop resulting in greater pumping costs and increased maintenance. A scale-forming water likewise causes greater pumping costs and stoppage of heat-transfer equipment, thus increasing maintenance.

Corrosion Prevention by Chemical Treatment. Deaeration of cold water in many cases cannot be justified, and it is necessary to resort to chemical treatments for decreasing corrosion. Among the most common of these treatments are sodium silicate, calcium hydrate-lime, and chromate compounds.

Sodium Silicate—Calcium Hydrate. With the use of sodium silicate or calcium hydrate, the effectiveness depends upon the promotion of inorganic protective films. Such films may consist of extraneous materials derived from salts dissolved in the water, such as carbonate or silicate of lime, or may be composed of an oxide produced by the attack upon the metal itself. In both cases the efficiency depends on their density and the bond they make with the metal. In this type of chemical treatment the film is derived either from the natural constituents of the water itself, the added chemicals merely changing the equilibriums so as to bring about deposition, or it is furnished by the chemicals as in the case of the silicate. Methods of treatment of this type have been successful in large-scale operations, such as city water supplies. For small-scale applications, however, they are less adaptable because the dosage and even the possibility of any benefits depend upon the composition of the local water and thus often present problems that demand local study and control. Too often, both in investigation and application, the uncompromising fact that water is anything but the same material anywhere has been forgotten. This simple fact cannot be given too much emphasis. The results of research or experience in corrosion by natural water are rarely of general application because they usually have to do with a particular water supply. Water from another source, or even the same water at another time, may not fit into the developed picture.

Chromate Compound In the second type of chemical treatment, in which the protective film is an oxide of the metal itself, the chemical character of the water also plays a part, but one more easily compensated for. The phenomenon in this type is essentially that of passivity. In true passivity, the oxide film is invisible. In fact, it is interesting to note that, as it has been pointed out, invisibility is a measure of effectiveness. To become thick enough to be visible a film must grow, and that means that some attack must occur through it. Furthermore, a thick film is likely to crack. The classical case of passivity is the effect produced on iron by immersion in concentrated nitric acid. The effect is not permanent when the environment is changed.

Fortunately, the soluble chromates are highly effective passivating agents which are not corrosive and consequently can be kept permanently in contact with the surfaces to be protected without danger to the equipment.

Chromates contain chromate anhydride (CrO_3) which becomes converted into inert chromic oxide (Cr_2O_3) by giving up half of its oxygen. It is thus a strong oxidizer. In this respect it resembles other passivating agents, producing a film of oxide of iron. But, again in common with other oxidants, it is also a cathodic depolarizer, thereby tending to promote corrosion as does dissolved oxygen. This latter tendency, however, is successfully combated by the character of its anodic action. The effectiveness of the protective oxide film there produced appears to be due not only to the simple production of a film of iron oxide but also to the remarkable density of the film arising from the effusion in it of a portion of chromic oxide produced by the primary anodic reaction and from the effect of the alkali produced in the immediate proximity by the cathodic action. This last promotes immediate dense precipitation on the anode of the solution which is first formed there, whereas in ordinary corrosion, this solution must diffuse for some distance before a sufficient amount of alkali is encountered to precipitate it, with a result of loss and nonprotective product. Of these factors, the inclusion of chromic oxide is probably the most important as the other actions are also common to other oxidizing agents which have no, or less, protective effect.

The dosage of chromate is important. Its effective action logically depends on its presence at the seat of the reaction in sufficient concentration to play the role just described. It has been indicated that where there is local stagnation of the water, pitting is probable. Where chromate is used it must be present in the circulating water in sufficient concentration to maintain its protective reactions in such places by sufficiently rapid diffusion. If the concentration is too low, the diffusion may be too slow, in which event pitting may actually be promoted by the chromate because of its depolarizing action in surrounding nonstagnant cathodic areas and its effect in increasing the electrolytic conductivity of the water.

Dosage, therefore, is largely dependent on the physical conditions of the equipment as it affects the development of stagnant points. Reentrant corners, crevices, and cracks are largely a matter of design. More important for equipment already in the field are existing rust, loose scale, and tubercles which act as traps for water, thus promoting pitting. When these are present it is dangerous to depend upon increased chromate dosage to offset their effect. They should be removed.

SECTION 8

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POWER-PLANT THERMODYNAMICS

BY H. L. SOLBERG

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AND C. L. BROWN

Assistant Professor in Mechanical Engineering, Purdue University

DEFINITIONS

Thermodynamics. Thermodynamics is the science of heat energy and its transformations. The thermodynamics of power plants is concerned with the study of heat energy derived from burning fuel and the conversion of this energy into mechanical or electrical work.

Energy. Energy has been defined as the capacity for doing work. Fuel has chemical energy which is released by burning and converted to heat energy. Part of this heat energy may be converted into mechanical work in accordance with the laws of thermodynamics. Before these laws may be considered, it is necessary to define certain physical quantities. Some of these are familiar, but in order to avoid possible confusion, all will be defined and the customary units will be specified.

Pressure. Pressure (p) is measured in pounds per square inch (psi) and may be expressed as gauge pressure (psig) or absolute pressure (psia). An ordinary pressure gauge measures pressures above atmospheric, and to get absolute pressure, the pressure of the atmosphere as measured by a barometer must be added to the gauge reading. Since atmospheric pressure is generally read in inches of mercury, this must be converted to psi by multiplying by the factor 0.491.

EXAMPLE 1: Find the absolute pressure of steam in a boiler if the gauge reads 250 psig and the barometric pressure is 28.5 in. of mercury.

SOLUTION:

$$p = 250 + (28.5 \times 0.491) = 264 \text{ psia}$$

Pressures below atmospheric pressure such as are found in condensers are usually measured in inches of mercury vacuum. The absolute pressure in the condenser may be found by subtracting the vacuum reading from the atmospheric pressure.

EXAMPLE 2: Find the absolute pressure in a condenser in which there is a 28-in. vacuum when the atmospheric pressure is 29.5 in. Hg.

SOLUTION:

$$p = 29.5 - 28 = 1.5 \text{ in. Hg abs}$$
$$p = 1.5 \times 0.491 = 0.74 \text{ psia}$$

Temperature. Temperature (t) is measured in the English system of units in degrees Fahrenheit. Sometimes it is necessary to use absolute temperature (T, degrees Rankine) which is measured above absolute zero, the lowest temperature which can exist. Absolute zero is about −460°F so that, if the temperature is known in degrees Fahrenheit, the absolute temperature is $T = 460 + t$. Thus 100°F is 560°R.

Heat. Heat is a form of energy which is transferred from one body to another because of a temperature difference. The addition of heat to a substance will produce either a rise in temperature or a change of state, as, for example, the evaporation of water. Heat, as well as other forms of energy, can be measured in British thermal units (Btu). One Btu is the amount of heat required to raise the temperature of 1 lb of water at atmospheric pressure 1°F.

Specific Heat. The specific heat of a material is the amount of heat required to raise the temperature of 1 lb of the substance 1°F. Thus, from the above, the specific heat of water at normal temperatures is unity. However, specific heats vary with temperature and pressure.

Work. Work is done when a force is exerted on a body, and the body moves in the direction of the force. The units of work are foot-pounds (ft-lb). For example, if a weight of 10 lb is lifted 5 ft, 50 ft-lb of work is done. Work can be converted into heat, and 778 ft-lb equal 1 Btu so that work can be expressed in Btu by dividing the work in foot-pounds by 778.

Power. Power is the rate of doing work. The two common units of power are horsepower (hp) and kilowatt (kw). These are defined as

$$1 \text{ hp} = 33{,}000 \text{ ft-lb per min} = 2545 \text{ Btu/hr}$$
$$1 \text{ kw} = 44{,}240 \text{ ft-lb per min} = 3413 \text{ Btu/hr}$$

A hoist lifting a weight of 33,000 lb a distance of 1 ft in 1 min develops 1 hp. Another hoist which lifts the same weight in 6 sec does the same work but does it ten times as fast and hence develops 10 hp.

Enthalpy. Enthalpy (*h*) is the term used for the energy content of water and steam. In steam tables and charts, enthalpy is measured in Btu per pound.

Entropy. Entropy is an important but rather mysterious quantity. It will be sufficient here to define entropy as a measure of energy made unavailable or wasted. During the expansion of steam in an engine or turbine, if the entropy is not changed, no energy is wasted. If the entropy increases, however, some energy is lost and cannot be converted into useful work. Thus, the most efficient expansion is one which occurs at constant entropy and is called a "constant-entropy" process.

Power Cycle. A power cycle is a group of processes which form a closed circuit, the purpose of which is the conversion of heat energy into mechanical or electrical work. For example, the simple steam cycle consists of (1) pumping water into the boiler, (2) evaporating the water and superheating the resulting steam, (3) expanding the steam in a turbine or engine, and (4) condensing the exhaust steam. The condensate then goes to the feed pump, and the cycle is repeated.

Cycle Efficiency. Cycle efficiency is, like all efficiencies, a ratio and may be defined as the ratio between the work produced in the turbine or engine to the heat supplied to the steam in the boiler.

Steam Rate. The steam rate of a prime mover is defined as the number of pounds of steam required at the throttle per hour to develop either 1 hp or 1 kw. Since steam plants usually generate electrical power, the steam rate will hereafter be given as pounds per kilowatthour.

CHARACTERISTICS OF WATER AND STEAM

When water is heated under constant pressure, its temperature increases until a point is reached at which it begins to boil. During evaporation, the temperature remains constant. At each pressure there is a definite boiling temperature which is called the "saturation temperature" and corresponds to the pressure. Water at saturation temperature and pressure is said to be saturated. For example, the saturation temperature at 14.7 psia is 212°F. Water at 14.7 psia and 70°F is not saturated and must be heated to 212°F before it becomes saturated.

Steam free of water at saturation temperature is called "saturated steam." As long as water is present, the addition of heat will not raise the temperature but will merely evaporate more water. If saturated steam is removed from contact with water and heat is added, the temperature will increase, and the steam becomes "superheated." The condition of superheated steam may be specified by giving the temperature and pressure. Sometimes steam is referred to as having a certain superheat. This is the difference between its temperature and the saturation temperature corresponding to the pressure. However, if the steam is at saturation temperature, it may be dry (free of liquid water) or contain any amount of water. To specify the condition of such wet steam it is necessary to give the pressure (or temperature) and the quality, which is defined as the ratio of the weight of dry steam to the weight of the mixture of steam and water.

Water may exist at pressures higher than that corresponding to its temperature as shown by the steam tables. Water under such conditions is said to be "compressed." For example, water at 212°F under a pressure of 200 psia is compressed water. A characteristic of compressed water is that the addition of heat will not cause evaporation but will merely cause a rise in temperature until the boiling point or saturation point corresponding to the elevated pressure is reached.

Use of Steam Tables and Mollier Chart. Values of the various properties of water and steam have been determined on the basis of experimental data. There are several sets of tables, but the latest and most widely used is ' Thermodynamic Properties of Steam" by Keenan and Keyes, published by John Wiley & Sons, Inc. All thermodynamic data used here will be taken from these tables.

The tables are divided into four sections.

Table 8-1 is a tabulation of the properties of saturated water and steam based on temperature. If the temperature is known, then from the table pressure, enthalpy, and entropy may be found for saturated water and saturated steam. For saturated water enthalpy and entropy are represented by the symbols h_f and s_f and for dry saturated steam by h_g and s_g, respectively. The increase in enthalpy during evaporation, which is merely the difference between h_g and h_f, is given in the table with the symbol h_{fg}. Likewise, the increase in entropy during evaporation is designated by the symbol s_{fg}.

Table 8-2 is laid out in the same fashion except that values are given for a series of pressures instead of temperatures. The first table is used to find the properties of feed water and for cases where the temperature is known. The second table is used when pressure is known.

When the quality of wet steam is known, the enthalpy and entropy can also be computed from the steam tables. A pound of wet steam with a quality x consists of x lb of saturated steam and $(1 - x)$ lb of saturated water. If the enthalpy of the water h_f and steam h_g are found from the tables, the enthalpy of the mixture can be found as

$$\begin{aligned} h &= xh_g + (1 - x)h_f \\ &= xh_g + h_f - xh_f \\ &= h_f + x(h_g - h_f) \\ &= h_f + xh_{fg} \qquad (1) \end{aligned}$$

where h_{fg} is $h_g - h_f$, or the difference in enthalpy between saturated steam and water at the same temperature. Entropy of wet steam can be calculated in the same way.

Example 3: Find the enthalpy of steam at 100 psia and 95 per cent quality.

Solution:

$$h = 298.4 + 0.95 \times 888.8 = 1142.8 \text{ Btu/lb}$$

Table 8-3 gives the properties of superheated steam. Since the condition of superheated steam cannot be specified without knowing both temperature and pressure, this table is set up on that basis. If pressure and temperature are known, enthalpy and entropy can be read from the table.

Table 8-4 gives enthalpy and entropy values for compressed water at various temperatures and pressures Note that the effect of pressure on these properties is much less pronounced than the effect of temperature. For example, the enthalpy of saturated water at 200°F (p = 11.5 psia), h_f = 167.99 Btu per lb. For compressed water at 200°F and 600 psia, h = 169.30 Btu per lb. The difference between the two values of enthalpy is only 1.31, while the difference in pressures is 588.5 psia. For most calculations, it is sufficiently accurate to take the value for saturated water at the same temperature as the compressed water.

Data on steam can also be presented in chart form. Several types of charts can be devised, but the most useful one is the enthalpy-entropy or Mollier chart. Figure 8-1 is a skeleton Mollier chart which is intended to illustrate the use of the large chart which is published as a part of all standard steam tables. The enthalpy and

Table 8-1. Saturated H_2O, Solid, Liquid, and Vapor*

Temp, °F	Press, lb/in.² abs	Volume, ft³/lb			Internal energy, Btu/lb			Enthalpy, Btu/lb			Entropy, Btu/(lb × °R)		
		V_f	V_{fg}	V_g	U_f	U_{fg}	U_g	H_f	H_{fg}	H_g	S_f	S_{fg}	S_g
Solid—Vapor													
−40	0.0019	0.01737	133,900	133,900	−177.00	1174 9	1021.6	−177.00	1221.2	1044.2	−0.3654	2.9087	2.5433
−20	0 0062	0.01739	42,200	42,200	−168.16	1172.7	1017.7	−168.16	1221.2	1053.0	−0.3448	2.7764	2.4316
0	0.0185	0.01742	14,770	14,770	−158.93	1170.0	1011.1	−158.93	1220.7	1061.8	−0.3241	2.6546	2.3305
20	0.0505	0 01745	5,658	5,658	−149.31	1167 0	1004.5	−149.31	1219.9	1070.6	−0.3038	2.5425	2.2387
32	0 0885	0.01747	3,306	3,306	−143.35	1164 9	997.9	−143.35	1219.1	1075.8	−0.2916	2.4793	2 1877
Liquid—Vapor													
32	0 08854	0 01602	3.306	3,306	0 00	1021 7	1021.7	0.00	1075.8	1075.8	0.00	2.1877	2.1877
35	0.09995	0 01602	2,947	2,947	3.02	1019 5	1022.6	3.02	1074.1	1077.1	0.0061	2.1709	2.1770
40	0 12170	0 01602	2,444	2,444	7 95	1016.4	1024.3	8.05	1071.3	1079.3	0.0162	2 0393	2.0948
45	0.14752	0 01602	2,036	2.036	13 05	1012.8	1025.9	13.06	1068.4	1081 5	0.0262	2.1167	2 1429
50	0.17811	0.01603	1,703	1,703	18.07	1009.5	1027.6	18.07	1065.6	1083.7	0.0361	2.0903	2.1264
60	0 2563	0.01604	1,206.6	1,206 7	28 06	1002 9	1031.0	28.06	1059.9	1088.0	0.0555	2 0393	2 0948
70	0 3631	0 01606	867.8	867.9	38 04	995 9	1034.0	38.04	1054.3	1092 3	0.0745	1.9902	2.0647
80	0 5069	0.01608	633.1	633 1	48 02	989 2	1037 2	48 02	1048.6	1096 6	0.0932	1.9428	2 0360
90	0 6982	0 01610	468.0	468 0	57.99	982 5	1040.5	57.99	1042.9	1100.9	0 1115	1 8972	2.0087
100	0 9492	0 01613	350 3	350.4	67.97	975.7	1043.7	67.97	1037.2	1105.2	0.1295	1.8531	1.9826
120	1.6924	0.01620	203.25	203 27	87.91	962 1	1050.0	87.92	1025.8	1113.7	0.1645	1.7694	1.9339
140	2.8886	0.01629	122.99	123.01	107.88	948 4	1056 3	107.89	1014 1	1122.0	0 1984	1 6910	1 8894
160	4.741	0 01639	77.27	77.29	127.88	934 5	1062.4	127.89	1002.3	1130.2	0 2311	1 6174	1.8485
180	7.510	0.01651	50.21	50.23	147.90	920 4	1068.3	147.92	990 2	1138.1	0.2630	1.5480	1.8109
200	11.526	0.01663	33.62	33.64	167.95	906.3	1074.2	167.99	977.9	1145.9	0.2938	1.4824	1.7762
212	14 696	0.01672	26.78	26.80	180 02	897.5	1077.5	180.07	970.3	1150.4	0.3120	1.4446	1 7566
220	17.186	0 01677	23 13	23 15	188.08	891.7	1079.8	188 13	965 2	1153.4	0 3239	1.4201	1.7440
240	24 969	0 01692	16 306	16 323	208 26	876.8	1085.1	208 34	952 2	1160.5	0.3531	1.3609	1.7140
260	35 429	0 01709	11 746	11 763	228 53	861 7	1090.2	228 64	938.7	1167.3	0.3817	1.3043	1.6860
280	49 203	0.01726	8.628	8 645	248 90	846 2	1095 1	249.06	924.7	1173.8	0.4096	1.2501	1.6597
300	67.013	0.01745	6.449	6 466	269 37	830 1	1099 5	269 59	910 1	1179 7	0.4369	1.1980	1 6350
320	89.66	0 01765	4 896	4 914	289 99	813 7	1103 7	290.28	894 9	1185 2	0 4637	1.1478	1.6115
340	118.01	0.01787	3 770	3 788	310 74	796 7	1107.4	311.13	879.0	1190.1	0.4900	1.0992	1.5891
360	153 04	0.01811	2 939	2 957	331 67	779 0	1110 7	332.18	862 2	1194.4	0.5158	1.0519	1.5677
380	195.77	0.01836	2.317	2 335	352.78	760.7	1113 5	353.45	844 6	1198.1	0.5413	1.0059	1.5471
400	247 31	0.01864	1 8447	1 8633	374 12	741 6	1115 7	374 97	826 0	1201.0	0.5664	0.9608	1 5272
450	422 6	0.0194	1 0799	1 0993	428 6	690.0	1118 6	430.1	774 5	1204.6	0.6280	0.8513	1.4793
500	680 8	0.0204	0.6545	0 6749	485 2	631.5	1116 7	487 8	713.9	1201.7	0.6887	0.7438	1.4325
600	1,542 9	0.0236	0 2432	0 2668	610 3	479 0	1089 3	617 0	548 5	1165.5	0 8131	0.5176	1.3307
700	3,093.7	0.0369	0.0392	0 0761	802 2	149.6	951 8	823 3	172 1	995 4	0.9905	0.1484	1.1389
705.4	3,206.2	0 0503	0	0 0503	872.9	0	872 9	902.7	0	902 7	1.0580	0	1.0580

* KEENAN, J. H., and F. G. KEYES, "Thermodynamic Properties of Steam," John Wiley & Sons, Inc., New York, 1936. *U* values computed from data in reference. Table reprinted from LICHTY, "Thermodynamics," 2d ed., McGraw-Hill Book Company. Inc., 1948, as adapted from KEENAN and KEYES.

Table 8-2. Saturated H_2O, Liquid and Vapor*

Press, lb/in.² abs	Temp, °F	Volume, ft³/lb			Internal energy, Btu/lb			Enthalpy, Btu/lb			Entropy, Btu/(lb × °R)		
		V_f	V_{fg}	V_g	U_f	U_{fg}	U_g	H_f	H_{fg}	H_g	S_f	S_{fg}	S_g
In. Hg. 1	79.03	0.01608	652.28	652.3	47.05	990.0	1037.0	47.05	1049.2	1096.3	0.0914	1.9473	2.0387
2	101.14	0.01614	339.18	339.2	69.10	974.9	1044.0	69.10	1036.6	1105.7	0.1316	1.8481	1.9797
3	115.06	0.01618	231.58	231.6	82.99	965.5	1048.5	82.99	1028.6	1111.6	0.1560	1.7896	1.9456
3	141.48	0.01630	118.69	118.71	109.36	947.3	1056.7	109.37	1013.2	1122.6	0.2008	1.6855	1.8863
5	162.24	0.01640	73.50	73.52	130.12	933.0	1063.1	130.13	1001.0	1131.1	0.2347	1.6094	1.8441
10	193.21	0.01659	38.40	38.42	161.14	911.1	1072.2	161.17	982.1	1143.3	0.2835	1.5041	1.7876
15	213.03	0.01672	26.27	26.29	181.06	896.7	1077.8	181.11	969.7	1150.8	0.3135	1.4415	1.7549
20	227.96	0.01683	20.07	20.089	196.10	885.8	1081.9	196.16	960.1	1156.3	0.3356	1.3962	1.7319
25	240.07	0.01692	16.286	16.303	208.34	876.8	1085.1	208.42	952.1	1160.6	0.3533	1.3606	1.7139
30	250.33	0.01701	13.729	13.746	218.73	869.1	1087.8	218.82	945.3	1164.1	0.3680	1.3318	1.6993
40	267.25	0.01715	10.482	10.498	235.90	856.1	1092.0	236.03	933.7	1169.7	0.3919	1.2844	1.6763
50	281.01	0.01727	8.498	8.515	249.93	845.4	1095.3	250.09	924.0	1174.1	0.4110	1.2474	1.6585
60	292.71	0.01738	7.158	7.175	261.90	836.0	1097.9	262.09	915.5	1177.6	0.4270	1.2168	1.6438
70	302.92	0.01748	6.189	6.206	272.38	827.8	1100.2	272.61	907.9	1180.6	0.4409	1.1906	1.6315
80	312.03	0.01757	5.465	5.472	281.76	820.3	1102.1	282.02	901.1	1183.1	0.4531	1.1676	1.6207
100	327.81	0.01774	4.415	4.432	298.08	807.1	1105.2	298.40	888.8	1187.2	0.4740	1.1286	1.6026
120	341.25	0.01789	3.712	3.728	312.05	795.6	1107.6	312.44	877.9	1190.4	0.4916	1.0962	1.5878
140	353.02	0.01802	3.202	3.220	324.35	785.2	1109.6	324.82	868.2	1193.0	0.5069	1.0682	1.5751
160	363.53	0.01815	2.816	2.834	335.39	775.8	1111.2	335.93	859.2	1195.1	0.5204	1.0436	1.5640
180	373.06	0.01827	2.514	2.532	345.42	767.1	1112.5	346.03	850.8	1196.9	0.5325	1.0217	1.5542
200	381.79	0.01839	2.270	2.288	354.68	759.0	1113.7	355.36	843.0	1198.4	0.5435	1.0018	1.5453
220	389.86	0.01850	2.069	2.087	363.27	751.3	1114.6	364.02	835.6	1199.6	0.5537	0.9835	1.5372
240	397.37	0.01860	1.900	1.9183	371.29	744.1	1115.4	372.12	828.5	1200.6	0.5631	0.9667	1.5298
260	404.42	0.01870	1.756	1.7748	378.86	737.3	1116.1	379.76	821.8	1201.5	0.5719	0.9510	1.5229
280	411.05	0.01880	1.6323	1.6511	386.01	730.7	1116.7	386.90	815.3	1202.3	0.5801	0.9363	1.5164
300	417.33	0.01890	1.5244	1.5433	392.79	724.3	1117.1	393.84	809.0	1202.8	0.5879	0.9225	1.5104
320	423.29	0.01899	1.4296	1.4485	399.26	718.3	1117.6	400.39	803.0	1203.4	0.5952	0.9094	1.5046
340	428.97	0.01908	1.3435	1.3645	405.46	712.4	1117.9	406.66	797.1	1203.7	0.6022	0.8970	1.4992
360	434.40	0.01917	1.2704	1.2895	411.39	706.8	1118.2	412.67	791.4	1204.1	0.6090	0.8851	1.4941
380	439.60	0.01925	1.2030	1.2222	417.10	701.3	1118.4	418.45	785.8	1204.3	0.6153	0.8738	1.4891
400	444.59	0.0193	1.1420	1.1613	422.6	695.9	1118.5	424.0	780.5	1204.5	0.6214	0.8630	1.4844
500	467.01	0.0197	0.9081	0.9278	447.6	671.0	1118.6	449.4	755.0	1204.4	0.6487	0.8147	1.4634
600	486.21	0.0201	0.7497	0.7698	469.4	648.3	1117.7	471.6	731.6	1203.2	0.6720	0.7734	1.4454
700	503.10	0.0205	0.6349	0.6554	488.8	627.5	1116.3	491.5	709.7	1201.2	0.6925	0.7371	1.4296
800	518.23	0.0209	0.5478	0.5687	506.6	607.8	1114.4	509.7	688.9	1198.6	0.7108	0.7045	1.4153
1,000	544.61	0.0216	0.4240	0.4456	538.4	571.0	1109.4	542.4	649.4	1191.8	0.7430	0.6467	1.3897
1,500	596.23	0.0235	0.2530	0.2765	605.1	486.1	1091.2	611.6	556.3	1167.9	0.8082	0.5269	1.3351
2,000	635.82	0.0257	0.1621	0.1878	662.2	403.4	1065.6	671.7	463.4	1135.1	0.8619	0.4230	1.2849
2,500	668.13	0.0287	0.1021	0.1307	717.3	313.3	1030.6	730.6	360.5	1091.1	0.9126	0.3197	1.2322
3,000	695.36	0.0346	0.0512	0.0858	783.4	189.3	972.9	802.5	217.8	1020.3	0.9731	0.1885	1.615

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* KEENAN, J. H., and F. G. KEYES, "Thermodynamic Properties of Steam," John Wiley & Sons, Inc., New York, 1936.
Table reprinted from LICHTY, "Thermodynamics," 2d ed., McGraw-Hill Book Company, Inc., 1948, as adapted from KEENAN and KEYES.

Table 8-3. Superheated H_2O, Vapor*

P, lb/in.^{2}; T, °F; V, ft^3/lb; U, Btu/lb H, Btu/lb; S, Btu/(lb × °R)

Press, lb/in.^{2} abs (sat temp)	Prop	Temperature, °F											
		200	300	400	500	600	700	800	900	1000	1200	1400	1600
1 (101.74)	V U H S	392.6 1077.7 1150.4 2.0512	452.3 1112.1 1195.8 2.1153	512.0 1146.9 1241.7 2.1720	571.6 1182.5 1288 3 2.2233	631.2 1218.9 1335.7 2.2702	690.8 1255.9 1383 8 2.3137	750.4 1293.9 1432 8 2.3542	809.9 1332.8 1482.7 2.3923	869.5 1372.6 1533.5 2.4283	988.7 1455.7 1637.7 2.4952	1107.8 1540.7 1745.7 2.5566	1227.0 1630.4 1857.5 2.6137
5 (162.24)	V U H S	78.16 1076.5 1148.8 1.8718	90.25 1111.5 1195.0 1.9370	102.26 1146.6 1241.2 1.9942	114.22 1182.4 1288.0 2.0456	126.16 1218.6 1335.4 2.0927	138.10 1255.8 1383 6 2.1361	150 03 1293.8 1432.7 2.1767	161.95 1332.7 1482.6 2.2148	173.87 1372.6 1533.4 2.2509	197.71 1455.7 1637.7 2.3178	221.6 1540.7 1745.7 2.3792	245.4 1630.4 1857.4 2.4368
14.696 (212.00)	V U H S	....	30.53 1109.7 1192 8 1.8160	34.68 1145.7 1239 9 1 8743	38 78 1181 7 1287 1 1.9261	42.86 1218.2 1334.8 1.9734	49.64 1255.5 1383.2 2.0170	51.00 1293.6 1432.3 2.0576	55.07 1332.5 1482.3 2.0958	59.13 1372.3 1533.1 2.1319	67.25 1455.5 1637.5 2.1989	75.37 1540.6 1745.5 2.2603	83.48 1630.2 1857.3 2.3174
30 (250.33)	V U H S	....	14.816 1107.1 1189.3 1.7336	16 597 1144.1 1237.9 1.7937	18 933 1180.6 1285.7 1.8464	20.95 1217.5 1333 8 1.8940	22.96 1254.9 1382.4 1.9379	24 96 1293.1 1431.7 1.9786	26.95 1332.2 1481.8 2.0169	28.95 1372.0 1532.7 2.0530	32.93 1454.7 1637.2 2.1201	36.91 1540.4 1745.3 2.1815	40.89 1630.0 1857.1 2.2386
50 (281.01)	V U H S	....	8.773 1103.1 1184.3 1.6721	10.065 1142.0 1235.1 1 7349	11.309 1179.2 1283.9 1.7887	12.532 1216.5 1332.5 1.8368	13.744 1254.2 1381.4 1.8809	14.950 1292.6 1430.9 1.9219	16.152 1331.6 1481.1 1.9602	17.352 1371.5 1532.1 1.9964	19.747 1454.1 1636.8 2.0636	22.14 1540.1 1745.0 2.1251	24.53 1629.8 1856.8 2.1822
100 (327.81)	V U H S	....	....	4 937 1136.2 1227 6 1.6518	5.589 1175.7 1279.1 1.7085	6.218 1214.0 1329.1 1.7581	6.835 1252.3 1378 9 1.8029	7.446 1291.1 1428 9 1.8443	8.052 1330.5 1479 5 1.8829	8.656 1370 7 1530 8 1.9193	9.860 1453.2 1635.7 1.9867	11 060 1539.5 1744.2 2.0484	12.258 1629.3 1856.2 2.1056
200 (381.79)	V U H S	....	....	2.361 1123.0 1210 3 1.5594	2.726 1168.0 1268 9 1.6240	3.060 1208.8 1322 1 1 6767	3.380 1248.5 1373 6 1.7232	3 693 1288.1 1424.8 1.7655	4.002 1328.1 1476 2 1.8048	4.309 1368.5 1528.0 1.8415	4.917 1451.7 1633.7 1.9044	5.521 1538.2 1742 6 1.9713	6.123 1628.3 1855.0 2.0287

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* Keenan, J. H., and F. G. Keyes, "Thermodynamic Properties of Steam," John Wiley & Sons, Inc., 1936. U values computed from data in reference. Table reprinted from Lichty, "Thermodynamics," 2d ed., McGraw-Hill Book Company, Inc., 1948, as adapted from Keenan and Keyes.

Table 8-3. Superheated H_2O, Vapor* (*Cont.*)

Press, lb/in.² abs (Sat temp)	Prop	Temperature, °F											
		200	300	400	500	600	700	800	900	1000	1200	1400	1600
300 (417.33)	*V*	......	......	......	1.7675	2.005	2.227	2.442	2.652	2.859	3.269	3.674	4.078
	U	......	......	......	1159.4	1203.4	1244.6	1285.0	1325.5	1366.5	1450.2	1537.0	1627.3
	H	......	......	......	1257.6	1314.7	1368.3	1420.6	1472.8	1525.2	1631.7	1741.0	1853.7
	S	......	......	......	1.5701	1.6268	1.6751	1.7184	1.7582	1.7954	1.8638	1.9260	1.9835
400 (444.59)	*V*	......	......	......	1.2851	1.4770	1.6508	1.8161	1.9767	2.134	2.445	2.751	3.055
	U	......	......	......	1150.0	1197.5	1240.5	1281.9	1323.0	1364.4	1448.6	1535.8	1626.3
	H	......	......	......	1245.1	1306.9	1362.7	1416.4	1469.4	1522.4	1629.6	1739.5	1852.5
	S	......	......	......	1.5281	1.5894	1.6398	1.6842	1.7247	1.7623	1.8311	1.8936	1.9485
500 (467.01)	*V*	......	......	......	0.9927	1.1591	1.3044	1.4405	1.5715	1.6996	1.9504	2.197	2.442
	U	......	......	......	1139.4	1191.3	1236.3	1278.8	1320.6	1362.3	1447.1	1534.6	1625.3
	H	......	......	......	1231.3	1298.6	1357.0	1412.1	1466.0	1519.6	1627.6	1737.7	1851.3
	S	......	......	......	1.4919	1.5588	1.6115	1.6571	1.6982	1.7363	1.8056	1.8683	1.9262
1000 (544.61)	*V*	......	......	......	......	0.5140	0.6084	0.6878	0.7604	0.8294	0.9615	1.0893	1.2146
	U	......	......	......	......	1153.7	1212.7	1261.9	1307.5	1351.6	1439.3	1528.4	1620.5
	H	......	......	......	......	1248.8	1325.3	1389.2	1448.2	1505.1	1617.3	1730.0	1845.0
	S	......	......	......	......	1.4450	1.5141	1.5670	1.6121	1.6525	1.7245	1.7886	1.8474
2000 (635.82)	*V*	......	......	......	......	......	0.2489	0.3074	0.3532	0.3935	0.4668	0.5352	0.6011
	U	......	......	......	......	......	1147.9	1221.7	1278.5	1328.8	1423.3	1516.0	1610.0
	H	......	......	......	......	......	1240.0	1335.5	1409.2	1474.5	1596.1	1714.1	1832.5
	S	......	......	......	......	......	1.3783	1.4576	1.5139	1.5603	1.6384	1.7055	1.7660
3000 (695.36)	*V*	......	......	......	......	......	0.0984	0.1760	0.2159	0.2476	0.3018	0.3505	0.3966
	U	......	......	......	......	......	1006.1	1169.5	1245.1	1304.3	1406.7	1503.4	1599.7
	H	......	......	......	......	......	1060.7	1267.2	1365.0	1441.8	1574.3	1698.0	1819.9
	S	......	......	......	......	......	1.1966	1.3690	1.4439	1.4984	1.5837	1.6540	1.7163
4000	*V*	......	......	......	......	......	0.0287	0.1052	0.1462	0.1743	0.2192	0.2581	0.2943
	U	......	......	......	......	......	742.6	1096.9	1206.1	1277.8	1389.8	1490.6	1589.3
	H	......	......	......	......	......	763.8	1174.8	1314.4	1406.8	1552.1	1681.7	1807.2
	S	......	......	......	......	......	0.9347	1.2757	1.3827	1.4482	1.5417	1.6154	1.6795
5000	*V*	......	......	......	......	......	0.0268	0.0593	0.1036	0.1303	0.1696	0.2027	0.2329
	U	......	......	......	......	......	721.6	992.2	1160.6	1248.9	1372.5	1477.7	1579.0
	H	......	......	......	......	......	746.4	1047.1	1256.5	1369.5	1529.5	1665.3	1794.5
	S	......	......	......	......	......	0.9152	1.1622	1.3231	1.4034	1.5066	1.5839	1.6499

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* Keenan, J. H., and F. G. Keyes, "Thermodynamic Properties of Steam," John Wiley & Sons, Inc., 1936. *U* values computed from data in reference. Table reprinted from Lichty, "Thermodynamics," 2d ed., McGraw-Hill Book Company, Inc., 1948, as adapted from Keenan and Keyes.

Table 8-4. Liquid H_2O*

P, lb/in.²; T, °F; V, ft³/lb; U, Btu/lb; H, Btu/lb; S, Btu/(lb × °R).

Abs press, lb/in.² (sat temp)	T	32	100	200	300	400	500	600	650	680	700	705.4	
	P	0.08854	0.9492	11.526	67.013	247.31	680.8	1542.9	2208.2	2708.1	3093.7	3206.2	
Saturated liquid	V	0.016022	0.016132	0.016634	0.017449	0.018639	0.020432	0.023629	0.02680	0.03054	0.03692	0.05030	Saturated liquid
	U	0	67.97	167.95	269.37	374.12	485.23	610.26	684.8	742.0	802.18	872.86	
	H	0	67.97	167.99	269.59	374.97	487.82	617.0	695.7	757.3	823.3	902.7	
	S	0	0.12948	0.29382	0.43694	0.56638	0.68871	0.68871	0.8828	0.9351	0.9905	1.0580	
200 (381.79)	V	0.016011	0.016121	0.016623	0.017438	....	....	....	....	....	....	....	0.01839
	U	0.02	67.91	167.78	269.17	....	....	....	....	....	....	....	354.68
	H	0.61	68.51	168.40	269.82	....	....	....	....	....	....	....	355.36
	S	0.00003	0.12943	0.29361	0.43683	....	....	....	....	....	....	....	0.5435
600 (486.21)	V	0.015987	0.016100	0.016600	0.017406	0.018595	....	....	....	....	....	....	0.0201
	U	0.02	67.85	167.46	268.63	373.30	....	....	....	....	....	....	469.4
	H	1.80	69.64	169.30	270.56	375.36	....	....	....	....	....	....	471.6
	S	0.00007	0.12921	0.29308	0.43600	0.056542	....	....	....	....	....	....	0.6720
1,000 (544.61)	V	0.015965	0.016075	0.016580	0.017380	0.018552	0.020368	....	....	....	....	....	0.0216
	U	0.03	67.69	167.13	268.12	372.38	483.91	....	....	....	....	....	538.4
	H	2.99	70.67	170.20	271.34	375.81	487.68	....	....	....	....	....	542.4
	S	0.00015	0.12895	0.29262	0.43530	0.56438	0.68730	....	....	....	....	....	0.7430
2,000 (635.82)	V	0.015912	0.016033	0.016526	0.017311	0.018444	0.020154	0.023303	....	....	....	....	0.0257
	U	0.08	67.34	166.38	266.82	370.17	479.98	605.9	....	....	....	....	662.2
	H	5.97	73.28	172.50	273.23	377.00	487.44	614.5	....	....	....	....	671.7
	S	0.00022	0.12830	0.29143	0.43352	0.56181	0.68213	0.8088	....	....	....	....	0.8619
3,000 (695.36)	V	0.015859	0.015985	0.016474	0.017242	0.018339	0.019961	0.022750	0.025477	0.02882	....	....	0.0346
	U	0.19	66.97	165.60	265.51	368.12	476.33	597.5	676.8	729.1	....	....	783.4
	H	9.00	75.85	174.75	275.08	378.30	487.41	610.1	691.0	745.1	....	....	802.5
	S	0.00028	0.12769	0.29026	0.43182	0.55935	0.67929	0.8007	0.8743	0.9231	....	....	0.9781
4,000	V	0.015807	0.015940	0.016424	0.017174	0.018239	0.019787	0.022307	0.024492	0.02654	0.02871	0.02951	....
	U	0.18	66.66	164.86	264.28	366.18	473.01	590.5	665.2	705.3	742.5	754.3	....
	H	11.88	78.46	177.02	277.00	379.68	487.66	607.0	682.3	724.9	763.8	776.1	....
	S	0.00029	0.12706	0.28908	0.43017	0.55698	0.67568	0.7938	0.8622	0.9008	0.9347	0.9453	....
5,000	V	0.015755	0.015896	0.016374	0.017119	0.018143	0.019627	0.021936	0.023785	0.02536	0.02675	0.02721	....
	U	0.17	66.34	164.14	263.11	364.26	469.91	584.6	654.1	680.8	721.6	730.8	....
	H	14.75	81.05	179.29	278.95	381.05	488.07	604.9	676.1	714.3	746.4	756.0	....
	S	0.00022	0.12641	0.28790	0.42854	0.55464	0.67230	0.7878	0.8527	0.8872	0.9152	0.9234	....

* Computed from Keenan and Keyes, "Thermodynamic Properties of Steam," Table IV, p. 75, John Wiley & Sons, Inc., New York, 1936. Table reprinted from Lichty, "Thermodynamics," 2d ed., McGraw-Hill Book Company, Inc., 1948, as adapted from Keenan and Keyes.

entropy of saturated steam can be determined by finding the intersection of the saturated-steam line and the proper pressure line. The enthalpy and entropy of superheated steam are found by locating the intersection of the lines representing the pressure and temperature of the superheated steam. If the pressure and quality of wet steam are known, enthalpy and entropy can be found at the intersection of the quality and pressure lines.

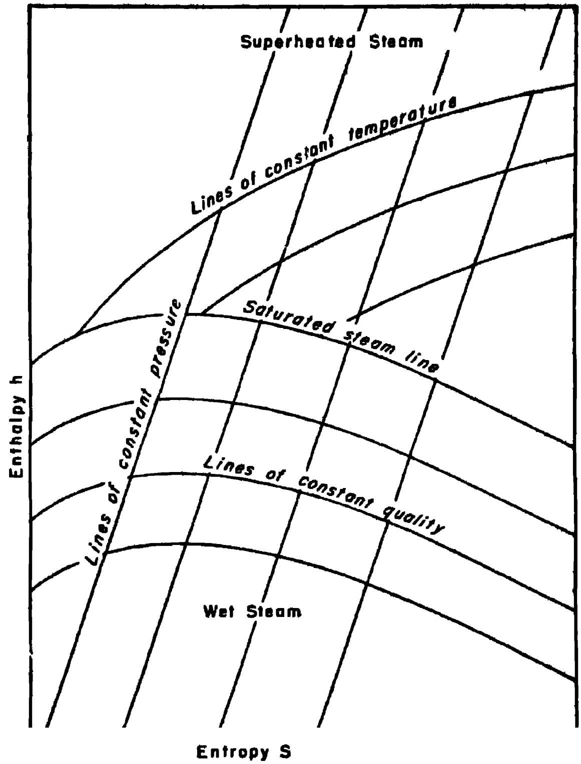
Superheated Steam
Lines of constant temperature
Saturated steam line
Lines of constant pressure
Lines of constant quality
Wet Steam
Enthalpy h
Entropy S

FIG. 8-1. Mollier chart for steam.

FIRST LAW OF THERMODYNAMICS

The first law of thermodynamics states that energy can neither be created nor destroyed but only altered in form. When applied to a boiler, turbine, or similar device operating under steady conditions, the first law says that the energy flowing in is equal to the energy flowing out. This energy may be carried by steam or water; it may be heat or mechanical work. This is expressed in equation form for 1 lb of fluid as

$$Q + h_1 = W + h_2 \qquad (2)$$

where Q = heat *in* per pound of fluid
W = work *out* per pound of fluid
h_1 = enthalpy of entering fluid
h_2 = enthalpy of leaving fluid

All quantities are expressed in Btu. If heat flows out or work in, the fact is expressed by a minus sign. The use of this equation will be illustrated by several examples.

Heat-transfer Devices and the First Law of Thermodynamics.

Example 4: A *boiler* receives water at 200°F and delivers steam at 400 psia and 600°F. Compute the heat added per pound of steam.

Solution: Water flowing into the boiler is evaporated into steam and superheated. Heat flows through the walls of the boiler and superheater tubes, but no mechanical work is done in the boiler. Therefore, the equation becomes

$$Q = h_2 - h_1 \tag{3}$$

where Q = heat added per pound of steam
h_1 = enthalpy of feed water
h_2 = enthalpy of superheated steam

The values for h are found from the tables and substituted in Eq. (3). Then,

$$Q = 1307 - 168 = 1139 \text{ Btu/lb}$$

Example 5: A *boiler* receives water at 200°F and delivers steam at 100 psia with 97 per cent quality. Compute the heat added per pound of steam.

Solution: Here the enthalpy of the steam is given by Eq. (1) as

$$h = 298 + 0.97 \times 889 = 1{,}160$$

and $$Q = 1160 - 168 = 992 \text{ Btu/lb}$$

Example 6: A *process cooker* receives steam at 20 psia with a quality of 95 per cent and discharges saturated water through a trap. Compute the number of pounds of steam required per hour to supply 100,000 Btu per hr to process.

Solution: A steam-heated cooker is like a boiler working in reverse. Steam enters and condenses, giving up heat. It is specified in the statement of the problem that the condensate is removed as saturated water, *i.e.*, water at the saturation temperature corresponding to the pressure. Equation (3) applies with

$$h_2 = 196.2 \text{ (at 20 psia)}$$
$$h_1 = 196.2 + 0.95 \times 960.1 = 1{,}108.3$$

and $$Q = 196.2 - 1108.3 = -912.1 \text{ Btu/lb}$$

The minus sign indicates that heat is flowing out of the steam. The weight of steam required per hour is

$$100{,}000/912.1 = 110 \text{ lb/hr}$$

In many cases involving the condensation of steam, the condensate may be "subcooled" or discharged at a temperature below the saturation temperature corresponding to the steam pressure. For most engineering computations, the enthalpy of the subcooled condensate may be considered as being equal to the enthalpy of saturated water at the actual temperature of the condensate.

Example 7: A *surface condenser* operating with a 28-in. vacuum receives 50,000 lb per hr of steam at a quality of 88 per cent. The atmospheric pressure is 30 in. Hg. Condensate is discharged to a hot well at 90°F. If cooling water enters at 75°F and leaves at 85°F, compute the gallons of water per minute required.

Solution: The steam side of the condenser is similar to the cooker of Example 6. The heat removed per pound of steam is

$$Q = h_2 - h_1$$

For wet steam at 2 in. Hg absolute (30 in. − 28 in.)

$$h_1 = 69 + 0.88 \times 1037 = 982 \text{ Btu lb}$$

The condensate is subcooled water because its temperature of 90°F is lower than the saturation temperature at 2 in. Hg (101°F), but its enthalpy can be found with sufficient accuracy by assuming it saturated at 90°F, so h_2 = 58 Btu per lb.

$$Q = 58 - 982 = -924 \text{ Btu/lb}$$

This is the heat removed from each pound of steam. The amount removed per minute is

$$-924 \times 50{,}000/60 = -770{,}000 \text{ Btu/min}$$

This heat must be carried away by the cooling water. The heat picked up by each pound of water may be calculated in two ways. The first method consists in finding the change in enthalpy of each pound of cooling water. The water enters at 75°F with an enthalpy of 43 Btu per lb and leaves at 85°F with an enthalpy of 53 Btu per lb. Therefore, each pound of water picks up

$$Q = 53 - 43 = 10 \text{ Btu/lb}$$

The second method consists in using the specific heat (amount of heat required to raise the temperature of 1 lb 1°F) which for water is unity. Therefore, each pound picks up

$$Q = 1 \times (85 - 75) = 10 \text{ Btu/lb}$$

The number of pounds of cooling water required is

$$770{,}000/10 = 77{,}000 \text{ lb/min}$$

and since 1 gal of water weighs 8.34 lb, the condenser requires

$$77{,}000/8.34 = 9{,}230 \text{ gal/min}$$

Example 8: An *open* (*direct-contact*) *feed-water heater* heats 50,000 lb per hr of water from 90 to 212°F with steam at atmospheric pressure. If the steam is saturated, compute the number of pounds per hour required.

Solution: The heat required per pound of water heated is

$$Q = 180 - 58 = 122 \text{ Btu}$$

and the total heat is 122 × 50,000 = 6,100,000 Btu per hr. Each pound of steam enters with an enthalpy of 1,150 Btu This steam is condensed in the heater to saturated water at 212°F with an enthalpy of 180. Hence,

$$Q = 180 - 1{,}150 = -970 \text{ Btu/lb of steam}$$

The number of pounds of steam required is

$$6{,}100{,}000/970 = 6{,}300 \text{ lb/hr}$$

The total flow of hot water from the heater is then

$$50{,}000 + 6{,}300 = 56{,}300 \text{ lb/hr}$$

Example 9. One hundred pounds per minute of *oil* is cooled in a *heat exchanger* from 200 to 100°F by water entering at 75°F and leaving at 90°F. If the specific heat of the oil is 0.45 Btu per lb per deg F, how many gallons of cooling water per minute are required?

Solution: The heat given up by the oil is

$$Q = 100 \times 0.45 \times (200 - 100) = 4500 \text{ Btu/min}$$

Each pound of water picks up

$$Q = 1 \times (90 - 75) = 15 \text{ Btu/lb}$$

The amount of water required is

$$4{,}500/15 = 300 \text{ lb/min}$$

or

$$300/8.34 = 36 \text{ gal/min}$$

Expansion of Steam in Turbines and Engines and the First Law of Thermodynamics. The behavior of steam in passing through an engine or turbine can be illustrated by a Mollier chart (Fig. 8-2). Steam at condition a is admitted to a turbine. If the steam is superheated, point a can be located on the Mollier chart at the intersection of the lines representing its temperature and pressure. If it is wet, the point is at the intersection of lines representing its pressure and quality. In Fig. 8-2, steam at point a is superheated, but the discussion to follow is also true for wet steam.

Since the purpose of the turbine is to convert as much of the heat energy of the steam as possible into work, it is desirable that the enthalpy of the exhaust steam be as small as possible. When steam is expanded to a definite pressure, the final point on the chart will be somewhere on the line representing that pressure. If the expansion is at constant entropy, the point will be directly below point a (point b). If friction is present, the entropy will increase, and the final point is at c, for example.

The ideal expansion takes place at constant entropy and may be represented on the Mollier chart as a vertical straight line.

The heat flow through the cylinder of a turbine or engine is quite small and may be neglected except in very small machines. For this condition, Eq. (2), which expresses the first law of thermodynamics, may be written as follows:

$$W = h_1 - h_2 \tag{4}$$

The application of this equation to the isentropic or ideal expansion of steam through a turbine is illustrated by Example 10.

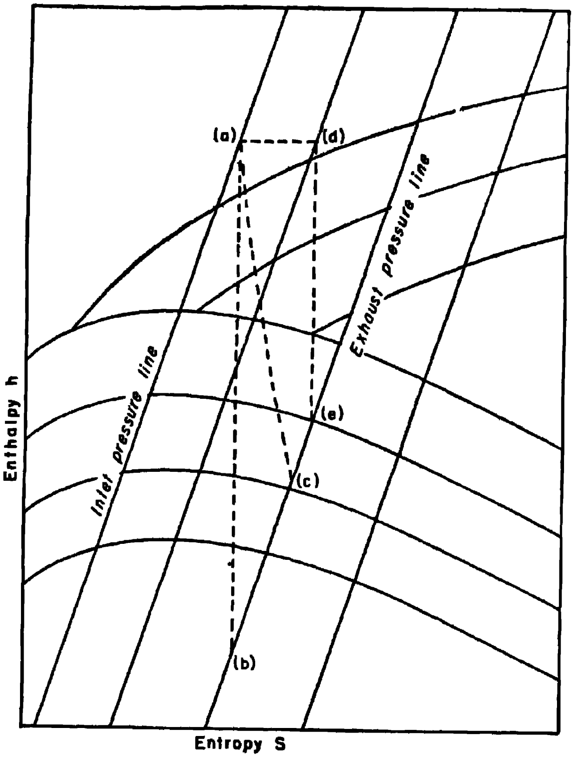

FIG. 8-2. Mollier chart showing turbine expansion lines.

EXAMPLE 10: A turbine receives steam at 1,400 psia and 840°F and expands to a pressure of 2 in. Hg abs. What is the steam rate in pounds per kilowatthour if the turbine expands isentropically?

SOLUTION: From the Mollier chart, h_1 is 1396 Btu per lb. The enthalpy of the exhaust steam h_2 can be found by going straight down the chart (constant entropy) until the 2-in. pressure line is reached and is 859 Btu per lb. By Eq. (4)

$$W = 1{,}396 - 859 = 537$$

Therefore each pound of steam develops 537 Btu of work. One kilowatt requires 3,413 Btu per hr, so the steam rate is

$$3{,}413/537 = 6.35 \text{ lb/kwhr}$$

The steam rate as determined from an isentropic expansion is called the "ideal" or theoretical steam rate and is the minimum steam flow required to develop 1 kw under ideal conditions. In the actual machine friction increases the entropy during expansion, and less work is done per pound of steam because the enthalpy of the exhaust steam is greater.

Throttling Processes and the First Law of Thermodynamics. A third general type of process involves the throttling or reduction of pressure by an orifice or throttle valve. Usually the heat losses are very small and can be neglected, and there is no external work done. The energy equation then becomes

$$h_1 = h_2 \tag{5}$$

i.e., the energy content of the throttled steam is unchanged.

This fact is the basis of the throttling calorimeter. If wet steam at moderate pressures is throttled to about atmospheric pressure, it will become superheated if not originally too wet. This can be seen on the Mollier chart by drawing a horizontal line (constant enthalpy) from the point representing the initial condition of the steam to the exhaust pressure line. The enthalpy of the superheated steam can be found from the tables or chart, and this is also the enthalpy of the steam at the higher pressure. The quality of that steam can be read from the chart or computed. This is illustrated by Example 11.

EXAMPLE 11: Steam is throttled from 100 psia to 20 psia and 250°F. Compute the initial quality.

SOLUTION: In this case, h_2 = 1,167.3 from the superheated steam tables or the Mollier chart (Fig. 8-2). At 100 psia, h_f = 298.4, and h_{fg} = 888.8

$$h_1 = h_f + xh_{fg}$$

but h_1 is known to equal h_2, so

$$1{,}167.3 = 298.4 + (888.8)x$$
$$x = (1{,}167.3 - 298.4)/888.8 = 0.976$$

Hence the quality of the steam at 100 psia is 97.6 per cent.

When steam flows through a throttling governor on a turbine or engine, there is no enthalpy change through the valve. However, there is a decrease in the available energy of the steam, or, in other words, a decrease in the amount of work which can be developed. The effect of the throttling governor on available energy can be shown on a Mollier chart (Fig. 8-2). Steam at point *a* could be expanded isentropically to point *b* or throttled to point *d* and then expanded isentropically to point *e*. It can be seen from Fig. 8-2 that less work is done by the steam that has been throttled. The available energy is less because the entropy has increased.

SECOND LAW OF THERMODYNAMICS

The second law of thermodynamics deals with one of the fundamental principles of science, the concept that energy is being converted in all natural processes into forms in which less of the energy can be converted into work. This principle is met in so many fields of science that the many statements of the second law which appear in books on thermodynamics, physics, and chemistry often seem to be unrelated to one other. This discussion will be confined to that phase of the second law which is concerned with the efficiency of conversion of heat into work.

Let it be assumed that a source of heat exists at a constant high temperature T_1, that the heat which cannot be converted into work may be rejected to a reservoir such as the atmosphere or a large body of water at a constant temperature T_2, and that a suitable working medium such as steam or air is contained in a cylinder behind a gastight piston. This piston slides without friction in a cylinder so arranged that heat may be transferred to the working medium from the high-temperature source and rejected from the medium to a low-temperature sink or receiver. In 1824, Sadi Carnot demonstrated that the maximum amount of work could be obtained from a given quantity of heat if the following conditions exist: (1) The heat transferred from the source to the working fluid is transferred at the maximum possible temperature which is the temperature of the source T_1; (2) the heat that cannot be converted to work is rejected at the lowest possible temperature which is the temperature of the sink T_2; and (3) expansion of the working medium from T_1 to T_2 and compression from T_2 to T_1 must take place without friction or heat transfer. Under these ideal conditions, the theoretical or maximum efficiency of conversion of heat into work is independent of the working medium used and is given by

$$e = (T_1 - T_2)/T_1 = 1 - T_2/T_1 \tag{6}$$

Equation (6) is often used as a statement of the second law of thermodynamics.

Equation (6) shows that the theoretical efficiency can be increased by increasing the initial temperature T_1 or reducing the exhaust temperature T_2. Suppose cooling water is available at 70°F, or 70 + 460 = 530°R, and a source of heat exists at 700°F, or 700 + 460 = 1,160°R. Then the maximum possible efficiency under those conditions is

$$e = 1 - 530/1,160 = 0.543$$

Heat energy is low-grade energy which can be transformed into work only with difficulty in complicated apparatus and at relatively low efficiency. In all natural processes that involve heat transfer and fluid friction, the temperature level of the energy tends to decrease, and less of it may be converted into work.

RANKINE CYCLE

The ideal or theoretical cycle of the simple power plant is the Rankine cycle which is illustrated in Fig. 8-3. This cycle is performed in four pieces of equipment which are connected by suitable piping. They are (1) the boiler and superheater in which steam is generated by heat released by burning fuel, (2) the engine or turbine in which work is done by the expansion of the steam, (3) the condenser which converts the exhaust steam into water and rejects the waste heat to cooling water, and (4) a boiler-feed pump which returns the condensate to the boiler.

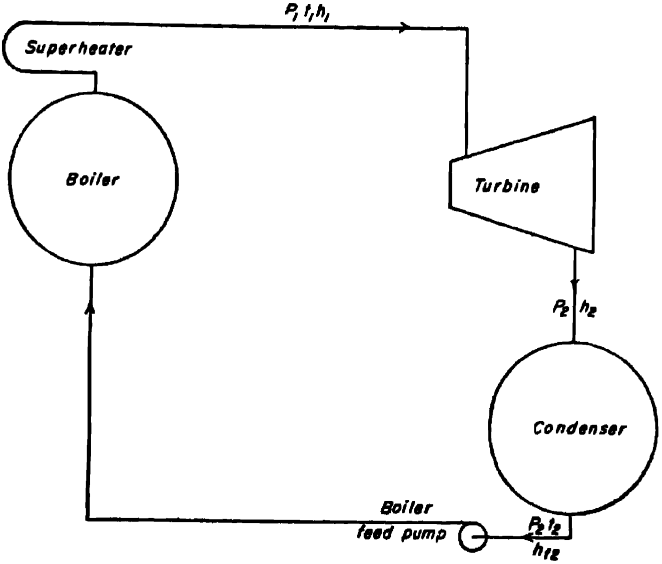

FIG. 8-3. Rankine cycle flow diagram.

The efficiency of this type of plant will be worked out under the following conditions: no pressure drop or heat transfer in the piping system, isentropic expansion in the turbine, saturated water in the condenser, and no work in the boiler-feed

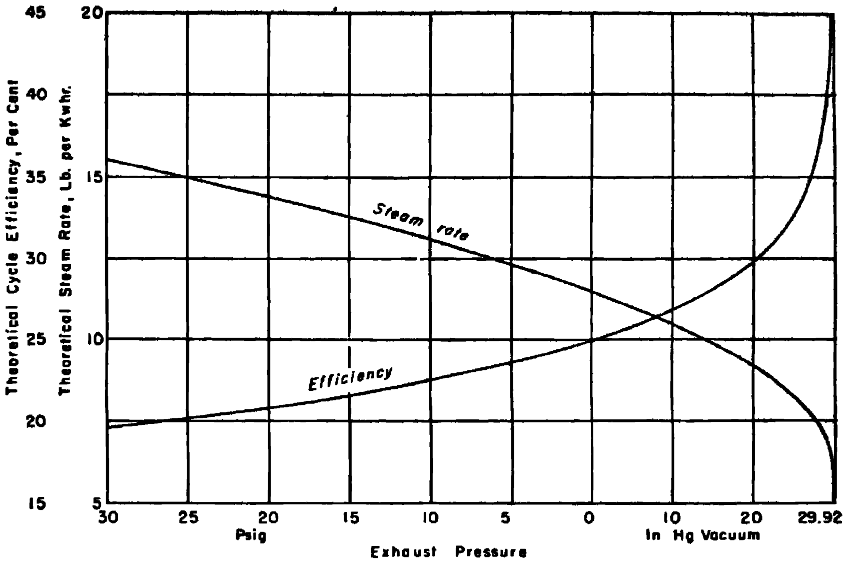

FIG. 8-4. Effect of exhaust pressure on steam rate and cycle efficiency.

pump (the amount of work required by the pump is very small for moderate boiler pressures and can be neglected for ordinary calculations). Then, from Eq. (1) for the first law, using the symbols from Fig. 8-3.

In the boiler and superheater

$$Q_{in} = h_1 - h_{f2}$$

In the turbine

$$W = h_1 - h_2$$

In the condenser

$$Q_{out} = h_2 - h_{f2}$$

$$\begin{aligned} \text{Efficiency} &= (\text{work output})/(\text{heat input}) \\ &= (h_1 - h_2)/(h_1 - h_{f2}) \end{aligned} \qquad (7)$$

where h_1 = enthalpy of steam leaving the superheater and entering the turbine, pressure p_1

h_2 = enthalpy of exhaust steam after isentropic expansion to the back pressure p_2

h_{f2} = enthalpy of saturated water at pressure p_2

The calculation of the efficiency of the Rankine cycle may be illustrated by Example 12.

EXAMPLE 12: Compute the efficiency of the Rankine cycle if steam is supplied at 400 psig and 720°F with exhaust at atmospheric pressure 14.7 psia.

SOLUTION: From the Mollier diagram (Fig. 8-2) or superheated-steam table at 414.7 psia and 720°F, h_1 = 1373 Btu per lb. After isentropic expansion to 14.7 psia, h_2 is found from the Mollier chart to be 1075 Btu per lb. At 14.7 psia h_{f2} is given as 180 Btu per lb in steam table 8-2. Then

Heat supplied in boiler and superheater = 1373 − 180 = 1193 Btu/lb
Heat converted into work in turbine = 1373 − 1075 = 298 Btu/lb
Heat rejected to cooling water = 1075 − 180 = 895 Btu/lb

Efficiency = 298/1193 = 25 per cent

The theoretical steam rate = 3,413/(298) = 11.45 lb per kwhr

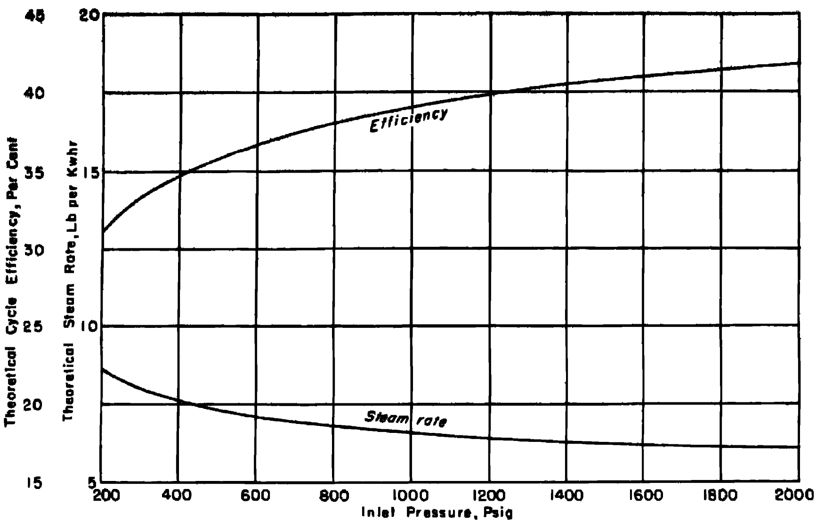

FIG. 8-5. Effect of inlet pressure on steam rate and cycle efficiency.

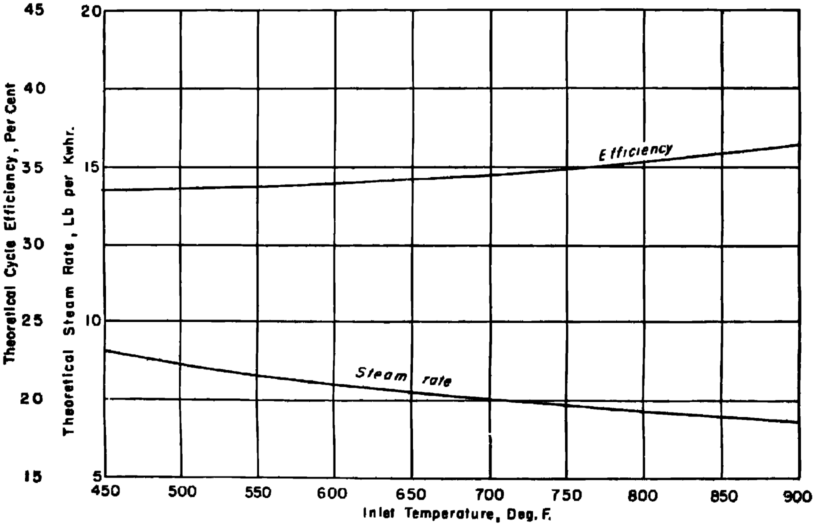

FIG. 8-6. Effect of steam temperature on steam rate and cycle efficiency.

The effect on the efficiency of the Rankine cycle of changing the back pressure from 14.7 psia to 2 in. Hg abs may be illustrated by Example 13.

EXAMPLE 13: Compute the efficiency of the Rankine cycle for the same data as in Example 12, except that the back pressure is 2-in. Hg abs.

SOLUTION: Upon isentropic expansion from 414.7 psia and 720°F. to 2 in. Hg, h_2 is found from the Mollier chart (Fig. 8-2) to be 918 Btu per lb. Also, at 2 in. Hg, h_{f2} is 69 Btu per lb. Then

Heat supplied = 1373 − 69 = 1304 Btu/lb
Work done = 1373 − 918 = 455 Btu/lb

Heat rejected in condenser = 918 − 69 = 849 Btu/lb
Efficiency = 455/1,304 = 34.9 per cent
Theoretical steam rate = (3,413)/(455) = 7.5 lb/kwhr

A comparison of the results of Examples 12 and 13 shows the improvement in efficiency and reduced steam rate which results from a reduction in the exhaust pressure. Figures 8-4, 8-5, and 8-6 illustrate the effect on steam rate and efficiency of changing the exhaust pressure, the initial pressure, and the initial temperature of the Rankine cycle from the conditions specified in Example 13.

INDUSTRIAL-POWER-PLANT CYCLES

Where electric power is the only output of the industrial plant, it is generally more economical to purchase power than to generate it unless low-cost or by-product fuel is available or the plant is large enough to justify the refinements of design found in central-station plants. Most industrial plants not only generate power but also furnish steam for space heating and process loads. Most of the energy in h-p steam cannot be converted into work in even the most modern power plants as indicated by the second law of thermodynamics However, by proper arrangement of the cycle, most of the energy in the exhaust steam may be used for heating or process purposes in industrial plants having a good balance between power and heating requirements. Under such conditions, much of the fuel cost as well as other boiler-plant costs are not chargeable to power, and the cost of such "by-product" power is quite low.

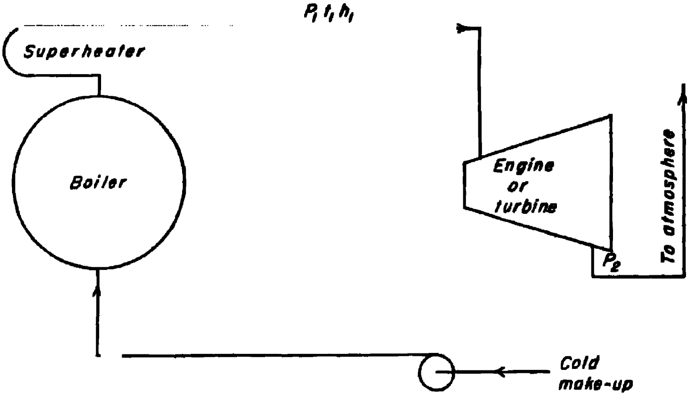

FIG. 8-7. Simple open cycle with exhaust to atmosphere.

The simplest power-plant cycle is shown in Fig. 8-7. Power is the only output, and most of the energy supplied in the steam-generating unit is wasted to the atmosphere. This cycle is used in portable plants and those burning waste fuel.

In Fig. 8-8, a direct-contact feed-water heater has been added to this cycle, and part of the energy in the exhaust steam is recovered in the heater, thus reducing the loss to the atmosphere. In Fig. 8-9, a heating or process load is added to the cycle of Fig. 8-8. If this load is too low to absorb all the exhaust steam, energy is wasted to the atmosphere. If the process load requires more steam than is needed by the turbine to generate the electrical load, a pressure-reducing valve is provided to by-pass steam around the turbine.

Figure 8-10 shows a cycle with an automatic-extraction turbine which exhausts to a condenser at low pressure and to a process load at an intermediate pressure. This cycle is used where the steam required for power is in excess of the steam required for process during a large portion of the total time of operation. Instead of wasting the energy in the surplus exhaust steam to the air, as in the cycle of Fig. 8-9, the sur-

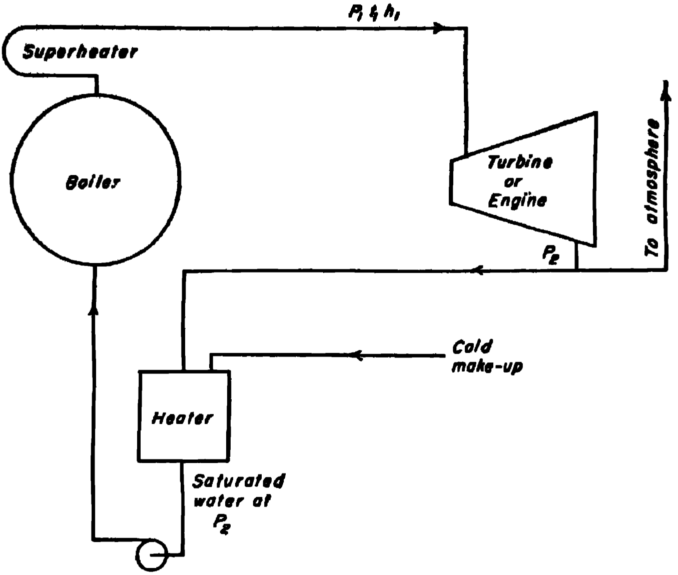

FIG. 8-8. Open cycle with direct-contact feed-water heater.

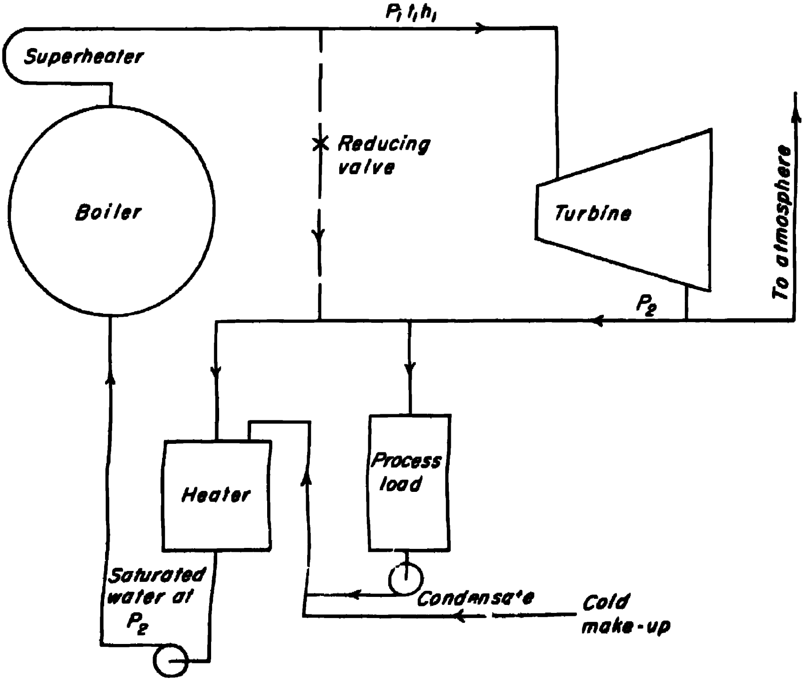

FIG. 8-9. Noncondensing cycle with process steam load.

plus steam is expanded to the condenser pressure, thereby doing more work. The loss of low-temperature energy is reduced.

The comparative performance of the various cycles illustrated in Figs. 8-3 and 8-7 to 8-10 is shown in Table 8-5 for constant initial steam conditions and a uniform power load of 1,000 kw. The amount of heat rejected to the atmosphere or to condenser cooling water should be compared. Also, in the cycles of Figs. 8-9 and 8-10, the process load could be so heavy that insufficient steam is available from the turbine, and h-p steam must be throttled to the process load. Then the energy rejected to

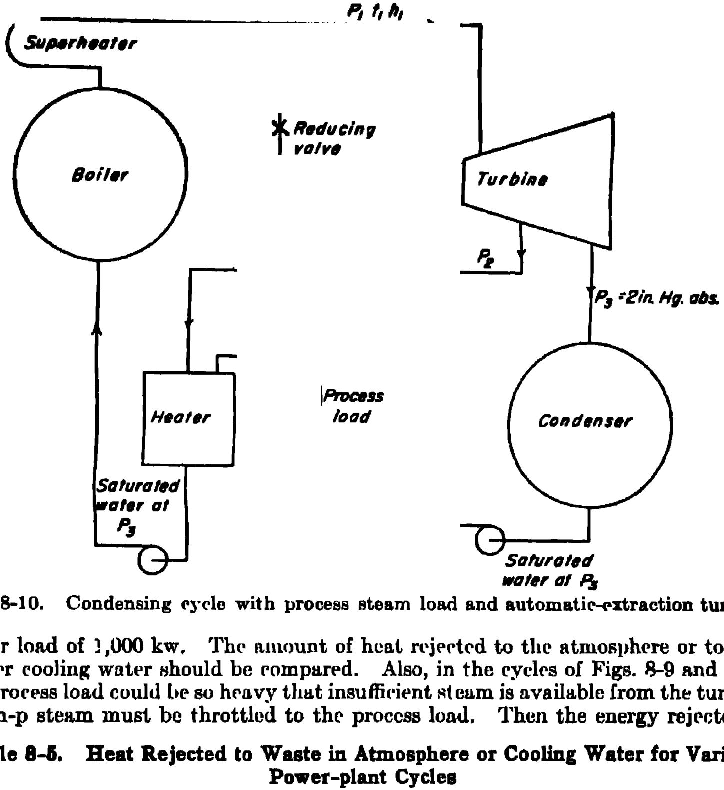

Fig. 8-10. Condensing cycle with process steam load and automatic-extraction turbine.

Table 8-5. Heat Rejected to Waste in Atmosphere or Cooling Water for Various Power-plant Cycles

Conditions: p_1 = 400 psig; t_1 = 720°F; isentropic expansion in turbine; power output of turbo unit = 1,000 kw; saturated water from condenser and direct-contact heater; 25°F subcooling of condensate from process loads; 60°F make-up.

Cycle	P_2 absolute	Process load, Btu per hr	Heat supplied in boiler and superheater, Btu per hr	Work rate of turbine, Btu per hr	Heat absorbed in feed-water heater, Btu per hr	Heat rejected to atmosphere or cooling water, Btu per hr
Fig. 8-3	14.7 psi	....	13,650,000	3,413,000	....	10,250,000
Fig. 8-3	2 in. Hg	....	9,760,000	3,413,000	....	6,350,000
Fig. 8-7	14.7 psi	....	15,400,000	3,413,000	....	11,990,000
Fig. 8-8	14.7 psi	....	13,650,000	3,413,000	1,740,000	8,510,000
Fig. 8-9	14.7 psi	2,000,000	13,650,000	3,413,000	1,270,000	7,980,000
Fig. 8-9	14.7 psi	5,000,000	13,650,000	3,413,000	860,000	4,390,000
Fig. 8-9	14.7 psi	9,000,000	13,650,000	3,413,000	410,000	840,000
Fig. 8-10	40 psi	2,000,000	10,400,000	3,413,000	1,130,000	5,010,000
Fig. 8-10	40 psi	5,000,000	12,300,000	3,413,000	1,090,000	3,690,000
Fig. 8-10	40 psi	9,000,000	14,000,000	3,413,000	415,000	1,910,000

the atmosphere or condenser is zero. (Actually, a small amount of steam must flow through the l-p section of the extraction turbine to keep it from overheating.)

Some industrial-power-plant cycles become very complex. Steam may be bled from an automatic extraction turbine at two different pressures instead of one. In some plants, a variable amount of steam at an intermediate pressure may be available as exhaust steam from steam hammers or steam engines and a mixed-pressure turbine may be used. This turbine operates like an automatic-extraction turbine except that steam at intermediate pressure, instead of being extracted, is supplied to the turbine and expanded to condenser pressure. Steam-engine-driven auxiliaries may operate on h-p steam and exhaust to a feed-water heater or a process load. In all cases, the ideal condition is one in which the heating and process load is supplied by steam which has generated power and in which the loss of low-temperature energy to condenser cooling water or atmosphere is reduced to a minimum.

FUELS*

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The energy demands of the United States for industrial and utility steam-power purposes are met by the use of solid, liquid, and gaseous fuels—chiefly by bituminous coal. The approximate percentages contributed by each type of fuel to satisfy these requirements are bituminous coal, 70 per cent; anthracite coal, 2 per cent; fuel oil, 13 per cent; and natural gas, 15 per cent.

TOTAL MARKETED PRODUCTION

Bituminous Coal. In recent years, the production of bituminous coal has varied from a low of approximately 300 million tons in 1932 to over 600 million tons in 1944.

Table 8-6. Production of Bituminous Coal

	Per cent
West Virginia	27
Pennsylvania	24
Illinois	13
Kentucky	12
Ohio	5
Virginia	3
Alabama	3
Indiana	2
All others	11
	100

Table 8-7. Petroleum Refining Areas

	Per cent
California	29.5
Gulf Coast (Texas and Louisiana)	24.8
East Coast	17.3
Indiana, Illinois, and Kentucky	12.3
Texas, Arkansas, Louisiana inland	6.5
Oklahoma, Kansas, and Missouri	5.2
Rocky Mountain	2.4
Appalachian	2.0
	100.0

While practically half the 48 states produce bituminous coal, 89 per cent of the total is from eight states—all located east of the Mississippi River. The production is shown in Table 8-6.

Anthracite Coal. The production of anthracite is approximately 10 per cent that of bituminous, and of this about 85 per cent is used in domestic and commercial heating equipment. Practically all of it originates in Pennsylvania. In 1944, the total was 61 million tons.

Residual Fuel Oil. Production of residual fuel oil in 1944 amounted to 461.4 million barrels and was distributed among the several petroleum refining areas, in order of magnitude, as shown in Table 8-7.

Table 8-8. Production of Natural Gas

	Per cent
Texas, Louisiana, and Oklahoma	63.8
California	13.5
West Virginia, Kentucky, Pennsylvania, Ohio, New York	11.4
All others	11.3
	100.0

Natural Gas. The total marketed production of natural gas for domestic, commercial, and industrial fuel in 1944 was 2,486 billion cu ft. The total production was distributed as shown in Table 8-8.

DISTRIBUTION OF FUELS

The characteristics of the fuels economically available are limited in the first place by the location of the consuming plant. Because of the relatively high cost of overland transportation per unit of energy, nearly all coal is consumed within 300 miles of its place of production or at more distant points to which a substantial part of the transportation is by water. For the same reason coal from one producing district is unable to compete successfully with coal produced nearer to a given point of consumption, unless favored by distinctly lower production costs or by chemical and physical properties of the coal which make it better adapted to specific uses. This has resulted in a well-defined pattern of distribution which is described in detail below.

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* Superior numbers refer to specific references listed at the end of this subsection.

The cost of transportation of fuel oil by pipe line or tanker and of natural gas by pipe line is considerably lower than for coal per mile per unit of energy. Thus both these fuels have been able to move from three to five times as far as coal to reach consuming areas to which suitable transportation facilities are available. The availability of both oil and gas at a given point of consumption, however, is determined by the cost of secondary distribution to the consumer. Gas, of course, can be delivered only to a service-connected consumer who is located either near a trunk pipe line or within the limits of a city gas-distribution system.

Crude oil or the lighter fuel oils can be transported by pipe line or by barge or tanker, but the heavy fuel oil commonly used by industry for steam generation cannot be transported by pipe line because of its high viscosity. Hence, heavy fuel oil is ordinarily marketed only within a short radius of refineries or waterside terminals from which it can move overland only by truck or railroad tank car. Generally such overland movement in the industrial area east of the Mississippi River is in the direction of some producing coal field, so that the further the oil moves, the greater is its cost to the consumer, while the cost of coal becomes less. The light oils can be transported overland by pipe line to secondary centers of distribution and consequently are economically available over a wider area, especially since these grades of oil are usually in competition only with the more costly domestic sizes of coal for residential heating or for use in diesel engines.

The result is that a large part of the total fuel-oil consumption is concentrated along the North Atlantic seaboard to which oil can be transported cheaply by tanker and which is most distant from the coal-producing fields of all the territory in the northern states east of the Mississippi.

A comparatively small part of the total fuel oil used in stationary plants or for residential heating is consumed near the oil fields because natural gas is usually produced in or near the same areas and costs much less per unit of energy at the point of production.

In summary, in 1941, 85 per cent of all the coal consumed in the United States for all purposes, except railroad fuel, was used in the northern industrial states east of the Mississippi River. In the same area in that year 70 per cent of all the fuel oil was used, and three-quarters of this was in the New England and Middle Atlantic states. On the other hand, 70 per cent of the natural gas used for fuel was consumed in the territory to the south and west of the preponderantly coal-burning area, and 83 per cent of this was used west of the Mississippi River.

Natural gas and the several grades of fuel oil all have substantially the same physical and chemical properties everywhere. On the other hand, the characteristics of the coal economically available at different locations relatively near each other in the Northeast (where most of the coal is consumed) vary in many important respects depending upon the location of the mines. Therefore, the pattern of fuel distribution in this area is of primary importance. This distribution pattern is described and the typical range of the chemical and physical properties of the coal available in each subdivision of the northeastern part of the country is given in the following material.

The industrial Northeast can be divided into six market areas, within any one of which the coals economically available for industrial use originate in only a few producing districts and have a distinctive range in their more important physical and chemical properties. These six divisions are shown on the accompanying map, Fig. 8-11. It should be noted, however, that the boundaries of these consuming areas have been determined primarily by the relative delivered cost of coal from the different producing districts and the quality of coal from competing producing districts. Neither is unchangeable, so that, with the passage of time and changing conditions, they will vary somewhat. Also these boundaries are not impassable, so that within a short distance of a boundary, coals available in the adjoining consuming area should be considered. For special purposes requiring a quality of coal not ordinarily economical for steam generation at a given location, a source of supply at some distance may have to be chosen.

Consuming Area I. Consuming area I embraces that part of eastern and southern New England adjacent to the seacoast and most distant by direct rail haul from any coal-producing field. The coal for industrial use comes almost entirely from the low- and medium-volatile fields of southern West Virginia by rail to tidewater at Hampton Roads and then to New England ports by collier. The typical quality characteristics of these coals are moisture, 2.5 per cent; volatile, 16 to 23 per cent; ash, 6.5 per cent; Btu, 14,300. The softening temperature of ash for coals from these fields varies widely but is usually found between 2400 and 2800°F. The grindability-index values are generally in the higher ranges, 80 to 95 on the ball-mill scale or 100 to 115 Hardgrove equivalent.

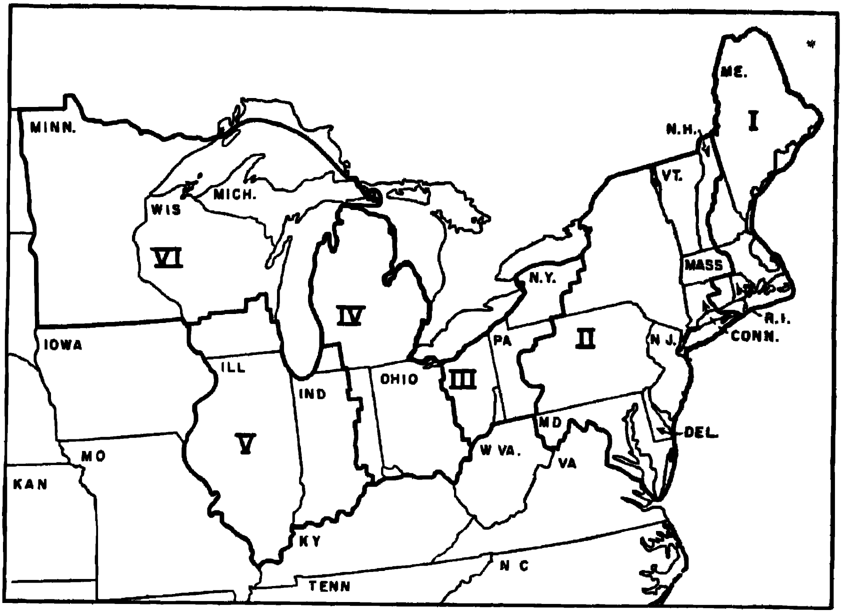

FIG. 8-11. Northeastern coal-consuming areas.

NOTE: These coal-quality figures and those in similar tables which follow represent a quality for industrial-steam sizes of coal that corresponds to the major part of production in each district. The quality of coal shipped by individual mines will vary considerably above and below these figures, in some cases by a substantial amount.

Consuming Area II. Consuming area II includes the western portion of New England, the eastern sections of New York and Pennsylvania, and all New Jersey, Delaware, and Maryland. Within this area lie the anthracite fields of Pennsylvania, and the low-, medium-, and high-volatile bituminous fields of central Pennsylvania and Maryland. Almost the entire output of these producing fields is consumed within this area. Immediately adjacent are the high-volatile bituminous fields of northern West Virginia, and a substantial part of its output is also shipped to this consuming area.

Industrial consumers in this area (except at points close to the mines) have an exceptionally wide range in the properties of coal available, from the small sizes of anthracite through the whole range of volatile content, softening temperature of ash, friability, grindability, and sulfur found in bituminous coal produced in the whole Appalachian region. Along the coast, heavy fuel oil is also available. Fuel selection and the design of steam-generating equipment in this area require careful study, depending upon the exact location of the plant.

The typical quality and character of the coals available are shown in Table 8-9.

Table 8-9

	Moisture, per cent	Volatile, per cent	Ash, per cent	Btu
Central Pennsylvania	2.5	18–23	8.0	14,000
Central Pennsylvania	2.5	30	8.0	13,800
Northern West Virginia (high volatile)	3.0	35	9.0	13,300

In the central Pennsylvania fields, the ash-softening temperature of the various classes of coal will vary from 2200 to 2900°F. The lower volatile coals are usually found easy to pulverize, having indexes of from 85 to 95 on the ball-mill scale, 105 to 115 on the Hardgrove. The higher volatile coals are somewhat harder, with indexes of from 60 to 75 ball-mill scale, 80 to 95 Hardgrove.

The northern West Virginia high-volatile coals have ash-softening temperatures of 2000 to 2300°F. Their grindability indexes range from 45 to 55 ball-mill scale, 60 to 75 Hardgrove.

Consuming Area III. Almost all points within consuming area III, which includes the western parts of New York State and Pennsylvania and the eastern part of Ohio, lie within a relatively short distance of mines which are themselves within the area. The industrial coal consumed in the area originates chiefly in western Pennsylvania and eastern Ohio. To a lesser extent it originates in the high-volatile fields of northern West Virginia adjoining it on the south.

The typical quality of these coals is shown in Table 8-10.

Table 8-10

	Moisture, per cent	Volatile, per cent	Ash, per cent	Btu
Western Pennsylvania	3.0	32.0	9.0	13,400
Western Pennsylvania	4.0	37.0	9.0	13,050
Eastern Ohio	5.0	37.0	10.0	12,600

The softening temperature of the ash in these coals is usually in the low or medium range, 2000 to 2500°F. Grindability values are from 40 to 50 ball-mill scale, 55 to 70 Hardgrove.

Consuming Area IV. Consuming area IV, which includes the western part of Ohio, the eastern part of Indiana, and the southern peninsula of Michigan, draws its supply of industrial coal delivered by all-rail haul almost entirely from the southern West Virginia and eastern Kentucky producing fields (chiefly the high-volatile mines), except in northwestern Ohio where Ohio coal is also economically available. At Michigan Great Lakes ports, these coals are available, as well as coals from Ohio and western Pennsylvania, and the high-volatile coals from northern West Virginia shipped by water from Lake Erie loading piers.

The typical quality of the coal available in this area (except for those already described as being available in other consuming areas) is moisture, 3.0 per cent; volatile, 34 to 37 per cent; ash, 6.5 per cent; Btu, 13,800.

The softening temperatures of ash in these coals vary over a wide range but are usually found in the medium or high range of 2400 to 2900°F. Their grindability indexes are from 35 to 50 ball-mill scale, 50 to 70 Hardgrove.

Consuming Area V. In consuming area V, which includes western Indiana, all of Illinois, and the southern part of Wisconsin, about two-thirds of the coal used by industry originates in the Indiana and Illinois mines. Nearly all the remainder comes from the high-volatile fields of southern West Virginia and eastern Kentucky. Some western Kentucky coal is also shipped into this area.

The typical quality of these coals, in addition to those already described, is shown in Table 8-11.

Table 8-11

	Moisture, per cent	Volatile, per cent	Ash, per cent	Btu
Illinois and Indiana	9.0	34.0	9.0	11,800
Illinois and Indiana	13.0	34.0	10.0	10,900
Western Kentucky	7.0	37.0	10.0	12,000

The ash-softening temperatures for these coals fall in the low and medium range, 2000 to 2400°F. The grindability-index range is 35 to 50 ball-mill scale, 50 to 70 Hardgrove.

Consuming Area VI. In consuming area VI, which includes the upper peninsula of Michigan, most of Wisconsin (from Milwaukee north), and all of Minnesota, the variety of coal available for industrial use is similar in its wide range to that encountered in consuming area 2, because a large part of the coal is transported via the Great Lakes from Lake Erie ports through which coal can be economically shipped from central and western Pennsylvania, both northern and southern West Virginia, and eastern Kentucky and Ohio. Illinois coal is also shipped into this area by rail and river. Fuel selection in this area must be made with particular reference to the fuels available at the exact location where they are to be used. The quality of coal shipped from all the producing fields that supply this area has been described above under the headings of the consuming areas which are principally dependent upon them.

Southeastern Consuming Area. In the territory to the south of that shown on Fig. 8-11, coal plays a minor part, except in Virginia, North and South Carolina, and Georgia, where bituminous coal is economically available from the southern West Virginia fields described under consuming areas I and IV and from the Alabama and Tennessee producing fields. Western Kentucky coal for industrial use is chiefly distributed in Kentucky and Tennessee. The typical quality of these coals is moisture, 3.0 per cent; volatile, 31 to 35 per cent, ash, 8.0 per cent, Btu, 13,500.

The softening temperature of ash for these coals is from 2100 to 2900°F. The grindability indexes are 35 to 55 ball-mill scale, 50 to 75 Hardgrove.

Western Consuming Area. West of the Mississippi, the coal-producing fields are relatively small and widely scattered. The coal is mainly distributed to market areas local to each producing field.

The rank of the coal from these fields shows a wide variation and covers groups from low-volatile bituminous to lignite. The various ranks of coal produced in 12 states are listed in Table 8-12 in the order of tonnage produced. The combined production of these states is approximately 10 per cent of the nation's tonnage. Reference should be made to Table 8-14 for typical analytical data pertaining to the rank.

Table 8-12

State	*Rank of bituminous produced*
Wyoming	High-volatile A, B, C, subbituminous A, B
Colorado	High-volatile A, B, C, subbituminous A
Utah	High-volatile A, B, C
Montana	High-volatile B, C, subbituminous A, B, C
Missouri	High-volatile B, C
Kansas	High-volatile A, B
Oklahoma	Low and medium volatile
North Dakota	Lignite
Iowa	High-volatile C, subbituminous C
Arkansas	Low-, medium-, and high-volatile A
New Mexico	High-volatile A, C
Washington	High-volatile A, B, C, subbituminous A, B

Table 8-13. Classification of Coals by Rank[*]

Legend: F.C. = fixed carbon; V.M. = volatile matter; Btu = British thermal units.

Class	Group	Limits of fixed carbon or Btu mineral-matter-free basis	Requisite physical properties
1. Anthracitic	1. Metaanthracite	Dry F.C., 98 per cent or more (dry V.M., 2 per cent or less)	
	2. Anthracite	Dry F.C., 92 per cent or more and less than 98 per cent (dry V.M., 8 per cent or less and more than 2 per cent)	
	3. Semianthracite	Dry F.C., 86 per cent or more and less than 92 per cent (dry V.M., 14 per cent or less and more than 8 per cent)	Nonagglomerating[†]
2. Bituminous[§]	1. Low-volatile bituminous coal	Dry F.C., 78 per cent or more and less than 86 per cent (dry V.M., 22 per cent or less and more than 14 per cent)	
	2. Medium-volatile bituminous coal	Dry F.C., 69 per cent or more and less than 78 per cent (dry V.M., 31 per cent or less and more than 22 per cent)	
	3. High-volatile A bituminous coal	Dry F.C., less than 69 per cent (dry V.M., more than 31 per cent); and moist[‡] Btu, 14,000[‖] or more	
	4. High-volatile B bituminous coal	Moist[‡] Btu, 13,000 or more and less than 14,000[‖]	
	5. High-volatile C bituminous coal	Moist Btu, 11,000 or more and less than 13,000[‖]	Either agglomerating or nonweathering[¶]
3. Subbituminous	1. Subbituminous A coal	Moist Btu, 11,000 or more and less than 13,000[‖]	Both weathering and nonagglomerating
	2. Subbituminous B coal	Moist Btu, 9500 or more and less than 11,000[‖]	
	3. Subbituminous C coal	Moist Btu, 8300 or more and less than 9500[‖]	
4. Lignitic	1. Lignite	Moist Btu, less than 8300	Consolidated
	2. Brown coal	Moist Btu, less than 8300	Unconsolidated

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[*] This classification does not include a few coals which have unusual physical and chemical properties and which come within the limits of fixed carbon or Btu of the high-volatile bituminous and subbituminous ranks. All these coals either contain less than 48 per cent dry mineral-matter-free fixed carbon or have more than 15,500 moist mineral-matter-free Btu.

[†] If agglomerating, classify in low-volatile group of the bituminous class.

[‡] Moist Btu refers to coal containing its natural bed moisture but not including visible water on the surface of the coal.

[§] It is recognized that there may be noncaking varieties in each group of the bituminous class.

[‖] Coals having 69 per cent or more fixed carbon on the dry, mineral-matter-free basis shall be classified according to fixed carbon, regardless of Btu.

[¶] There are three varieties of coal in the high-volatile C bituminous coal group, *viz.*, variety 1, agglomerating and nonweathering; variety 2, agglomerating and weathering; variety 3, nonagglomerating and nonweathering.

PROPERTIES OF FUELS

Coal. Coal is classified by rank, as shown in Table 8-13, as adopted by the ASTM under Standard Designation D-388-38. Within the limits for each group set by this scheme, there will be found in Table 8-14 representative proximate analyses typical of each one, using the normal seam-moisture content.

For industrial steam-generating purposes, the rank and grade of coal is customarily judged by the use of the proximate analysis, which includes moisture, volatile, ash, sulfur, and fixed carbon. Other physical properties that may affect the efficient recovery of the energy are commonly determined. These are calorific value, ash-softening temperature, grindability, and coking property. The facts disclosed by the proximate analysis and the effects of the various other properties both physical and chemical are briefly described.

Table 8-14. Proximate Analyses of Groups of Coal

Group or rank	Moisture	Volatile	Ash	Btu
Metaanthracite	2.0	2.0	20.0	10,925
Anthracite	4.0	3.5	10.0	12,775
Anthracite	4.0	6.0	10.0	12,900
Semianthracite	4.0	10.0	10.0	13,150
Low-volatile bituminous	2.0	16.0	7.0	14,325
Low-volatile bituminous	2.5	19.0	7.0	14,200
Medium-volatile bituminous	2.5	23.5	7.0	14,200
Medium-volatile bituminous	2.5	29.0	8.0	13,850
High-volatile bituminous A	3.0	33.0	9.0	13,400
High-volatile bituminous A	4.0	37.0	9.0	13,050
High-volatile bituminous B	6.0	37.0	9.0	12,500
High-volatile bituminous B	9.0	34.0	9.0	11,800
High-volatile bituminous C	12.0	34.0	10.0	11,050
High-volatile bituminous C	15.0	34.0	10.0	10,550
Subbituminous A	17.0	34.0	11.0	9,900
Subbituminous B	20.0	33.0	11.0	8,700
Subbituminous C	24.0	33.0	11.0	8,000
Lignite	37.0	26.0	7.0	6,800

Moisture. Moisture is present in coal in two forms, inherent and free (surface) water. The inherent or seam moisture is considered a natural part of the coal substance in its virgin state. Coal containing only this type of moisture does not have the appearance of wetness and may actually be dusty with relatively high percentages of water depending on the rank of the coal. The analyses given in Table 8-14 report this type of moisture.

After mining, the total moisture content may increase or decrease depending not only upon the wetting or drying due to climatic conditions during transportation and storage but also as a result of cleaning processes using water and the addition of water to minimize dust. This is the free water which is mechanically retained on the surface of the coal particles or between them and is visible or apparent to the touch.

The total moisture content of coal affects the efficiency of a steam-generating plant to only a slight extent. The loss is of the order of 0.1 per cent in efficiency for each 1.0 per cent of moisture. Excess surface moisture may have a very considerable adverse effect on pulverizer capacity, particularly when operating in a range close to the maximum. High surface moisture has a noticeable influence on the coking property of certain coals, particularly the high-rank, low-volatile coals with low inherent moisture. With certain types of stoking equipment, these coals often form a dense coke mat which can be definitely modified by the addition of water to the coal as fired.

Volatile and Fixed Carbon. The volatile content of coal, excluding moisture, is that part driven off by heating and is burned as a gas. The remaining combustible solid matter is the fixed carbon. Relatively small differences in the per cent of volatile or fixed carbon are usually of little practical significance to the steam-plant operator. There are, however, circumstances under which it is necessary or desirable to specify maximum or minimum permissible limits.

For hand- or stoker-fired plants with furnaces having inadequate space to mix the volatile matter with air for combustion, it is found important to specify an upper limit of volatile to minimize the loss of heating value or, of more likely importance, the emission of smoke. Under a given set of conditions, it is a reasonable presumption that the amount of smoke is proportional to the per cent of volatile, although it is often modified by the size consist of the coal and its coking property. Conversely, in pulverized-coal firing, it is often necessary to set a low limit in order to maintain flame propagation, particularly in completely water-cooled furnaces.

The varying composition of the volatile matter in coal is shown by data taken from tests made by the U.S. Bureau of Mines in determining the carbonizing and by-product properties of different types of coals. Table 8-15 gives the percentages of several of the volatile constituents in per cent of coal. The volatile matter of anthracite contains no tarry constituents, and the gas is approximately 85 per cent hydrogen.

Table 8-15. Volatile Constituents in Coal

Type of coal	Tar, per cent	Hydrocarbons in gas, per cent	Btu per cu ft of gas
Low-volatile bituminous	3.0	32	545
Medium-volatile bituminous	5.0	35	570
High-volatile bituminous A	6.5	37	605
High-volatile bituminous B	6.5	37	610
High-volatile bituminous C	6.0	32	555
Subbituminous B	3.0	22	450
Lignite	0.2	9	325

Ash. Ash, or the noncombustible mineral matter remaining after coal has been burned, is composed largely of compounds of silicon, aluminum, iron, calcium, magnesium, and the alkalies. Very small quantities of sulfates and phosphates are of common occurrence. While probably more than half the known chemical elements have been reported as occurring in coal, they have been too low in percentage to be of importance for recovery.[1] The typical limits of the common elements in coal ash are shown in Table 8-16.

Table 8-16

	Per cent
Silicon dioxide	40-60
Aluminum oxide	20-35
Ferric oxide	5-25
Calcium oxide	1-15
Magnesium oxide	½-4
Alkali oxides	1-4

The mineral matter associated with coal is from two sources, inherent and adventitious. Inherent mineral matter is usually small in amount in American coals, *i.e.*, less than 1 or 2 per cent and undetectable except as a residue of combustion. Its origin is the mineral matter of the living plant life that went to form the coal measure. The adventitious or extraneous ash is from clays, shales, pyrites, or other minerals occurring above, below, or as partings within the coal seam. These minerals may appear as isolated pieces of various sizes or as finely disseminated particles. Much of this extraneous ash can be and is removed by selective mining and various types of cleaning processes.

Apart from the economics of coal purchasing in which ash is bought and transported at the same rate as coal, the quantity present will have a significant effect on power-plant costs, refuse handling, and boiler and stoker upkeep. Increases in cost of ash handling and boiler and furnace upkeep will vary widely depending upon the individual plant and the amount of equipment installed for these purposes. An increase in ash content from 5 to 10 per cent requires the use of about 5 per cent more coal for a given amount of steam, while the quantity of refuse to be handled will be slightly more than doubled.

Sulfur. Sulfur is present in coal in three forms, an insignificant amount combined as sulfate, in an organic form combined with the coal substance, and as iron pyrites, the last assuming the greatest importance because of certain objectionable side effects introduced by the associated iron. Pyrites (or marcasite) is the disulfide of iron which in the combustion process can be broken down to ferric oxide and the oxides of sulfur or, under certain conditions, ferrous sulfide.

This compound easily fusible itself or in combination with other elements of the ash promotes the formation of clinkers or slags, and it is the likelihood of this occurring that creates in the mind of the user the thought that high sulfur content is responsible for the difficulty. The relationship between sulfur content and ash fusibility is very general as shown by a study of about 5,000 tests made on coals from the Appalachian region. These tests indicated that 70 per cent of the coals having 2 per cent or more of sulfur had ash fusing below 2500°F, while of those having 1½ per cent or less, 70 per cent fused over 2600°F. Coal from a particular mine, however, may be very low in sulfur content and still have a low softening temperature of ash.

Another difficulty relative to use of high-sulfur coal, aside from that of air pollution by the oxides of sulfur, is the corrosive nature of these products of combustion in the flue gases in the presence of water vapor. Relatively low temperature of the flue gases, *i.e.*, below 350°F, favors the formation of sulfuric acid, and as a consequence corrosion of the metal parts of economizers, air heaters, flues, etc., can be expected with correspondingly high maintenance costs.

Ash-softening Temperature. Coal ash, being a complex mixture of mineral compounds, does not have a definite melting point but a range or interval. One point in this interval, defined as the "softening temperature," has been used for many years as an indication of the tendency to form clinker and slag. The test procedure in use today is the result of the fundamental work of Fieldner, Hall, and Feild in 1918.[2] While other factors affect clinker formation, the softening temperature of ash as determined in the laboratory is a very useful figure in predicting the probable behavior of the ash residues from coal when burned either on grates or in pulverized form. There are other means of determining clinkering and slagging tendencies, but unfortunately most of them are either so expensive or laborious as to preclude their commercial use.

The present standard test is to pulverize the ash finely and mold it into a cone. This is mounted on a refractory base and subjected to a controlled rate of heating in an atmosphere conforming approximately to a composition of 60 per cent of oxidizing gases and 40 per cent of reducing. As the heating of the cone progresses, observations are made concerning three points of the melting interval. The first point is when there is a slight rounding of the sharp tip of the cone. This is noted as the temperature of initial deformation. The heating is continued, and when the cone deforms to spherical shape approximately one-half its original height, that point is noted as the softening temperature. The fluid temperature is the point at which the ash assumes such a high degree of fluidity that it flows over the refractory base. In many instances, all three of these points are important. The one most commonly used is the softening point (sometimes called "fusion temperature"), and it is understood to be the one used in analytical results in the absence of the other two.

One reason why the ash-softening temperature as determined in the laboratory does not always agree with the practical observation in the plant is the intimate mixing and pulverizing of the ash in the laboratory sample which does not occur in practice. The ash in coal may be either a very finely divided material in the coal substance or occur as sizable pieces of slate, shale, or pyritic matter which may or may not be evenly distributed throughout the coal mass. These different portions of the coal ash are likely to have different softening temperatures. There is always the possibility of certain isolated pieces fusing and forming the nucleus of a clinker or slag. The work done in connection with this problem by Gould and Brunjes in 1940[3] pointed out the importance of separating the material to determine the proportion of fusible and refractory material in both size and gravity fractions.

In this work, one instance was cited of a coal ash having a 2700°F softening temperature by standard test with about 73 per cent of ash fusing over 2600°F and 24 per cent fusing under 2200°F. If, in firing, the lower fusing material were concentrated in any portion, the expectancy would be for clinker trouble to occur; yet the standard test indicated refractory ash. This same coal burned in pulverized form, *i.e.*, ground to pass a 200-mesh sieve, had in physically separable portions of the coal about 15 per cent showing ash-fusion temperatures of less than 2300°F and 85 per cent fusing over 2700°F.

Grindability. The grindability index of coal is the measure of its resistance to pulverization. It is measured by two methods in common use—the ball-mill method proposed by Yancey, Furse, and Blackburn[4] and the Hardgrove-machine method proposed by Hardgrove.[5] In the ball-mill index 100 signifies the softest or most easily pulverized coal, while in the Hardgrove index it is a standard reference point in the scale. On the latter scale an index figure of 118 would be comparable to its 100 value in the ball-mill index. Both methods appear as tentative standards in the ASTM with a conversion table. A curve plotted from equivalent values is given in Fig. 8-12.

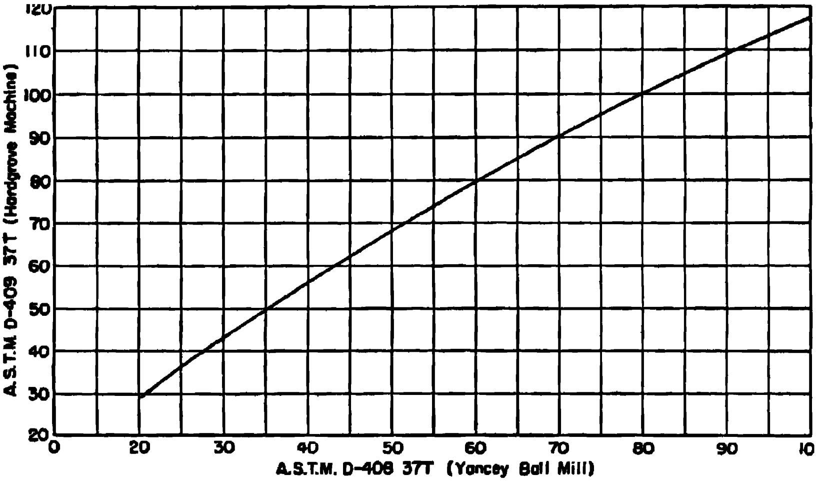

FIG. 8-12. Equivalent values of Hardgrove and ball-mill grindability indexes.

Coking Properties. The characteristic of coking has been and is still described by a number of terms, among them agglomerating, agglutinating, and swelling, all relating to the behavior of coal on heating. This property and its descriptive terminology are more important in the manufacture of coke than in burning coal for steam raising, but there are many instances when a convenient measure of the physical reaction of coal to heat in the furnace is desirable. In this connection, the visual observation of the coke mass after determining the volatile together with an estimation of its cohesiveness has been useful and is used as an indication of the rank of coals when fresh-mined samples are used. A grading proposed by Gilmore, Connell, and Nicolls[6] was based on this method and classified coals as noncoking, poor, fair, and good coking.

Within recent years, a comparatively simple means of determining the swelling power has been adopted as a tentative standard by the ASTM. This is designated as the free-swelling index. The test consists in rapid heating of the finely ground coal in a closed crucible after which the profile of the resulting coke button is matched against standard outlines. Seventeen profiles are given in the standard in half numbers from one to nine. Number one represents a nonswelling coal with a coherent button, while number nine is one swollen to the point of pressing against the crucible cover, a vertical expansion of approximately five times. Granular or noncoherent residues indicate a noncoking or nonagglomerating class of coal.

As a rule, all fresh-mined bituminous coal above high-volatile bituminous C in rank will coke and swell to some degree—the higher the rank, the greater the swelling power. Weathering or oxidation greatly affects the coking or swelling property and may destroy it entirely. The free-swelling index is often used as a means of detecting oxidized coal. When the normal swelling power of the particular coal is known and tests show profiles significantly below this value, it is safe to assume that weathering or oxidation has taken place.

Calorific Value. The calorific value as expressed in British thermal units (Btu) is the one definite measure of the energy in coal, all other items of the analysis representing those properties hindering or enhancing its efficient recovery. The Btu is defined as 1/180 the amount of heat required to raise 1 lb of water from 32 to 212°F.

It is most important to determine the Btu by means of the calorimeter since there is no reasonably accurate means of calculating this value from the other figures given in the proximate analysis.

Industrial Sizes of Anthracite. Compared with bituminous, the range in characteristics of anthracite is extremely narrow. The tonnage for industrial use is in the Nos. 3, 4, and 5 buckwheat sizes prepared through and over round-hole screens of openings as shown in Table 8-17.

Table 8-17. Screen Openings for Buckwheat Coal

Size	Through, in.	Over, in.
No. 3 buck	3/16	3/32
No. 4 buck	3/32	3/64
No. 5 buck	3/64	

Size and quality specifications are set by the Anthracite Committee of Pennsylvania and as of Sept. 1, 1947, are controlled by a state law fixing anthracite standards on sizes larger than No. 3 buck. The standard specifications for Nos. 3, 4, and 5 buckwheat are shown in Table 8-18.

The smaller sizes of anthracite are generally prepared, screened, and loaded with the aid of water. Consequently, the moisture content will run comparatively high, increasing with decreasing size consist. As shipped, the typical moisture limits of Nos. 3, 4, and 5 buckwheat are as shown in Table 8-19.

Table 8-18. Standard Specifications for Buckwheat Coal

	Oversize maximum, per cent	Undersize, per cent		Maximum ash, dry basis, per cent
		Maximum	Minimum	
No. 3 buck	10	20	10	15
No. 4 buck	20	30	10	15
No. 5 buck	30	No limit		16

Fuel Oil. The ASTM specifications classify fuel oil into five grades, Nos. 1, 2, 3, 5, and 6, and the detailed requirements for each are given in Table 8-20. Of these grades only No. 6, also termed bunker C or residual oil, is commonly used as an industrial fuel for steam purposes. Grades 1, 2, and 3 are the distillate oils which are marketed chiefly for residential domestic heating and certain specialty uses in industry. The distillate oils under a somewhat different grading also supply the diesel fuel market. Number 5 oil is usually a heavy product blended with distillate to lower the viscosity. Its market is a semi-industrial one and includes commercial heating plants and specialty uses.

Table 8-19. Moisture Limits of Buckwheat Coal

	Per cent
No. 3 buck	7–9
No. 4 buck	9–11
No. 5 buck	10–12

Number 6 or residual oil is generally tested for compliance with the standard specification and also for gravity and Btu content. The significance of these tests to the consumer will be given in a brief way. For a more detailed explanation reference should be made to the work of the committees on petroleum products of the ASTM.

Gravity. As oil is sold by volume and as its volume will vary according to the temperature, it is essential that a temperature reading be made in conjunction with

Table 8-20. Detailed Requirements for Fuel Oils[a]

Grade of fuel oil[b]	Flash point, deg F		Pour point, deg F	Water and sediment, per cent by volume	Carbon residue, per cent by weight	Ash per cent by weight	Distillation temperatures, deg F: 10 per cent point	90 per cent point		End point	Viscosity, sec: Saybolt universal (at 100°F)		Saybolt furol (at 122°F)	
	Min	Max	Max	Max	Max	Max	Max	Max	Min	Max	Max	Min	Max	Min
No. 1 (A distillate oil for use in burners requiring a volatile fuel)	100 or legal	165	0	Trace	0.05[d] on 10 per cent residuum		410			560[e]				
No. 2 (A distillate oil for use in burners requiring a moderately volatile fuel)	110 or legal	190	10[c]	0.05	0.25[f] on 10 per cent residuum		400	600						
No. 3 (A distillate oil for use in burners requiring a low-viscosity fuel)	110 or legal	230	20[c]	0.10	0.15			675	600[g]		45			
No. 5 (An oil for use in burners requiring a medium-viscosity fuel)	130 or legal			1.00		0.10						50	40	
No. 6 (An oil for use in burners equipped with preheaters permitting a high-viscosity fuel)	150			2.00[h]									300	45

[a] Recognizing the necessity for low-sulfur fuel oils used in connection with heat-treatment nonferrous metal glass and ceramic furnaces, and other special uses, a sulfur requirement may be specified in accordance with the following table

Grade of fuel oil	*Sulfur, max, per cent*
No. 1	0.5
No. 2	0.5
No. 3	0.75
No. 5	No limit
No. 6	No limit

Other sulfur limits may be specified only by mutual agreement between the purchaser and the seller.

[b] It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[c] Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0°F under any conditions.

[d] The carbon residue on 10 per cent residuum may be increased to a maximum of 0.12 per cent when this oil is to be used in other than sleeve-type blue-flame burners. This limit may be specified by mutual agreement between the purchaser and the seller.

[e] The maximum end point may be increased to 590°F when this oil is to be used in other than sleeve-type blue-flame burners.

[f] To meet certain burner requirements, the carbon-residue limit on this oil may be reduced to 0.15 per cent on 10 per cent residuum.

[g] This requirement may be waived if the API gravity is 26 or lower.

[h] The amount of water by distillation plus the sediment by extraction shall not exceed 2.00 per cent. The amount of sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

the density determination so that delivery volumes can be calculated back to the standard basis of measurement which is the specific or API gravity at 60°F. For fuel oil, the API (American Petroleum Institute) gravity is customary. This scale is related to the specific gravity as follows:

$$\text{Degrees API} = \frac{141.5}{\text{sp gr } 60°\text{F}/60°\text{F}} - 131.5$$

Correction of the observed volume to the standard basis is made according to the National Standard Petroleum Oil Tables prepared by the U.S. Bureau of Standards. Lacking such tables, a convenient method sufficiently accurate for most purposes is offered by the use of a coefficient of expansion of 0.00035 per deg F at 60° for oils ranging up to 14.9 API gravity and covers No. 6 grade. For example, if the observed gallonage at 110°F is 10,000, the volume at 60°F would be

$$[1.0 - (50 \times .00035)] \times 10{,}000 = 9{,}825 \text{ gal}$$

Calorific Value. The Btu value which must be made by a calorimetric test is necessary for the determination of the efficiency of operation. It is related in a general way to the API gravity for grade 6 as indicated by the approximate heat content shown in Table 8-21.

Table 8-21

API at 60°F	Btu per lb	Btu per gal
8	18,100	152,910
10	18,300	152,400
12	18,400	151,100
14	18,500	149,830

Flash Point. The flash point is the temperature at which the oil gives off a vapor that is inflammable when mixed with air. It is an essential test, for it indicates what safety precautions are necessary in storing and handling oil.

Water and Sediment. Water and sediment are undesirable and should be kept at or below the minimum allowable in the standard specification. Water interferes with proper combustion and continuous flame propagation. Sediments plug strainers, valves, burners, etc., and may have an abrasive action.

Viscosity. Viscosity is a characteristic of oil which is the measure of its resistance to flow. It is determined by the efflux time of a given quantity through a specified orifice and is reported for grade 6 as seconds Saybolt Furol (SSF) at 122°F. It is one of the most important determinations because the viscosity indicates the degree of preheating necessary for complete atomization at the burner. This preheat temperature is generally accepted as being the point at which the oil has a viscosity of 150 seconds Saybolt Universal, or approximately SSF 15. As viscosity differs among oils and varies inversely with the temperature, which variation may be different between oils, it is necessary to determine the point of SSF 15 by means of a plot of two viscosity points on the ASTM Standard Viscosity-Temperature Charts for Liquid Petroleum Products. For many oils, the proper temperature for atomization in relation to viscosity SSF at 122°F is as shown in Table 8-22.

Table 8-22

SSF at 122°F	*Atomization temperature, deg F*
50	165
100	195
150	205
200	215
250	220
300	225

Natural Gas. Natural gas is generally quite pure and consists chiefly of methane, the per cent of which, while varying widely, is usually found to be between 75 and 99 per cent, the remainder being principally ethane with small quantities of nitrogen and carbon dioxide. The gross Btu value per cubic foot ranges from 900 to 1300. A typical analysis would be as shown in Table 8-23.

Table 8-23

Methane	91.0%
Ethane	3.0%
Carbon dioxide	0.5%
Nitrogen	5.5%
Btu	1025

Gas volumes when reported have usually been corrected to a standard temperature and pressure. On the English system that base is 60°F and 30 in. Hg. The normal atmospheric pressure at sea level is 29.92 in. Hg but for convenience this has been called in round figures 30 in. for all gas work.

Gas volumes vary in direct proportion to their absolute temperatures. The latter figure is obtained by adding 460 to the Fahrenheit temperature. Gas volumes are inversely proportional to the absolute pressure which is the pressure in inches of mercury column plus 30.

For any given set of conditions, the particular gas volume is corrected both for temperature and pressure as shown in the following example.

Example: Determine the volume at standard conditions (60°F and 30-in. Hg) of 1,500 cu ft of gas which was metered at 90°F and 3-in. Hg, the barometer standing at 28.56 in.

Solution: $1{,}500 \times \left(\frac{60 + 460}{90 + 460}\right) \frac{(28.56 + 3.0)}{30} \quad 1{,}491.2 \text{ cu ft}$

Sometimes the metering of gases takes place over water, some of which will evaporate and exert a partial pressure. This will have to be considered in correcting the gas volume to standard conditions.

Table 8-24. Aqueous-vapor Tension

Deg F	Vapor tension, in. Hg	Deg F	Vapor tension, in. Hg	Deg F	Vapor tension, in. Hg	Deg F	Vapor tension, in. Hg	Deg F	Vapor tension, in. Hg
32	0.1804	48	0.3363	64	0.601	80	1.029	96	1.706
33	0.1878	49	0.3492	65	0.622	81	1.003	97	1.759
34	0.1955	50	0.3625	66	0.644	82	1.098	98	1.813
35	0.2034	51	0.3762	67	0.667	83	1.134	99	1.869
36	0.2117	52	0.3903	68	0.690	84	1.171	100	1.926
37	0.2202	53	0.4049	69	0.714	85	1.209	101	1.985
38	0.2290	54	0.4201	70	0.739	86	1.248	102	2.045
39	0.2382	55	0.4357	71	0.764	87	1.280	103	2.107
40	0.2477	56	0.4518	72	0.790	88	1.331	104	2.171
41	0.2575	57	0.4684	73	0.817	89	1.373	105	2.236
42	0.2677	58	0.4856	74	0.845	90	1.417	106	2.303
43	0.2782	59	0.5034	75	0.873	91	1.462	107	2.372
44	0.2890	60	0.522	76	0.903	92	1.508	108	2.443
45	0.3002	61	0.541	77	0.933	93	1.556	109	2.515
46	0.3118	62	0.560	78	0.964	94	1.605	110	2.589
47	0.3238	63	0.580	79	0.996	95	1.655		

Using the above example, the temperature correction remains the same. For the pressure correction, 28.56 in. + 3.0 in., or 31.56 in., is the absolute pressure of the gas. From this must be deducted the aqueous-vapor tension read from Table 8-24 for 90°F. The actual gas pressure becomes 31.56 in. − 1.417, or 30.143 in. Thus, the volume of dry gas at 60° is

$$1{,}500 \times \left(\frac{60 + 460}{90 + 460}\right) \times \frac{30.143}{30} = 1{,}423.2 \text{ cu ft}$$

COAL SAMPLING*

It should be borne in mind that in sampling the purpose is to represent by a small quantity the whole pile, or car, or barge of coal, as the case may be. One scoopful taken at random or a single lump selected here and there will not be representative. In taking a sample, always use a scoop of proper capacity for the increment weight chosen. Never use the hands, for there will be an involuntary inclination to select pieces or sections of either particularly good or especially bad appearance. The best samples are obtained from coal in motion, *i.e.*, while it is being unloaded or conveyed from one place to another. Whether the coal is in motion or at rest, sample increments of uniform weight should be spaced as evenly as possible over the whole lot being sampled. If the coal is in a pile, or car, or barge, the increments should be taken from a number of locations at different levels to avoid the results of segregation. The term "increment" designates that quantity obtained by a single motion of the sampling scoop through a stream of coal or into a pile or top of car or barge.

Table 8-25

Top size, in.	Weight of increment, lb	Number of increments
4	10	20
2	6	20
1¼	4	20
⅝	2	20

Size of the Gross Sample. The size of the gross sample will vary according to the per cent of ash and size of coal. The size-weight table (Table 8-25) is suitable for bituminous coals having up to 10 per cent ash. If it is suspected that the ash percentage is above this, it is advisable to double the number of increments.

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* For further development of this subject, see G. B. Gould.[7]

Reduction of the Gross Sample. Samples obtained by reduction of the gross sample should be crushed or ground by passing through a suitable mill. After grinding, the sample may be divided by a riffle sampler or equivalent to quantities shown in Table 8-26, depending on the size of the largest piece. If no means is available for mechanical crushing and division, the gross sample should be placed on a hard, clean surface and all lumps broken down to smaller than ⅛ in. by means of a tamper. It should be carefully mixed and shoveled into a conical pile, and the pile flattened out, and quartered by drawing lines at right angles through the center. Opposite quarters should be rejected and the remainder mixed. This crushing, mixing, and quartering should be continued in accordance with the above size-weight relationships. The operations involved in this procedure are illustrated in Fig. 8-13.

Table 8-26

Size of largest piece, in.	*Limit of sample reduction, lb*
⅜	30
¼	15
⅛	2

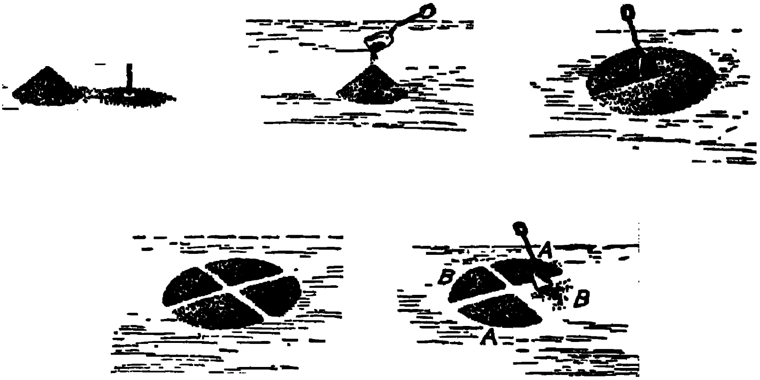

FIG. 8-13. Standard method of reducing gross coal sample by hand.

Sampling for Size Test of Anthracite. The gross sample should be systematically collected by suitable increments from various parts of the lot sampled. The increment weight for pea and No. 1 buck should be approximately 10 lb; for the smaller size, 5 lb. The gross sample weighing at least 100 lb should be mixed and reduced by quartering, without crushing, to the minimum amounts shown in Table 8-27.

Table 8-27

Size of coal	*Sample weight, lb*
Pea	50
No. 1 buckwheat	25
No. 2 buckwheat (rice)	10
No. 3 buckwheat (barley) . .	5

For ordinary operating requirements and when no dispute is involved, the foregoing simplified sampling procedure is sufficient. When greater accuracy is required reference should be made to the ASTM Standard D-492-46.

SAMPLING FUEL OIL

Methods for sampling fuel oil are given in detail by the ASTM in the Standard D-270-33. As conditions and facilities for taking samples are so variable, reference should be made to this whenever controversies are encountered or extreme accuracy is needed. The sampling of fuel oil in power plants is usually done by tapping the pipe line between pump and burner, although it is often necessary or required that samples be taken from cars, trucks, or tanks.

Sampling from a pipe line is accomplished by means of a fitting of three plug cocks constructed and placed to take a continuous sample from a cross section of the stream. The openings of these three cocks are set to obtain a steady stream at a rate to obtain 0.1 per cent of the lot as a sample but not to be less than 5 gal or more than 40 gal. The gross sample should be thoroughly mixed by a dasher stirrer or by rolling or shaking before the final 1-qt sample is withdrawn.

Cars, trucks, and tanks are sampled by means of a weighted beaker or bottle having an opening 1½ in. in diameter, the stopper of which can be withdrawn at the desired level. For full tanks of uniform cross section or diameter, one sample should be taken from the upper level at 10 per cent of the tank depth, three samples from the middle section, and one from the lower level 10 per cent of the tank diameter from the bottom. In horizontal cylindrical tanks, eight samples should be taken from the middle section. For partly filled tanks, reference should be made to the standard for the proper number of samples and levels. The composite sample should be at least 5 qt for vessels of uniform cross sections and 10 qt for horizontal cylindrical tanks.

For routine work, an all-levels sample from tanks is generally sufficient. This procedure consists in submerging a closed container to the draw-off level, then opening and raising it at such a rate that it will be nearly full when withdrawn. The gross sample of 5 to 10 qt should be mixed and 1 qt taken for the final portion.

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FUEL PREPARATION AND BURNING EQUIPMENT*

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SOLID FUELS

The preparation of coal for steaming purposes consists in cleaning it by one of the several available dry or wet processes and sizing it so that it is suitable for the particular type of fuel-burning equipment on which it is to be used

Bituminous coal is usually sized at the tipple as it is loaded into cars for shipment. Sizing practice varies in the different coal fields Table 8-28 represents some of the commercially available sizes produced

Table 8-28. Sizes of Bituminous Coal

Name	Size characteristics	Use
Run of mine	Coal as it comes from the mine without screening	Steam raising purposes and locally for domestic heating
5 in lump	Will not pass through a 5 in round hole screen	Domestic heating and some industrial uses
5- × 2 in egg	Passes through a 5 in and retained on a 2-in round-hole screen	Domestic heating hand-firing, gas producers
2- × $1\frac{1}{4}$-in nut	Passes through a 2-in and retained on a $1\frac{1}{4}$ in round hole screen	Domestic heating hand firing, small industrial stokers gas producers
$1\frac{1}{4}$- × $\frac{3}{4}$ in stoker coal	Passes through a $1\frac{1}{4}$ in and retained on a $\frac{3}{4}$ in round hole screen	Domestic and small industrial stokers
$\frac{3}{4}$ × 0 in slack	Passes through a $\frac{3}{4}$ in round hole equivalent screen	Industrial stokers and pulverizers

The sizing of anthracite is well established, and all mines prepare sized coal Table 8-29 gives the trade names and standard sizing for anthracite

The characteristics of coal that govern the selection of fuel-burning equipment, more than any other, are the properties of *caking* and *free burning*

Table 8-29. Anthracite Sizing

Name	Size characteristic (round hole screen)	Use
Broken	Through $4\frac{3}{8}$ in screen and over $3\frac{1}{4}$ in screen	Domestic heating
Egg	Through $3\frac{1}{4}$ in screen and over $2\frac{7}{16}$-in screen	Domestic heating
Stove	Through $2\frac{7}{16}$ in screen and over $1\frac{5}{8}$-in screen	Domestic heating
Chestnut	Through $1\frac{5}{8}$ in screen and over $\frac{13}{16}$ in screen	Domestic heating
Pea	Through $\frac{13}{16}$ in screen and over $\frac{9}{16}$-in screen	Hand firing
No 1 buckwheat	Through $\frac{9}{16}$ in screen and over $\frac{5}{16}$-in screen	Domestic stokers
No 2 buckwheat (rice)	Through $\frac{5}{16}$ in screen and over $\frac{3}{16}$-in screen	Domestic stokers
No 3 buckwheat (barley)	Through $\frac{3}{16}$ in screen and over $\frac{3}{32}$-in screen	Traveling-grate stokers
No 4 buckwheat	Through $\frac{3}{32}$ in screen and over $\frac{3}{64}$-in screen	Traveling grate stokers and pulverizers

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* Superior numbers refer to specific reference listed at the end of this subsection A portion of this material has been adapted by permission from *Combustion Engineering* published by Combustion Engineering-Superheater Inc

Caking coals, when heated in a furnace, pass through a plastic state during which the individual lumps fuse together into large masses of semicoke impervious to air flow. For this reason caking coals are burned in underfeed types of stokers as they have moving rams or other means for breaking up these masses of semicoke. Caking coals cannot be burned successfully on traveling- or chain-grate stokers because with these the fuel bed is undisturbed during its travel through the furnace. With spreader-stoker firing the caking is minimized owing to rapid oxidation during flight of the coal through the furnace atmosphere. This action results in a free-burning type of fuel bed.

In general, caking coals fall within the range of the high-volatile A, medium-volatile, and low-volatile bituminous coal groups (ASTM classification). Within these groups wide variation in behavior will be found.

Free-burning coals are ones in which the individual lumps do not fuse together but burn separately, or after fusion the masses break up quickly into fragments. Fuel beds of these coals need no agitation so that traveling- and chain-grate as well as spreader stokers are well adapted to them.

Anthracite, semianthracite, high-volatile bituminous C, subbituminous coal, and lignite fall into the free-burning ASTM classification.

The use of pulverized-fuel firing with any of the foregoing coals is governed by suitability for pulverizing and size of steam-generating unit with which it is to be used. Generally, anthracite and semianthracite are relatively difficult to pulverize and almost invariably call for stoker firing.

Coke breeze is another frequently available fuel for industrial steam making. It is a by-product in the manufacture of domestic and metallurgical coke. Usually all coke passing through a 3/4-in. round-hole screen is called "breeze." It is a low-volatile, free-burning, and extremely abrasive fuel which is burned almost exclusively on traveling-grate stokers.

There are many other solid fuels available for steam-making purposes, but most of them are limited in supply as they are by-products of some manufacturing process. Wet and dry wood, bark, spent-wood chips of various sorts, and bagasse all fall into this classification and are outside the scope of this section.

Equipment for Burning Solid Fuels

Stokers. Stokers available for feeding and burning various coals may be divided into four types:

1. Single-retort underfeed stokers
2. Multiple-retort underfeed stokers
3. Traveling- and chain-grate stokers
4. Spreader stokers

Single-retort Underfeed Stokers. The single-retort stoker, as illustrated in Fig. 8-14, is one that employs a stationary trough, or retort, open at the top and closed at the sides and at one end. The coal is fed into the trough through the open end by means of a reciprocating sliding bottom on which are mounted a feeder block at the hopper end and auxiliary pushers at intervals throughout its length. With each inward stroke the feeder block forces a quantity of coal into the retort.

Alternately moving and stationary grate bars extend laterally from each side of the retort to the dump grates located at the furnace side walls. The short lateral motion of the moving bars causes the coal to travel gradually toward the dump grates. The combined action of the feeder block, the auxiliary pushers, the sliding bottom, and the moving grate bars results in correct agitation, even distribution from front to rear, and a gradual feeding of the fuel toward the dump grates.

Air under pressure is supplied to the main wind box of the stoker directly below the retort. From here it enters the grate bars through openings in their underside at the retort ends. Some of the air flows through tuyère openings into the upper zone of the retort. The remainder is discharged from opening in the lower ends of the bars into the auxiliary wind box. It then passes up between the bars into the fuel bed. This design of air-distribution system provides a gradation in pressure so that, with decreasing fuel-bed thickness, there is a corresponding reduction in pressure.

The accumulated refuse on the dump grates is discharged periodically to the ash-pit as required.

Over-fire air ports are located in both the front and rear walls immediately above the retort. The over-fire air is used to complete the combustion of volatile matter rising from the distillation zone of the fuel bed.

Single-retort stokers for industrial application vary in size from approximately 3 ft 6 in. wide by 4 ft 0 in. long to a maximum of 12 ft 6 in. wide by 10 ft 0 in. long. Continuous coal-burning rates range from 250 to 5,400 lb per hr depending on the size and quality of the coal. Furnace heat-liberation rates for continuous operation should preferably be held to less than 40,000 Btu per cu ft per hr.

Multiple-retort Stokers. The multiple-retort stoker shown in Fig. 8-15 is one in which the underfeed section consists of a number of alternate, parallel, longitudinal strips comprising the retort and the tuyères. The retorts are open at both ends and the top but are closed on the sides. Tuyères form lateral extensions between adjacent retort sides. It is through these that air is supplied to this section of the stoker. No provision is made for air admission through the retorts. Thus, with this arrangement, approximately fifty per cent of the underfeed section is active grate area.

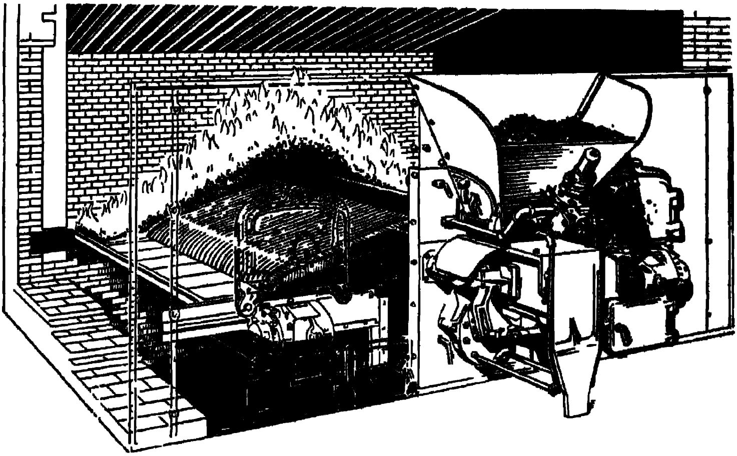

Fig. 8-14. View of a steam-driven Type E single-retort stoker with part of the front in phantom to show interior and fire on rear part of grate. (*Combustion Engineering—Superheater, Inc*)

At the lower end of the retort is an overfeed section comprising from 20 to 30 per cent of the total stoker area. It consists of deep, narrow bars with overlapping ribs grouped in movable carriers. A reciprocating motion is provided and arranged so that adjacent carriers have a slicing action for maximum agitation. The length of the stroke for alternate groups is adjustable from the stoker front. The entire overfeed zone is air-admitting and under damper control.

The ash-discharge section may be any one of three available types: dump grate, continuous ash discharge, and clinker grinder. When dump grates are used, the accumulated refuse is periodically discharged by lowering the dumps. With the continuous-discharge type the accumulated refuse is continuously moved through a gap between the bridge-wall overhang and the ash-discharge trays. This design is shown in Fig. 8-15. With the clinker grinder, a deep pit in which the grinder rolls are located is provided for ash accumulation. This refuse is then periodically discharged by operating the rolls.

In the operation of these stokers, coal is supplied to the retort by a reciprocating main ram. Secondary rams in the retort bottom and also the downward slope of the stoker cause the coal to travel progressively to the overfeed section. No lateral motion is provided to feed the fuel directly onto the tuyères. The shape of the throat through the front wall into the furnace for each retort allows the incoming coal to rise and form a relatively level bed. Ignition takes place at this point. The air supplied through the tuyères causes the coal resting on them to be consumed quickly. As a result a high ridge of smoldering caked coal remains over each retort. The feeding action of the rams advances the fuel, and some rolls on to the tuyères where it is burned at a high rate. Ignition penetration throughout the retort zone is relatively slow. To overcome this, the fuel-bed thickness is gradually decreased until the overfeed section is reached. In effect, the coal is lifted out of the retort because of its decreasing depth. This lifting action, aided by the shape of the auxiliary rams, maintains the supply of fuel to the tuyères. This side-feeding action is in step with the progressive lifting so that the fuel bed has a uniform height above and its top is parallel with the tuyère slope. A cross section of the fuel bed at the underfeed section will show a series of ridges over the retort and valleys over the tuyère.

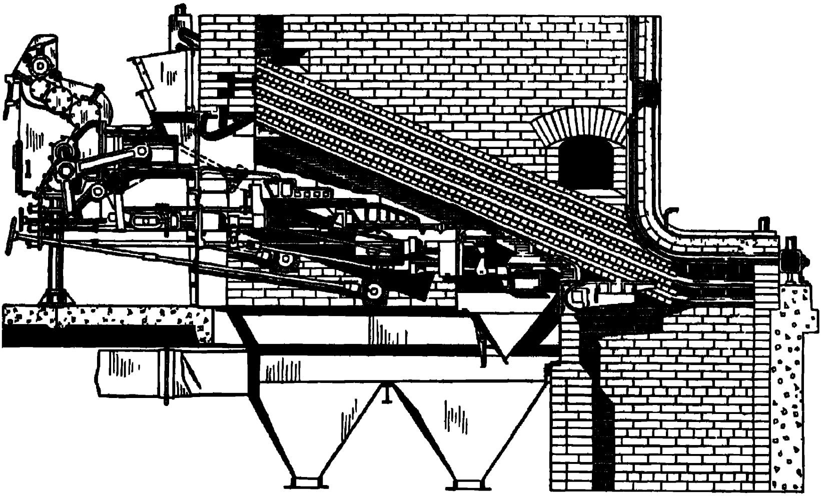

FIG. 8-15. Sectional side elevation showing details of the continuous-discharge type of multiple-retort stoker (*Combustion Engineering Superheater, Inc.*)

The height of the ridges and the depth of the valleys are governed by the caking characteristics of the coal.

Obviously it would be impossible to burn out uniformly a fuel bed that consists of alternately thick and thin ribbons of caked coal. Therefore, some means for leveling and compacting must be provided. It is for this purpose that a reciprocating overfeed section is used so as to shake down and level the fuel as it is discharged from the retorts. The slicing action between adjacent carriers serves effectively to break up large caked masses of coal and also minimize clinker formation. As a result the fuel bed in the overfeed zone is a level, intense coke fire requiring very little excess air to maintain low carbon loss in ashpit refuse.

Over-fire air ports are provided in the front wall above each retort to supply needed air for any unburned gas from the distillation zone.

The number of retorts for a satisfactory installation should never be less than four. The maximum number will depend on the furnace width available for installation. Stoker areas may vary from 70 to more than 500 sq ft. Continuous coal-burning rates will extend from 1,750 to 30,000 lb per hr depending on stoker size and coal quality. Furnace heat-liberation rates for continuous operation should preferably be held to less than 35,000 Btu per cu ft per hr.

Traveling- or Chain-grate Stokers. The traveling- or chain-grate stoker, shown in Fig. 8-16, provides a continuously moving, nonagitating type of grate surface. Coal is progressively supplied to the furnace in a level, uniformly thin bed, and the refuse is continually and automatically discharged to the ashpit. Prompt and rapid penetration of ignition is necessary for efficient operation. This is obtained by the use of suitably disposed arches which reflect heat onto the incoming green coal. The intensity of reflected heat will govern the speed with which it is possible to operate the stoker. It will also determine the rate at which ignition penetrates the fuel bed. The faster it is possible fully to ignite the coal, the higher the rate at which it can be burned.

The fuels customarily used on traveling- and chain-grate stokers are anthracite, coke breeze, lignite, and free-burning bituminous coal.

Anthracite and coke breeze have a low volatile-matter content and are relatively small in size with a very large percentage of "fines" or undersize. Unless provision is made to retain a maximum quantity of these "fines" in the furnace, excessive fly ash and high ashpit losses will result even though combustion rates are low. Arch design is an important factor. Furnaces in which the arch extends forward from the bridge wall to cover about 75 per cent of the stoker area make it possible to operate at relatively high combustion rates and excellent efficiencies. They ensure prompt and rapid penetration of ignition and minimize stratification by providing maximum turbulence in the furnace gas. Fly-ash carry-over is also reduced because much of it is redeposited on the fuel bed at the front of the furnace.

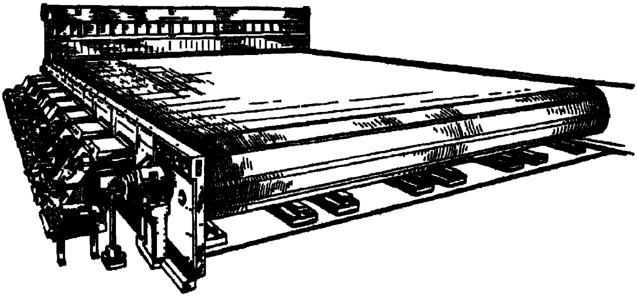

FIG. 8-16. View of a large traveling-grate stoker showing some details of construction. (*Combustion Engineering—Superheater, Inc.*)

Lignite, with its extremely high moisture content and tendency to "weather" when exposed to heat, is an ideal fuel for these stokers. Here again the use of rear-arch furnaces makes possible high combustion rates and excellent efficiency.

Free-burning bituminous coal is also suitable for chain-grate stokers. Because of the high volatile-matter content and more rapid penetration of ignition, somewhat shorter arches can be used.

The thin fuel beds on stokers of this type move relatively fast and are sensitive to air supply. Positive regulation is provided by a series of transverse zones, each under separate damper control.

Traveling- and chain-grate stokers for industrial purposes are built in sizes that range from 3 ft 0 in. wide by 9 ft 0 in. long to 24 ft 0 in. wide by 28 ft 0 in. long with corresponding areas of 27 to 672 sq ft. The allowable continuous-combustion rate for stokers of this type may range from 30 lb dry coal per square foot of grate per hour for anthracite to 50 lb dry coal per square foot per hour for lignite. Furnace heat-liberation rate should preferably be held below 55,000 Btu per cu ft per hr.

Spreader Stokers. The modern spreader stoker consists essentially of one or more units mounted on a cast-iron front, each comprising a coal hopper, a feeder which regulates the flow of coal in proportion to the load, and a distributor rotor which throws the coal into the furnace and distributes it to the grate. These stokers are available in two principal types: one for intermittent manual discharge of ash and the other for automatic continuous discharge of ash. The difference between these two designs lies in the grate construction. With the intermittent-ash-discharge type, either stationary or dumping grates are used. The plenum chamber below the grate for air distribution is sectionalized longitudinally into a number of compartments, one for each feeder unit. Dampers provide the regulation of air supply. The continuous-discharge design is provided with a traveling or chain grate. This grate is slowly and continuously moved forward to discharge the accumulated ash automatically. Transverse air compartments provide control of air distribution. In Fig. 8-17 some of the constructional details of a continuous-discharge spreader stoker are shown.

The spreader stoker uses the combined principles of suspension burning and the nonagitated type of fuel bed. Fresh coal is continually projected into the furnace above an already burning fuel bed. As a result the "fines" are quickly ignited and burned in suspension in the upper front furnace zone. The larger particles are exposed to the full effect of furnace temperature as they fall onto the fuel bed. The volatile matter is quickly driven off, and the tendency to coalesce is almost entirely destroyed. Thus the caking qualities have little or no effect. The active portion of the fuel bed is a thin layer of burning fuel on a thicker layer of accumulated ash Even distribution is required to maintain porosity for uniform air flow. Any localized thickening of the bed will set up high-resistance areas, and because of decrease of air flow through them, they are favorable to clinker formation. Ignition penetration is rapid and upward because fresh fuel is fed on top of an already burning bed.

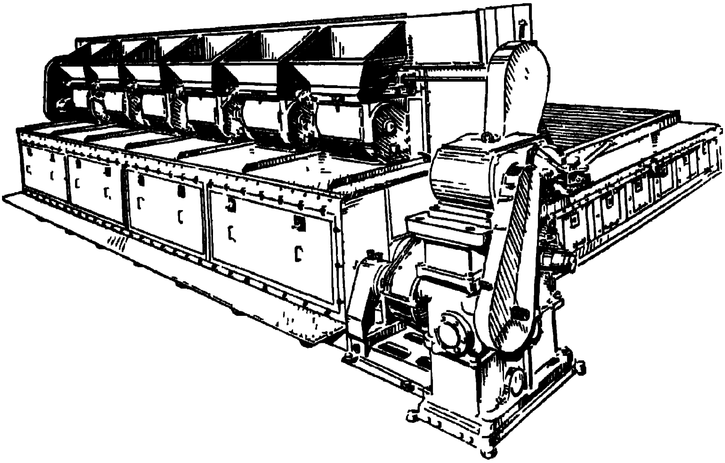

FIG. 8-17. A continuous-discharge-type spreader stoker. This stoker is 22 ft wide and 20 ft long. (*Combustion Engineering–Superheater, Inc*)

An outstanding characteristic of the spreader stoker is its ability to burn satisfactorily a wider range of fuels than stokers of any other type. Performance is best when fuel quality and sizing are good. This method of firing, however, was developed primarily to burn the lower grades of coal with high ash content and low ash-fusion temperature. Its operation with these fuels has been notably successful. It can handle with facility everything from semianthracite to lignite.

Since distribution to the grate is a most important factor, a satisfactory size consist of coal is necessary. A preferred specification is $\frac{3}{4}$ by 0 in. Under no condition should the top screen size exceed $1\frac{1}{4}$ in. "Bug dust" and similar size are also handled in a satisfactory manner.

Over-fire air or steam jets should always be provided to supply furnace turbulence which will break up any tendency toward gas stratification. Furnaces are of the "open type," devoid of any arches, with sufficient volume to keep the heat-liberation rate below 35,000 Btu per cu ft per hr at maximum continuous-operating rates where all refractory construction is used. For a suitably water-cooled furnace, this rate should not exceed 45,000 Btu per cu ft per hr.

Stoker areas may vary from 36 to 225 sq ft for the intermittent-ash-discharge type. For the continuous-discharge design these areas will range between 100 and 520 sq ft. Continuous-combustion rates should preferably be limited to less than 600,000 Btu per sq ft of grate per hour.

Pulverized Coal

Solid fuels, when burned in suspension, should contain an appreciable quantity of extremely fine dust so as to ensure prompt ignition. The amount of coarser material must be minimized if best combustion results are to be obtained.

The fineness to which coals should be pulverized will depend on many factors.

Caking coals, when exposed to furnace temperature, will swell and form lightweight, porous coke particles They may float out of the furnace before they are completely burned. As a result, carbon loss will be high unless pulverization is very fine. Free-burning coals, on the other hand, do not require the same degree of fineness because the swelling characteristic is absent.

High-volatile coals ignite more readily than those with a low volatile content. Therefore, they do not require the same degree of fine pulverization. With the exception of anthracite, however, the low-volatile coals are softer and may be said to have a higher grindability.

Some large furnaces may operate satisfactorily on high-volatile coal as coarse as 65 per cent minus 200 mesh. Small water-cooled furnaces, using low-volatile coal, may require a fineness of 85 per cent minimum through 200 mesh. Other influencing factors are burner and furnace design, disposition of furnace volume, length of flame travel, furnace temperature, and load characteristics. In general, however, small furnaces require finer pulverization than large ones.

The fineness of the product is usually expressed by the percentage of dust that will pass a sieve with specific size openings. For testing pulverized coal the most commonly used sieves are the 50-mesh sieve for determining the oversize and the 200-mesh sieve for determining the fine dust.

The mesh of a screen is designated by the number of openings per linear inch. Thus a 200-mesh screen has 200 openings to the inch, or 40,000 per sq in. The size of opening is governed by the diameter of wire used in making the screen. The U.S. Standard and W. S. Tyler are two commonly used screen sieves. The mesh and opening of these are shown in Table 8-30.

Table 8-30. Standard Screen Sizes

U. S. Standard			W. S. Tyler		
Mesh	In	Mm	Mesh	In.	Mm
20	0.033	0.84	20	0.033	0.83
30	0.023	0.59	28	0.023	0.59
40	0.0165	0.42	35	0.016	0.42
50	0.0117	0.30	48	0.0116	0.30
60	0.0098	0.25	60	0.0097	0.25
100	0.0058	0.149	100	0.0058	0.15
140	0.0041	0.105	150	0.0041	0.10
200	0.0029	0.074	200	0.0029	0.074
325	0.0017	0.044	325	0.0017	0.043

Pulverizers. The three basic principles for grinding coal are impact, attrition, and crushing. The four commonly used pulverizer types are ball, impact, ring-roll, and ball-race. Some of these mills employ one grinding principle, while others may use a combination of two or even all three principles in their operation.

Ball Mill. A ball mill is essentially a horizontal rotating cylinder whose length is somewhat greater than its diameter. The interior is fitted with heavy cast liners and filled to slightly less than half with balls that vary from 1 to 2 in. in diameter. Pulverization is accomplished through continued cascading of the ball and coal mixture as the mill is rotated. It is due in part to impact of the falling balls, to attrition through their slipping over each other as well as over the liners, and to crushing as they roll over the liners and also over each other. The larger pieces of coal are broken principally by impact, and the fine grinding is done by the rolling and sliding action of the balls.

Impact Mills. Impact mills consist primarily of a series of swing hammers or lugs revolving in a closed chamber. Grinding is accomplished by a combination of impact on the larger particles and attrition on the smaller ones.

Ring-roll and Ball-race Mills. Ring-roll and ball-race mills include a large class in which the coal is pulverized between two surfaces, one rolling over the other. The rolling member may be either a roll or a ball. If balls are used as the grinding element, they are confined between two races, and pressure to crush the coal is obtained by forcing the races together with heavy springs. The balls may be driven by rotating either the upper or lower race. Some additional grinding pressure is obtained from the centrifugal force of the rotating balls. In ring-roll mills, either the rolls or the grinding ring is rotated. In the first case, pressure to crush is obtained from centrifugal force as the roll assembly swings pendulumlike against the grinding ring. In the second case, the roll assembly is stationary, and springs are used to provide crushing pressure on the coal layer between roll and rotating grinding ring.

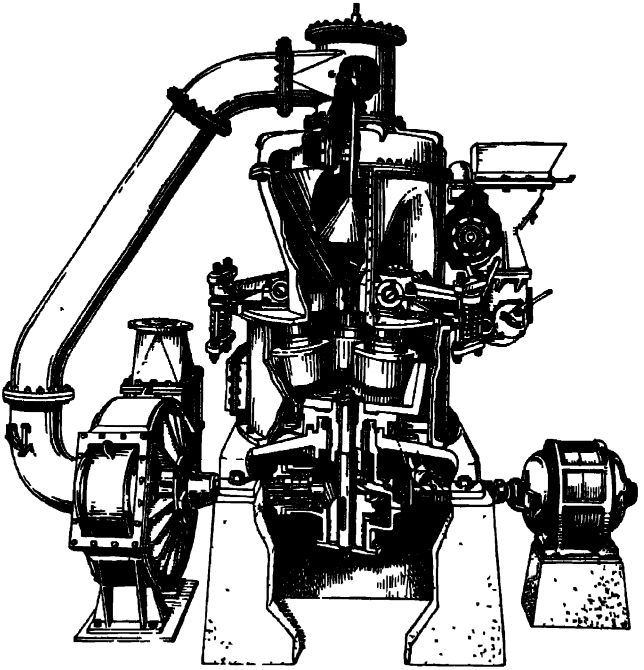

FIG. 8-18. A cut-away view showing interior details of the C-E Raymond bowl mill. (*Combustion Engineering—Superheater, Inc.*)

In all the mills just described an air system is provided to induce airflow for the purpose of removing the ground coal from them. In addition, some means of classification is used to maintain suitable size consist in the pulverized coal.

The C-E Raymond bowl mill, shown in Fig. 8-18, is of the ring-roll type. It consists essentially of a grinding ring carried by a revolving bowl. Stationary rolls, preloaded by adjustable spring pressure, are set so that they do not come in contact with the ring. A classifier, whose outlet is connected to the mill exhaust, is located above the grinding chamber. Fineness of coal at the classifier outlet is regulated to a large extent by position of adjustable vanes. A number of other influencing factors, however, will also have a bearing on vane setting.

Pulverizers will range in capacities from approximately 1,000 to 50,000 lb of coal per hour.

Some of the factors affecting mill capacity are grindability of coal, fineness required, moisture in coal, and air temperature available at pulverizer. Suitable correction for these must, therefore, be applied to obtain the true capacity of a given pulverizer.

Burners. The development of pulverized-coal firing for purposes of steam generation has been due, in large measure, to a better understanding of furnace design requirements. Uniform distribution of fuel and air to the furnace is also of prime importance. Turbulence provides the means for effective distribution and speed ignition of the incoming fuel and promotes rapid combustion by continually making available the free oxygen needed by the ignited combustible matter. These requirements are the governing factors in burner selection and application.

Methods used to fire pulverized coal are indicated by the manner in which fuel is injected into the furnace. These methods fall into three broad classifications, *viz.*, vertical firing, horizontal turbulent firing, and tangential firing. There are modifications, adaptations, and partial combinations of these three basic methods to make them suitable for variations in furnaces as well as in steam-generating-unit design.

Characteristics of the three basic firing methods differ widely, and each has its field of application. Thus the selection for any given installation is governed by a number of variables, of which the principal ones are size, shape, and volume of furnace available to develop the desired capacity. Furnace dimensions establish the maximum length of flame travel available The type of furnace-wall construction also influences selection, but rate of heat liberation may indicate a change in wall design. Quantity of coal to be burned, as well as its volatile matter and sulfur content, fusion temperature of ash, and fineness of pulverization will influence not only method of firing and type of wall construction to be used but also the method of ash disposal. Character of the load together with extent and duration of peaks and extremely low-load periods are also of great importance. In fact there are so many influencing variables that no hard and fast rules can be established. Much reliance must be placed on actual knowledge, supplemented by operating experience, which parallels the conditions surrounding the particular installation under consideration.

Each of the foregoing firing methods requires a different burner design because of variations in the manner in which air and coal are mixed to produce efficient and complete combustion. Fundamentally, however, all burner designs must be such that the air and coal are supplied to the furnace so as to provide stable and prompt ignition; positive adjustment and control of ignition point and flame shape, completeness of combustion; uniform distribution of excess air, temperature, and gas flow at furnace outlet; freedom from localized slag deposits; protection against overheating, internal fires, and excessive wear in the burner; and accessibility for adjustment and replacement of parts.

Vertical Firing. Details of a burner assembly for vertical firing are shown in Fig. 8-19. In this design the lower edge of the fuel nozzle is serrated to increase surface exposure of the issuing fuel stream to furnace radiation and thereby aid in speeding ignition. In addition, deflector plates, not shown in the illustration, are installed in the lower part of the burner housing. These deflectors cause the tertiary air to cut across individual fuel streams and thereby set up considerable turbulence which, in turn, promotes rapid penetration of ignition.

The air required for combustion is divided into three parts: primary air, secondary air, and tertiary air. The primary air, a relatively small percentage of the total supply, is used to transport the pulverized coal from the storage bin, or mill outlet, to the burner nozzle and then blow it into the furnace. The tertiary air, in approximately the same percentage as the primary air, is admitted under damper control through the burner housing around the fuel nozzles to the furnace. Secondary air, the major portion of total requirements, is admitted in a horizontal direction through a large number of small air ports which checkerboard the front wall of the furnace, and dampers are used to regulate quantity and distribution. This arrangement makes it possible to secure horizontal penetration between adjacent vertical fuel streams and thereby progressively supply the secondary air in a manner to ensure complete and uniform combustion.

Individual burners have capacities up to approximately 2,000 lb coal per hr. They are installed side by side on relatively close centers in an arch near the top of the furnace and project the coal vertically downward in a relatively thin flat stream. In this manner uniform distribution of fuel across the entire furnace width is obtained.

Horizontal Turbulent Firing. The development of direct-fired systems in which pulverized coal is transported directly from mill to burner, however, resulted in a demand for larger burners where individual capacity, in many cases, would be equal to that of the pulverizer. To meet these requirements it was necessary to provide a design in which the large concentrated fuel streams were quickly and uniformly spread out over a relatively wide area. In this manner high intensity of heat liberation in small sections of the furnace cavity is avoided, and there is instead an evenly distributed flow of heat through the furnace, although individual burner capacity may be as high as 150 million Btu per hr. These results are obtained through the use of designs similar to the one illustrated in Fig 8-20 in which a large central fuel nozzle is surrounded by a housing provided with adjustable vanes.

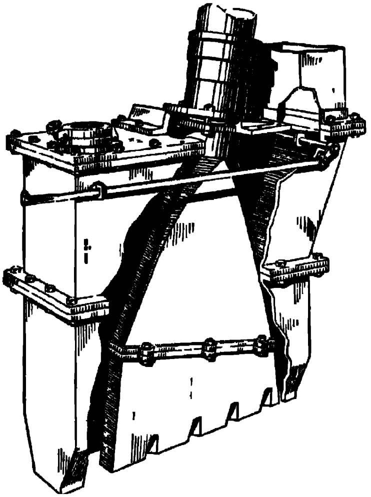

Fig 8-19 Details of a burner for vertical firing. (*Combustion Engineering—Superheater, Inc*)

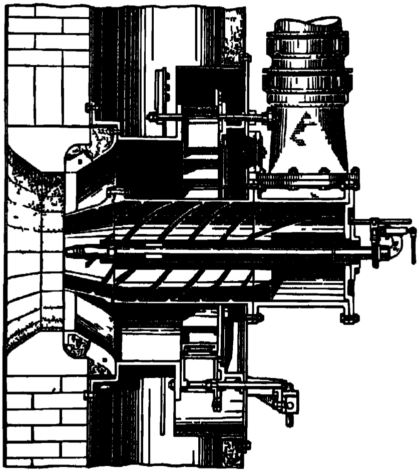

Fig. 8-20 A horizontal turbulent burner arranged for pulverized-coal and oil firing. (*Combustion Engineering—Superheater, Inc.*)

The central nozzle has a volute type of inlet so that rotation is imparted to the entering mixture of fuel and primary air. Vanes in the housing are used to control the degree of rotation imparted to the secondary air stream. As the streams from these two sections of the burner come together in the burner throat, their rotational motion establishes a high degree of turbulence and causes the mixture to spread quickly as it emerges into the furnace cavity. When looking directly at one of these burners in operation, the burning fuel and air mixture has the appearance of a large sunflower in which the flames seem to radiate, petallike, from the relatively dark center at the nozzle.

Tangential Firing. With tangential firing the burners are located in each of the four furnace corners close to the water screen, floor, or hopper bottom. Individual burner assemblies consist of two, three, or four fuel nozzles surrounded by secondary air-inlet boxes equipped with guide vanes and dampers. Many details of these burners are shown in Fig. 8-21.

In this method of firing, the fuel and air streams issuing from the four corners are directed along tangents to a small imaginary circle in the center of the furnace. No attempt is made to mix the secondary air and fuel at the burner, but sufficient primary air is used to ensure prompt ignition. Where these four streams meet, intense turbulence is set up, and a rotative motion is imparted to the burning mixture which spreads out and is uniformly distributed over the furnace cross section. The characteristic feature of uniformity of mixture in the furnace overcomes any effect such as results from unequal division of fuel or air with other methods of firing.

Furnace outlet temperature is a function of heat absorption by the waterwalls, as well as rate of heat liberation in the furnace cavity. Heat absorption by the walls varies with their cleanliness, as well as with the area subjected to direct radiation and convection transfer. Thus, to secure the required temperature of gas at the furnace outlet over a range in capacity, it is necessary to make a relative change in both furnace volume and area of heat-absorbing surface.

The foregoing requirements have led to the development of an adjustable tangential burner in which the fuel nozzles and air-directing vanes may be selectively tilted upward or downward by the operator through a range of 25 deg above or below horizontal. Thus, by directing the flames toward the upper zone of the furnace, the gas outlet temperature will be kept higher during the light load, or when walls are clean, than when the entire furnace is used. For heavy loads, or when walls are coated with dust, the entire furnace volume can be used by tilting the burners downward to maintain the outlet gas temperature below the point of slagging. At practically any given load point a change of approximately 200°F at furnace outlet or approximately 75°F in steam temperature, is available by operating the burner nozzles over the entire range of adjustment.

Ability to use the available furnace volume selectively, through positioning the zone of turbulent combustion and thereby varying effectiveness of furnace waterwall area, makes it possible to control furnace outlet temperature and thus regulate steam temperature over a wide range in capacity. A supplementary gas by-pass damper provides the means for final incremental steam-temperature adjustments.

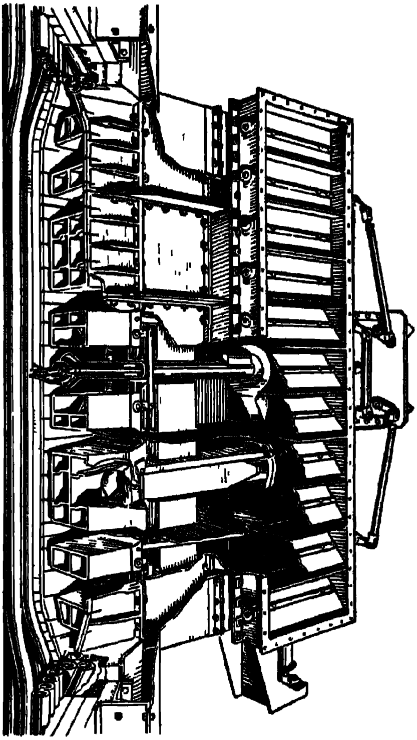

FIG. 8-21. Cut-away view showing details of tangential burner with adjustable coal nozzles. (*Combustion Engineering—Superheater, Inc.*)

LIQUID FUELS

Fuel oil may be defined as petroleum or any of its liquefiable residues remaining after the more volatile constituents have been removed.

Crude petroleum is sometimes burned in the state in which it is taken from the ground. In this condition, most of it will contain sufficient lighter gasoline fractions to lower the flash point and present a fire hazard. Through limited fractional distillation, or topping, the lighter gasoline can be removed, and a safe fuel oil can be produced. If the refining process is carried through extended fractional distillation

Table 8-31. National Bureau of Standards Detailed Requirements for Fuel Oils[a],*

Grade[b]		Flash point, deg F		Pour point, deg F	Water and sediment, per cent	Carbon residue, per cent	Ash, per cent	Distillation temperatures, deg F				Viscosity, sec			
								10 per cent point	90 per cent point		End point	Saybolt universal, at 100°F		Saybolt furol, at 122°F	
Number	Description of fuel oil	Min	Max	Max	Max	Max	Max	Max	Max	Min	Max	Max	Min	Max	Min
1	Distillate oil for use in burners requiring a volatile fuel	100 or legal	165	0	Trace	0.05 on 10% residuum[d]	....	410	. .	. .	560[e]				
2	Distillate oil for use in burners requiring a moderately volatile fuel	110 or legal	190	10[c]	0.05	0.25 on 10% residuum[f]	....	440	600						
3	Distillate oil for use in burners requiring a low-viscosity fuel	110 or legal	230	20[c]	0.10	0.15 straight	....	...	675	600[g]		45			
5	Oil for use in burners requiring a medium-viscosity fuel	130 or legal	...	...	1.00	....	0.10	..	...	...	.	...	50	40	
6	Oil for use in burners equipped with preheaters, permitting a high-viscosity fuel	150	.	..	2.00[h]	....	....	..	...	..	..	..	...	300	45

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* "Fuel Oil, Commercial Standards," National Bureau of Standards publication CS12-40.

[a] Recognizing the necessity for low-sulfur fuel oils used in connection with heat-treatment, nonferrous metal, glass and ceramic furnaces, and other special uses, a sulfur requirement may be specified in accordance with the following table:

Grade of fuel oil	*Sulfur (maximum) per cent*
1	0.5
2	0.5
3	0.75
5	No limit
6	No limit

Other sulfur limits may be specified only by mutual agreement between the buyer and seller.

[b] It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[c] Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0°F under any conditions.

[d] For use in other than sleeve-type blue-flame burners, carbon residue on 10-per cent residuum may be increased to a maximum of 0.12 per cent. This limit may be specified by mutual agreement between the buyer and seller.

[e] The maximum end point may be increased to 590°F when used in burners other than sleeve-type blue-flame burners.

[f] To meet certain burner requirements, the carbon-residue limit may be reduced to 0.15 per cent on 10-per cent residuum.

[g] The minimum distillation temperature of 600°F for 90 per cent may be waived if API gravity is 26 or lower.

[h] Water by distillation, plus sediment by extraction. Sum, maximum 2.0 per cent. The maximum sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

and cracking, such fuels as gasoline, kerosene, gas oil, light fuel oil, lubricating oil, heavy fuel oil, residual tar, pitch, and petroleum coke are produced.

For industrial purposes heavy fuel oil, or bunker C oil, is the one used for steam making.

Properties of Fuel Oil

The important properties of fuel oil are specific gravity, heating value, viscosity, flash and fire point, sulfur content, ash content, and congealing point.

Density is one of the most important properties, since fuel oil is sold on a volumetric basis. Standard density is taken at 60°F and referred to water at 60°F. It is usually designated in degrees API, for which the American Petroleum Institute's standard formula is

$$\text{Degrees API} = \frac{141.5}{\text{specific gravity at } 60°\text{F}/60°\text{F}} - 131.5$$

Between specific gravity of all petroleum products and their higher heating value there is a relationship which has been expressed in a formula by the U.S. Bureau of Standards[1] as

$$\text{Heat of combustion at constant volume (Btu per lb)} = 22{,}320 - 3{,}780 \times \text{sp gr (at } 60°\text{F}/60°\text{F)}$$

To provide standardization for fuel oil the National Bureau of Standards has established specifications for five different grades. The details of these specifications are given in Table 8-31.

Oil Analysis

It will be noted that grade 6, which is sometimes designated as heavy industrial fuel oil or bunker C oil, is specified mainly by viscosity. An average analysis of grade 6 oil may be taken as carbon, 87.3 per cent; hydrogen, 10.8 per cent, sulfur, 1.2 per cent; nitrogen, 0.2 per cent; oxygen and indeterminates, 0.5 per cent; and higher heating value, 18,500 Btu per lb

For handling fuel oil from storage tanks to burner nozzles, there is usually installed a compact duplex interconnected assembly of strainers, oil heaters, pumps, and automatic regulators, together will all other auxiliary equipment and instruments that may be required.

Oil Temperature

When pumping oil to storage tanks, or from them to oil heaters, heating to a temperature of approximately 100°F is ordinarily required. For use with steam-atomizing burners, heating to approximately 185°F is desirable. For mechanical atomization, the oil requires still greater fluidity and must therefore be heated to approximately 220°F.

Oil Pressure

For steam-atomizing burners the oil pumping pressure may vary from about 2 psi at minimum capacity to 125 psi at maximum. This pressure range is based on a fuel piping system of adequate size for the capacity required because burner capacities are obtained through the use of steam for atomization.

With mechanical atomization, oil pressure is used to produce atomization. The pressure required, therefore, depends on the load, as well as the sprayer plate used, and will ordinarily vary from 50 to 250 psi.

Oil Burners

Oil burners are classified according to the method used for securing atomization as follows:

1. Air-atomizing burners
2. Steam-atomizing burners
3. Mechanical-atomizing burners

Air-atomizing Burners. Only a relatively few industrial installations have burners of the air-atomizing type, and, therefore, our discussion will be confined to steam- and mechanical-atomizing types.

Steam-atomizing Burners. Steam-atomizing burners may be subdivided into two general groups: the *outside mix* and the *inside mix* according as the oil and steam mix outside or inside the burner nozzle.

Outside-mix nozzles usually produce a flat flame and are used generally with the natural-draft types of burners. In these the air supply is controlled by varying furnace draft with intermediate regulation provided by front-wall dampers. Burners of this type are used for relatively small installations where only moderate capacities are required and relatively wide furnaces are available.

Inside-mix types of nozzles will give a flat or hollow conical flame, depending on the design of tip used.

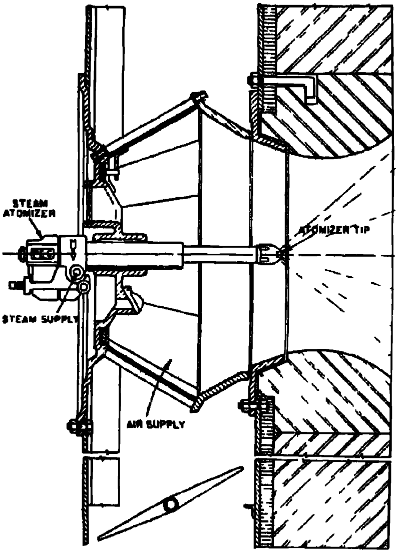

FIG. 8-22. Typical conical-flame, forced-draft, steam-atomizing burner. (*Peabody Engineering Corp.*)

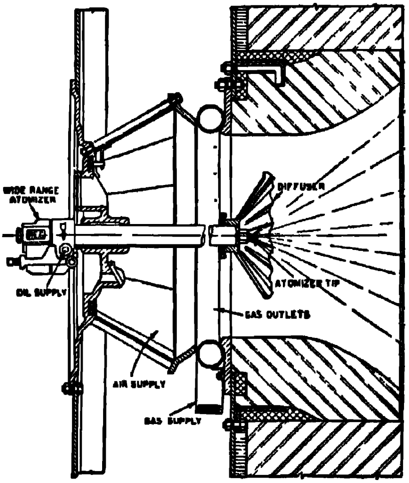

FIG. 8-23. Combination mechanical-atomizing and gas ring type of burner. (*Peabody Engineering Corp.*)

Nozzles that produce a conical flame are the ones which find greater acceptance for industrial steam making. With these is used a circular housing having some form of controllable register. For natural-draft installations, the air supply is provided by furnace suction, as well as by the aspirating effect of the steam as it issues from the burner nozzles. When large-capacity burners are required, the air is supplied to the burner register by a forced-draft fan. The pressure and volume of the air supply are governed by the fuel-injection capacity of the nozzles.

A forced-draft burner of the steam-atomizing type is shown in Fig 8-22. In the design illustrated a multiblade register is used to vary air flow and regulate its spin. An inside-mix steam atomizer provides a strong conical-shaped spray, and interchangeable tips are used to modify flame spread and oil capacity. Individual burners of this type may have capacities up to 120 million Btu per hr, and furnace liberation rates should preferably be held below 30,000 Btu per cu ft per hr at maximum continuous capacity.

Mechanical-atomizing Burners. Mechanical-atomizing burners consist of four principal elements: the atomizer, the diffuser, the air register, and the refractory burner-throat opening.

The *atomizer* is designed to break up the oil mechanically into a fine, mistlike, uniform spray which will burn with a minimum of excess air when projected into the furnace.

The *diffuser* is a shield, in the form of a perforated hollow metal cone, mounted near the furnace end of the mechanical-atomizer gun assembly. It protects the base of the spray from the high-velocity air and stabilizes the flame to prevent it from being blown away from the atomizer tip.

The *air register* is an integral part of the oil-burner assembly and usually consists of a series of overlapping vanes which deliver the air for combustion to the furnace throat with the correct degree of spin.

The *refractory burner-throat opening* has a carefully shaped contour designed to hold the air around the flame and simultaneously assist combustion by radiating heat to the burning fuel. A typical mechanical-atomizing-burner assembly is shown in Fig 8-23.

The capacity range for a given atomizer-tip orifice is limited to approximately 1½ or 2 to 1, with an oil-pressure change of as much as 200 psi. Thus, for satisfactory operation over reasonably wide load ranges, it is necessary to have a multiplicity of burners, as well as several sizes of atomizer assemblies for each burner.

Recently several systems have been developed for widening the operating range of mechanical atomizers to as much as 10 to 1 without the necessity of constantly changing sprayer plates and tip orifices Thus, with properly applied control it is possible to have satisfactory and close automatic regulation for fluctuating load conditions.

Individual burners of the mechanical-atomizing type may have capacities up to 100 million Btu per hr, and maximum continuous furnace-liberation rates should preferably not exceed 30,000 Btu per cu ft per hr.

GASEOUS FUELS

The principal gaseous fuels used for steam-generating purposes are, in the order of their importance, natural gas, blast-furnace gas, blended refinery gas, and coke-oven gas Performance characteristics of these gaseous fuels are governed by the characteristics and proportions of the various simple gases they contain Of greatest importance are ignition temperature, the speed at which various proportions of the gas and air will burn, flame temperature, and amount of heat radiated from the burning gas.

For industrial purposes, natural gas is the one that has widest application. Blast-furnace gas, blended refinery gas, and coke-oven gas are by-products of a manufacturing process and are usually not available except in the plant where produced. Consequently, our discussion will be confined to the use of natural gas.

Natural Gas

Natural gas is probably the closest approach to an ideal fuel because it can be burned with very low excess air and contains little or no solid residue. It consists

Table 8-32. Typical Analyses of Natural Gas

Location	Constituents, per cent by volume					Sp gr (air = 1.0)	Higher heating value, Btu/cu ft (Std cond)
	CO_2	O_2	N_2	CH_4	C_2H_6		
Kiefer, Okla.*	2.4	0.0	1.8	64.1	31.7	0.97	1272
Leavenworth, Kan *	0.8	0.6	6.3	88.9	3.4	0.62	964
Caddo Parish, La.*	0.9	0.0	1.5	97.6	0.0	0.75	1039
Quannah, Tex *	0.2	0.4	5.8	85.5	9.1	0.74	1011
Los Angeles, Calif †	0.9	0.0	0.1	85.8	13.2	0.65	1108
Buffalo, N. Y †	0.0	0.0	0.4	88.1	11.5	0.61	1093
Charleston, W. Va †	0.0	0.0	0.7	76.8	22.5	0.67	1172

* Bending, E A *Combustion*, July, 1931
† U.S. Bureau of Mines, *Tech. Paper* 158 July, 1917.

mainly of methane (CH_4) with smaller quantities of other hydrocarbons, principally ethane (C_2H_6), although carbon dioxide (CO_2) and nitrogen (N_2) are usually present in small amounts. Sometimes there are also appreciable amounts of hydrogen sulfide (H_2S). Oxygen (O_2) is present only when there has been an infiltration of atmospheric air. Helium, a rare inert gas, is contained in natural gas from some Texas fields. In Table 8-32 are shown typical analyses of natural gas.

Natural gas when found in areas containing oil deposits may be impregnated with heavy saturated hydrocarbons, which are liquid at ordinary pressure and temperature. This *wet* gas is then dried by stripping from it those liquids which constitute the so-called "casing-head gasoline." In view of the foregoing, it is well to remember that the terms "wet" and "dry," when associated with natural gas, refer to its gasoline content, not to its moisture content. For pipe-line transmission the gas is usually saturated with water vapor, by means of steam jets, to reduce gasket maintenance.

Natural-gas Burners. Burners for natural-gas firing are generally divided into l-p and h-p designs. This classification is derived from the gas pressure required to provide satisfactory operation.

Low-pressure Burners. Low-pressure burners are usually operated with gas pressures of from $\frac{1}{8}$ to 4 psi. A multijet construction delivers the gas in many small streams to discharge opening in the burner throat. Air for combustion, induced by furnace draft, flows around the jets and the gas-air mixture is then burned as it enters the furnace. Dampers are used to regulate air supply. Burners of this design are used where moderate capacities are desired for boilers having relatively wide furnaces.

High-pressure Burners. The use of high pressures for pipe-line transportation of gas over long distances is necessary to maintain low production costs. As a result the pressure in the gas main serving a plant may be 30 to 50 psi. Regulators are used to provide pressure step down to burner main and to maintain a constant pressure so that automatic regulators are effective in varying gas flow to meet fluctuations in steam demands. Where h-p gas is available, the burner operates on pressures from 2 to 25 psi.

High-pressure gas burners may be classified into four main types, each indicative of the manner in which gas is discharged into, and mixed with, the air for combustion.

The *gas-ring* type of burner is one in which a gas ring in the form of an annular manifold, located between air register and furnace wall, surrounds the burner opening. A series of orifices drilled around the inner surface of this ring spray gas angularly in the form of a screen across the air stream toward the center of the burner opening. Operation can be easily maintained with excess air as low as 7 per cent. Figure 8-23 shows a burner of this type, provided for alternate use of fuel oil.

The *center-diffusion-tube* type of burner is one in which the gas head with its orifices is located at the center of the burner. For combination firing, either a mechanical-atomizing or steam-atomizing oil-gun assembly is inserted through an inner sleeve in the gas tube. In this design, Fig. 8-24, the gas, as well as the oil, leaves the burner tip in the shape of a hollow cone and at a wide angle to the entering air. A diffuser plate, integral with the gas tip, assures steady and prompt ignition. Integral adjustable dampers are used to control air quantity and produce the desired spin necessary for good combustion.

The *turbine* type of gas burner is one using revolving instead of fixed orifices. These orifices distribute gas transversely to the air stream while rotating at high speed. With this design, the energy in the gas under pressure is utilized to perform the mechanical mixing and proportioning of the gas and air for combustion.

The *tangential* type is one in which banks of burner assemblies are located in the four corners of the furnace. Air and fuel-gas streams are directed along paths tangent to a small circle at the furnace center. This arrangement provides added turbulence and mixing and is well adapted to steam-generating units of large size.

Furnace Liberation. There are many natural gas-fired installations operating at high furnace heat-liberation rates. However, tests have demonstrated that there is a decided tendency for the burning mixture to short-circuit the available volume so that continuous or incomplete combustion may extend into the boiler and superheater zones of the unit. By providing generous furnace volume, based on a liberation rate which does not exceed 30,000 Btu per cu ft per hr at maximum continuous capacity, much operating difficulty can be avoided. The capacity of an individual burner is as high as 120 million Btu per hr.

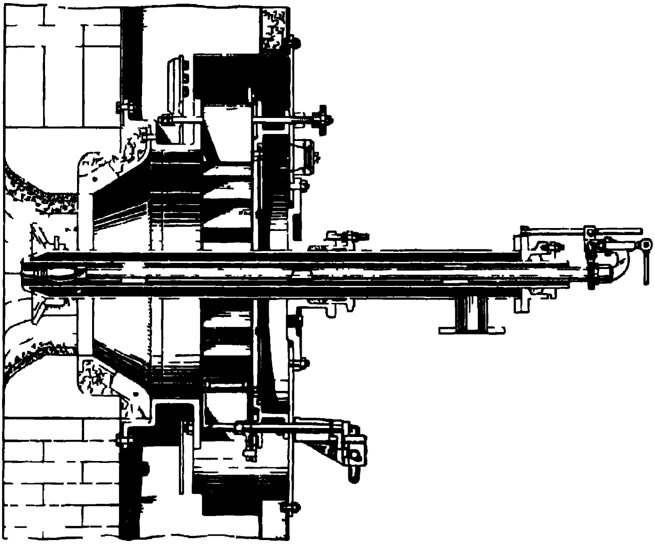

Fig. 8-24. Combination steam-atomizing oil and natural-gas burner of the center-diffusion tube type. (*Combustion Engineering-Superheater, Inc.*)

SUPPLEMENTARY FUELS

In the foregoing we have discussed burner designs which can alternately and without difficulty handle both liquid and gaseous fuels. Necessary provision can also be made for pulverized coal firing in the same burner casing by providing a suitable nozzle for the particular type of burner in use. Under these conditions the furnace design should be developed for the basic fuel or the one which will have the greatest influence from the standpoint of both economy and operation.

In some instances stokers may be used to burn the solid fuel. In these cases it is advisable to provide a separate burning zone for each fuel and the outlet of the two zones is then combined to provide a common inlet to the boiler section. This minimizes slag deposits on heat-absorbing surfaces because the products of combustion do not mix until the combustion process is completed.

Reference

1. U. S. Bureau of Standards *Misc. Pub.* 97, 1929.

BOILERS AND BOILER AUXILIARIES*

BY OTTO DE LORENZI

Director of Education, Combustion Engineering—Superheater, Inc.

The modern steam-generating unit, in its simplest form, consists of a grate, a furnace, and a boiler The larger and more comprehensive designs include mechanical fuel-burning equipment, water-cooled furnaces, superheaters, boiler surface, heat-recovery apparatus, and related auxiliaries. For maximum over-all reliability and operating economy, each component part must be correctly proportioned and properly related to the other elements of the unit so as to comprise a balanced design.

CAPACITY AND RATING

At present the means for expressing capacity and rating of modern steam-generating units is not clearly defined. These modern units usually consist of several different types of heat-absorbing equipment, and, therefore, combined performance cannot be expressed in either boiler horsepower or equivalent evaporation. This has led to the now generally accepted practice of stating capacity in actual pounds of steam generated per hour at a specific pressure and temperature and from feed water at a definite temperature. A more exact means for expressing capacity is in Btu per hour input to the steam, thereby accounting for variation in pressure, steam temperature, and feed-water temperature. This latter method has not yet been adopted by engineers for general use, but the transition which is now taking place will result in the development of more rational standards

TYPES OF BOILERS

A wide variety of boilers is now in general use for industrial and central-station steam plants. All boilers for these purposes fall into two general classifications: fire-tube boilers and water-tube boilers. Any discussion of these must, of necessity, be a general one. Description and illustration will include principal boiler types and some of their applications, together with range in size, capacity, pressure, and steam temperature.

Fire-tube Boilers. A fire-tube boiler is one in which tubes comprising the heat-absorbing surfaces are water-touched on the outside surfaces, while flue gas flows through them.

Fire-tube boilers are relatively inexpensive, but their overload capacity is limited, and exit gas temperature rises rapidly with increased output. The horizontal return-tubular boiler, commonly called the "HRT boiler," is the most widely used of this type.

An HRT boiler consists of a cylindrical shell into the ends of which heads are fitted. These heads are provided with suitable openings into which the tubes are expanded. Tube diameters may vary from 2 to 4 in. The present standard for shell construction is fusion welding so that there are no overlapping seams or butt-strapped joints. Shell sizes may vary from 54-in. diameter by 12-ft length to 108 in. diameter by 20-ft length.

Boilers of this type are principally used in small heating and industrial plants where the pressure and capacity requirements are relatively low and where only a limited amount of superheat is needed. The capacity usually falls in the range from 1,000 to 15,000 lb steam per hr with design pressure up to 250 psi and up to 150°F superheat. Rates of evaporation should not exceed 7 lb per sq ft per hr and preferably should be held at approximately five pound.

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* A portion of this material has been adapted by permission from *Combustion Engineering*, published Combustion Engineering-Superheater, Inc.

For coal firing, the smaller boilers are usually equipped with hand-fired grates and the larger sizes with stokers. Oil, gas, and waste fuel are also frequently used. A typical stoker-fired installation in which an HRT boiler, equipped with a girth-type superheater, is used is shown in Fig. 8-25.

A modification of the standard HRT boiler, which is compact, semiportable, and requires a minimum of refractory setting, is the C-E Premier boiler shown in Fig. 8-26. It consists essentially of two shells of different diameters joined together end to end to form a single unit. The shell having the larger diameter is at the rear and has a series of short tubes in its lower section. These tubes form the flues between the furnace and the rear combustion chamber. In the upper portion of the larger diameter shell are long tubes which extend forward through the small shell and conduct the flue gas from the rear combustion chamber to the smoke box at the front. Design pressure is limited to approximately 150 psi, and the capacity range is from 1,000 to 12,000 lb of steam per hr.

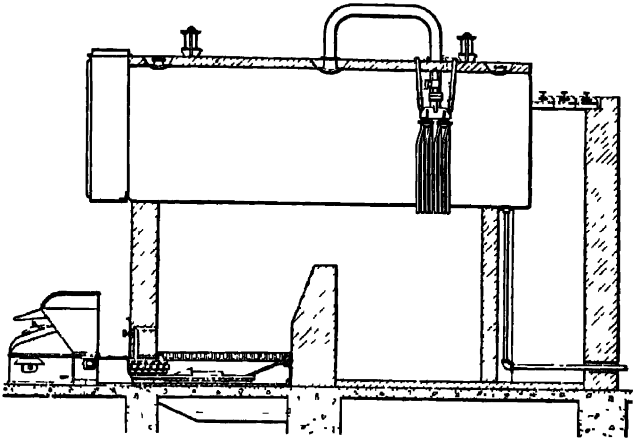

FIG. 8-25. A small modern unit comprising an HRT boiler equipped with girth-type superheater and fired by a single-retort stoker. (*Combustion Engineering Superheater, Inc.*)

These boilers are used primarily for small- and medium-sized industrial and heating plants. They are principally coal fired. Hand or stoker firing may be used. Oil, gas, and wood have also been used as fuel. In the case of wood, a Dutch-oven furnace is frequently installed.

Water-tube Boilers. A water-tube boiler is one in which the tubes comprising the heat-absorbing surfaces contain water, while the flue gas flows over their outside surfaces.

Water-tube boilers are broadly classified into the straight-tube and bent-tube types, each of which has a number of design variations. They are used almost exclusively for pressures above 150 psi and capacities over 15,000 lb steam per hr.

The horizontally inclined straight-tube boiler, in its best known forms—the sectional-header and box-header types—was at one time by far the most widely used. As pressures and capacities were gradually stepped up, however, their use declined because the required rate of steam output per foot of furnace width soon exceeded the circulation limit. As a result, the bent-tube boiler, because its greater design flexibility makes it possible to meet the demands of modern h-p and high-capacity installations, has now practically replaced the straight-tube type.

Straight-tube Boilers. The *box-header boiler* takes its name from the boxlike uptake and downtake headers which form the tube sheets and handhole sheet at each end of the main tube bank. Stay bolts are used to brace these box headers against internal pressure. The main tube bank is inclined at a slope of from 1 to 3 in 12 in., and may be arranged for either horizontal or vertical baffling as required. The boilers may be of the longitudinal or cross-drum type.

The *longitudinal drum* type of box-header boiler (see Fig. 8-27) consists of one, two, or three parallel drums to which the uptake and downtake headers are attached.

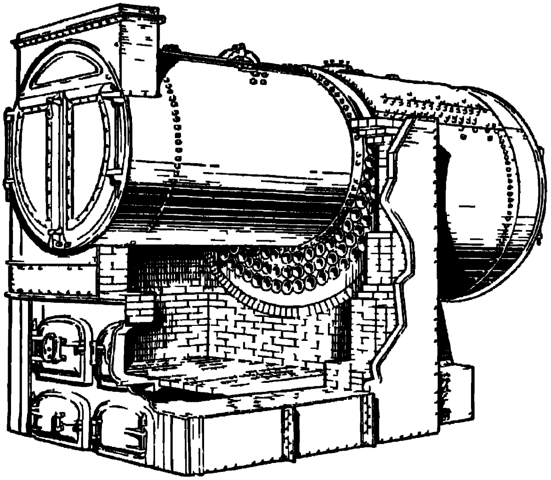

FIG. 8-26. A Premier boiler arranged for hand firing. (*Combustion Engineering–Superheater, Inc.*)

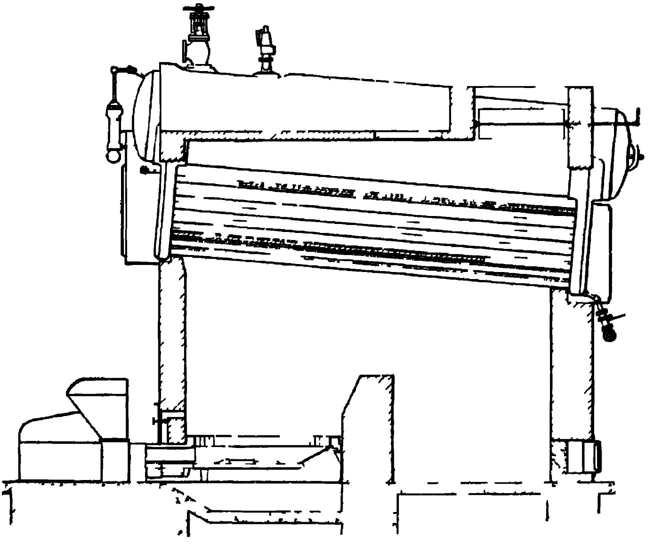

FIG. 8-27. A horizontally baffled inclined-tube box-header boiler equipped with single-retort stoker. (*Combustion Engineering–Superheater, Inc.*)

These drums may be either horizontal or parallel with the tube bank, and their number and diameter will depend on the capacity desired. Box-header boilers, because of the stayed and riveted construction, are limited to a maximum design pressure of 300 psi and the capacity ranges from 4,000 to 90,000 lb steam per hr.

The *cross drum* type of box-header boiler uses a single drum, arranged transversely to the tube bank and located above the downtake header. The front or uptake header is connected to the drum by horizontal circulating tubes, and connecting nipples complete the circuit between the downtake header and drum. Either longitudinal or cross baffling can be used. The cross-drum box-header boiler shown in Fig. 8-28 is provided with a combination of horizontal and cross baffling. The superheater is usually arranged above the tube bank, which limits superheat to approximately 200°F.

The *sectional-header boiler* is made up of a number of sinuous forged-steel headers into which the tubes are rolled. The tube bank carried between these headers contains the major portion of the heating surface and has a slope of 3 in 12 in Tube lengths are preferably 18 to 20 ft. The size of a boiler is varied by changing the heating surface in individual sections and by increasing or decreasing the number of sections. Two or three sections may be placed in series by using interconnecting nipples. This construction is used when an interdeck superheater is required or where an exceedingly high tube bank is necessary, as in the case of waste-heat or single-pass design. These boilers are generally of the cross-drum type using circulating and connecting nipples to complete the circuit with the tube bank. Steam drums vary in diameter from 42 to 72 in., depending on capacity and pressure. Design pressure varies from 160 to 2,000 psi, with capacity ranges between 1,000 and 525,000 lb steam per hr. Some features of this type of boiler are shown in Fig. 8-29.

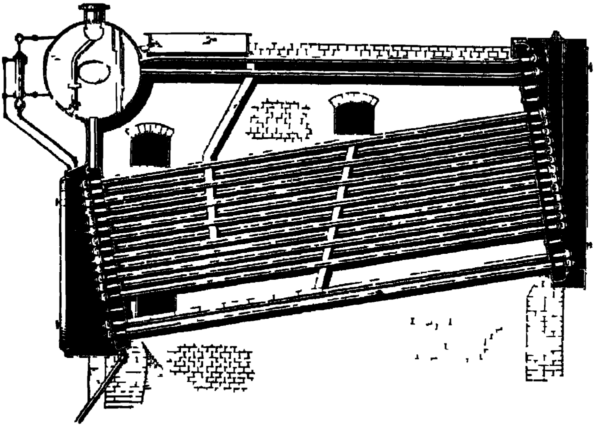

FIG 8-28. A cross-drum box-header boiler arranged with combination baffling (*Combustion Engineering--Superheater, Inc.*)

Bent-tube Boilers. Bent-tube boilers of conventional design usually have a lower, or mud drum, which is interconnected by means of bent tubes with one, two, or three upper steam-and-water drums. The heating surface, except in a few designs, is arranged in two or three banks, and vertical baffles or a combination of vertical and cross baffles are used. Superheaters may be located intertube in the first boiler bank or interbank between the first and second tube banks. Owing to flexibility in design these boilers may range in capacity from 6,000 to 1 million lb steam per hr with a design pressure range of from 160 to 1,800 psi with steam temperature 950°F. There are, of course, capacity pressure and steam-temperature limitations for each specific design. These will be covered in the discussion to follow.

The *four-drum* boiler was one of the first of bent-tube type to be extensively used. It consists of three upper steam-and-water drums and one lower water or mud drum. The tubes for interconnecting these drums are arranged in three banks with baffling to provide for three passes of the flue gas over this surface. The number of tubes in each row varies with the boiler size and diameter of lower drum. Capacity is usually the governing factor in determining boiler width and also influences vertical distance between lower and upper drums. There are several different ways in which tubes may be installed in the several boiler banks to provide rapid and positive circulation as well as adequate steam distribution to the various upper drums. The four-drum boiler shown in Fig. 8-30 is of a conventional type. The front and middle boiler banks are connected into both the front and middle upper drums to equalize discharge of the steam-and-water mixture and thereby improve circulation and reduce carry-over with the steam. Boilers of this general type are usually designed for pressure from 160 to 1,000 psi and a capacity range of from 7,500 to 350,000 lb steam per hr.

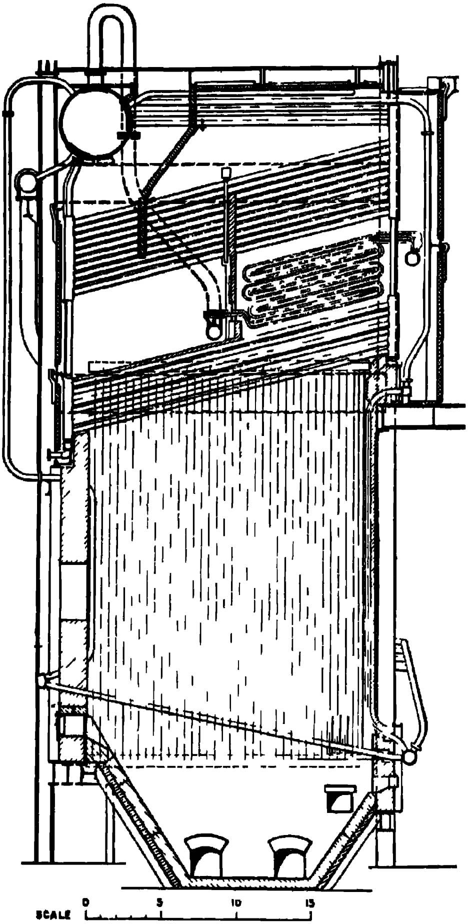

FIG. 8-29. A pulverized-fuel-fired sectional-header boiler of the cross-drum type equipped with an interdeck superheater. Furnace is fully water cooled by a bottom screen. Capacity 176,000 lb of steam per hr at 450 psi and 750°F. (*Combustion Engineering Superheater, Inc.*)

The *low-head* type of bent-tube boiler has been designed for limited space conditions so that it could compete with both the HRT and the cross-drum box-header

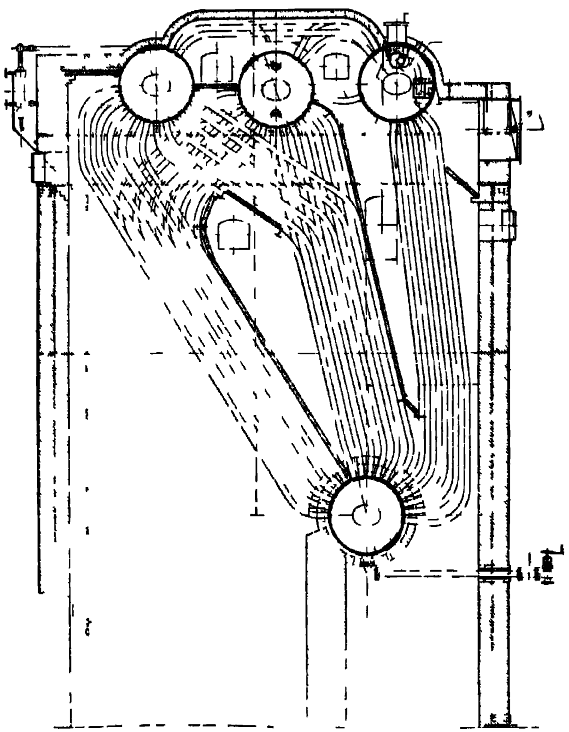

Fig. 8-30. A four-drum bent-tube boiler in which the first boiler bank is connected into both the first and middle upper drums. (*Combustion Engineering-Superheater Inc.*)

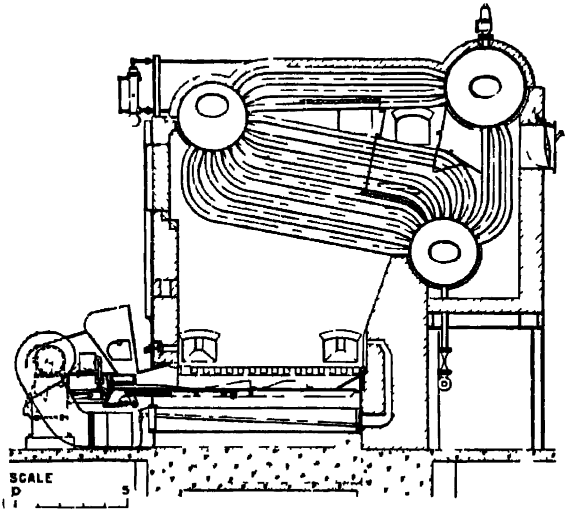

Fig. 8-31. Low-head bent-tube boiler fired by a single-retort stoker. (*Combustion Engineering-Superheater Inc.*)

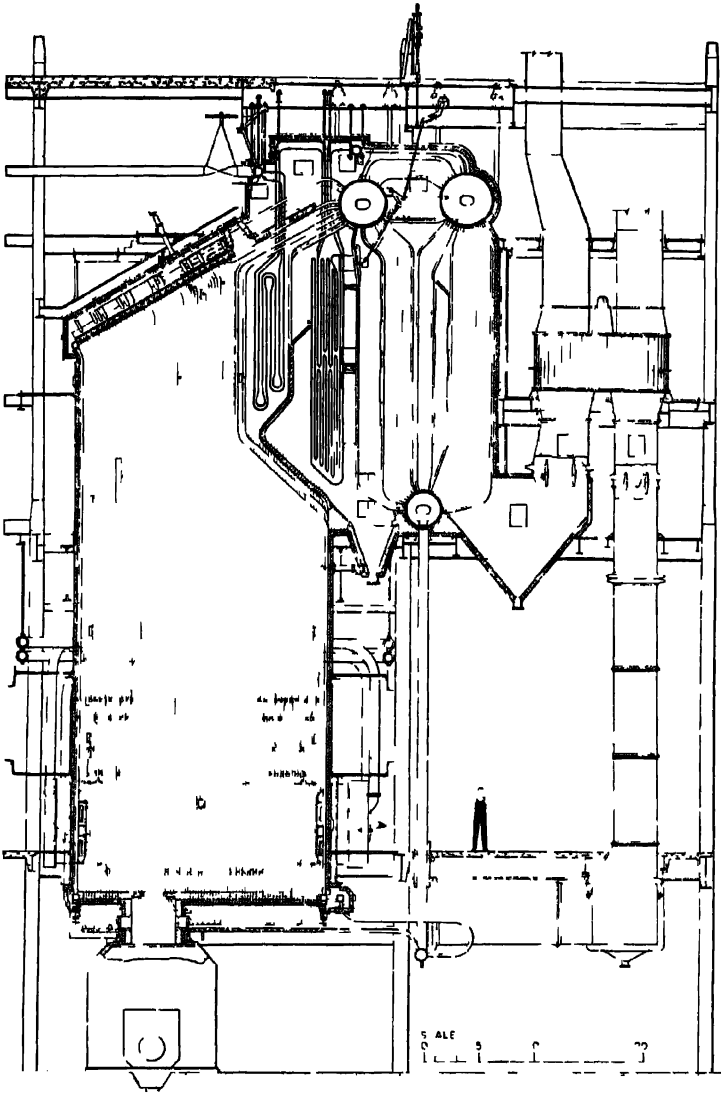

FIG 8-32 A pulverized-fuel-fired public-utility type steam-generating unit with tangential burners continuous slag-tap furnace three drum boiler and air heater Capacity, 400 000 lb of steam per hr at 875 psi and 900°F (*Combustion Engineering Superheater Inc*)

boilers Many of its characteristic features are shown in Fig 8 31 Its capacity range varies from 6,000 to 10 000 lb steam per hr with a design pressure range from 150 to 450 psi When operated at relatively high capacity the exit gas temperatures are higher than with straight-tube or other bent-tube design of similar size These boilers are well standardized as to construction and are available in a relatively wide range of sizes

The three-drum boiler, *high-head design*, is one that has never been standardized to the same extent as the four-drum types. Many have been installed in public-utility and large industrial plants. This boiler is provided with two upper steam-and-water drums and a lower mud drum. The tube banks are varied in design and arrangement for the specific condition under which this boiler is to be operated. This is perhaps best illustrated in Fig. 8-32 which shows one of several units installed for a large public-utility plant. The boilers have fully water-cooled furnaces arranged for pulverized-fuel firing. A large two-stage interbank superheater is used to provide a steam temperature of 900°F. Following the superheater is the convection surface arranged in two banks. Finally, external to the boiler is a large air heater. This unit has an over-all efficiency of 85 per cent at a capacity of 400,000 lb steam per hr at 875 psi and 900°F. It is of interest to know that the distribution of heat-absorbing surfaces, exclusive of air heater, is as follows: boiler, 47.4 per cent; furnace, 17.6 per cent, and superheater, 35 per cent.

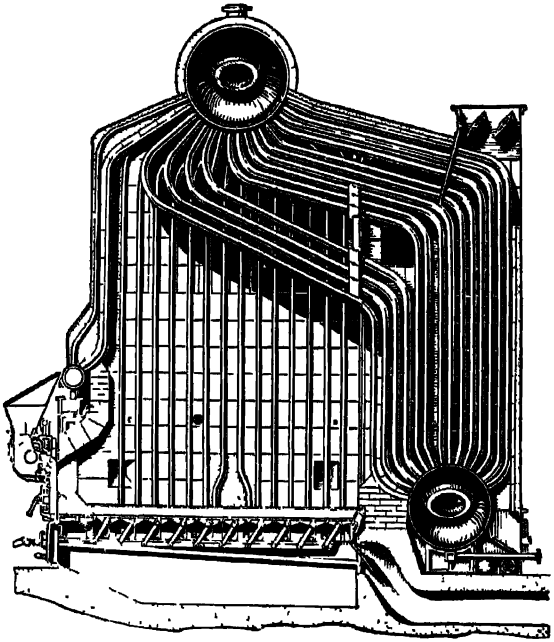

FIG. 8-33. A package-type boiler equipped with spreader stoker. (*Combustion Engineering Superheater, Inc.*)

Two-drum boilers are perhaps the simplest of the bent-tube types. Originally they were limited in application owing to low capacity and wet steam resulting from improper arrangement of tubes entering the upper steam-and-water drum and faulty drum internals which were ineffective in separating water from the steam. Modern redesign, however, has changed this so that boilers of this type are now available in well-standardized units for the capacity range from 15,000 to 300,000 lb steam per hr with design pressure from 160 to 1,000 psi and steam temperature up to 900°F. Typical of the small- and medium-sized industrial steam-generating units is the *package type* shown in Fig. 8-33. These may be fired by spreader stokers, single-retort underfeed stoker, oil, or gas. The capacity ranges from 18,000 to 45,000 lb steam per hr in a series of standard sizes. For larger capacities a steam-generating unit of the type shown in Fig. 8-34 is used. These are available in standardized sizes up to a capacity of 300,000 lb steam per hr with design pressure up to 1,000 psi and steam temperature up to 900°F Pulverized-fuel, oil, and gas firing are most frequently used, however, and with some furnace modifications stokers of any type may be readily applied. A particular feature in this design which contributes to uniformity of furnace condition, as well as to steam dryness and distribution, is the symmetrical arrangement of the unit in relation to the furnace center line. As a result of this, the gas enters the tube bank at low velocity at a practically constant temperature across the full width of the unit. There is no concentration of heat in any particular spot, and furnace conditions are uniformly constant The release of steam to the drum is uniform along its length, thereby eliminating surging water level and periodic carry-over of water slugs into the superheater.

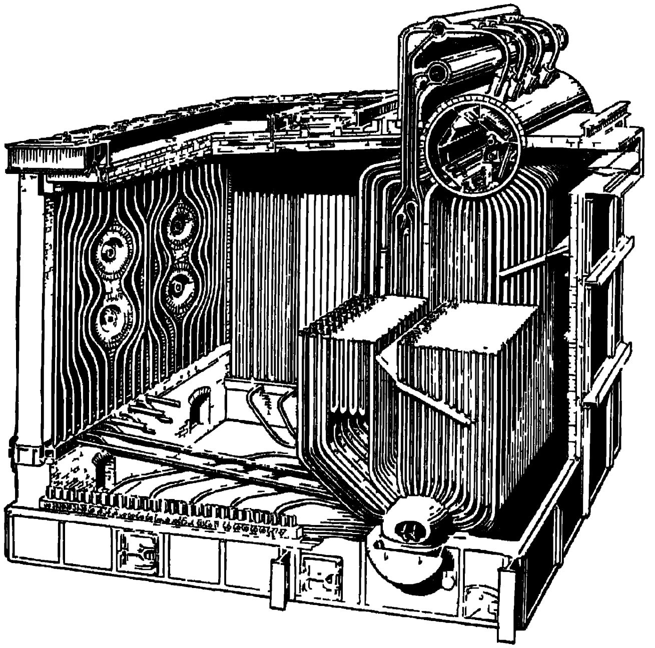

FIG. 8-34. Sectional view of a steam generator arranged for pulverized coal, oil, or gas firing. (*Combustion Engineering- Superheater, Inc*)

High-pressure steam generators are by far most widely used in the modern central stations of the public utilities. These units consist of completely water-cooled furnaces, large superheaters, and sometimes reheaters, boiler convection bank, economizers in some cases, and large air heaters. The amount of convection surface installed will frequently depend on whether or not an economizer is used. The unit designs are not entirely standardized because they are selected to meet exacting specifications which all too frequently vary from installation to installation. Generally there are two pressure classifications: 900 to 1,000 psi and 1,200 to 1,700 psi. Steam temperature ranges from 850 to 1050 F, and provisions are usually made for maintaining constant temperature from maximum load down to some predetermined partial load point. In addition to other operation requirements, furnace volume, together with disposition and extent of furnace heat absorbing surfaces, plays an important part in determining superheater location, arrangement, and size, as well as amount of boiler convection surface needed Thus for high pressure and high steam temperature, when complete furnace water cooling is used, most of the steam is generated in the furnace walls. As a result, only a small convection bank is required

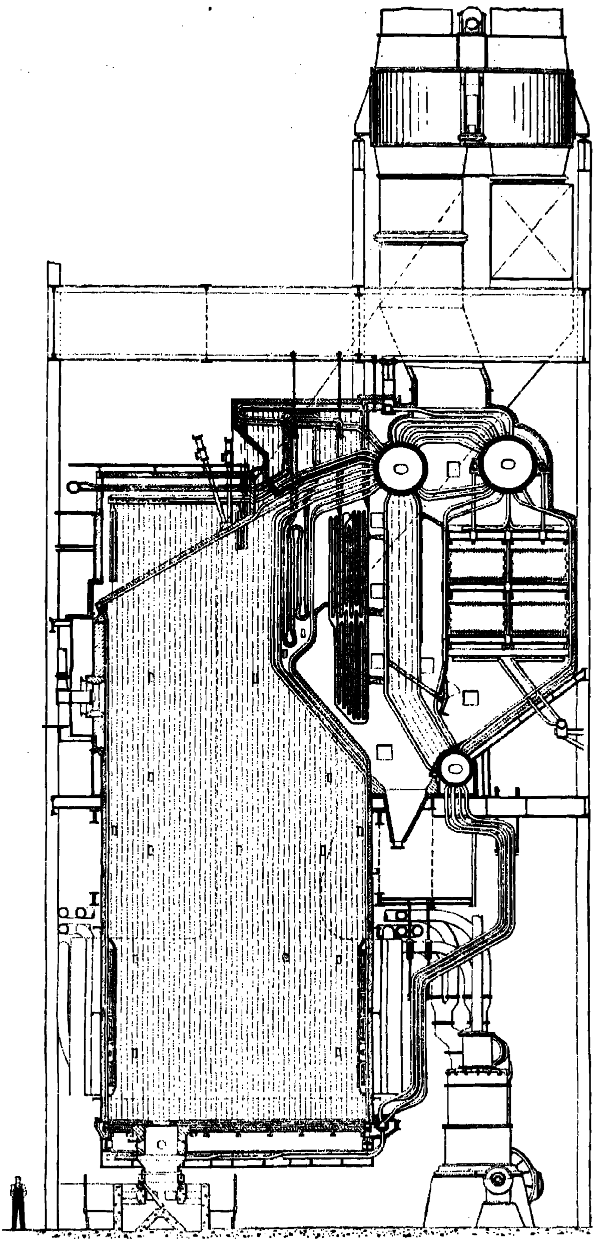

FIG. 8-35. A large steam-generating unit having a capacity of 1,000,000 lb of steam per hr at 1625 psi and 955°F. Continuous slag-tap furnace arranged for tangential firing, with vertically adjustable burners, is used. (*Combustion Engineering—Superheater, Inc.*)

between the superheater outlet and the economizer inlet. An arrangement of this general type is shown in Fig. 8-35 where the unit has a continuous capacity of 1 million lb steam per hr at 1,675 psi and 955°F at an over-all efficiency of 88.8 per cent. In this instance, the heat-absorbing surface, exclusive of the air heater, is disposed as follows: boiler, 14.6 per cent; furnace, 12.4 per cent; superheater, 34.7 per cent; and economizer, 38.3 per cent. There are a number of additional arrangements of high-capacity h-p units which will be used to illustrate other subject matter in this section.

SUPERHEATERS AND DESUPERHEATERS

The superheater is a device for imparting additional heat to saturated steam so that its temperature is raised and its volume increased. It consists of heat-absorbing

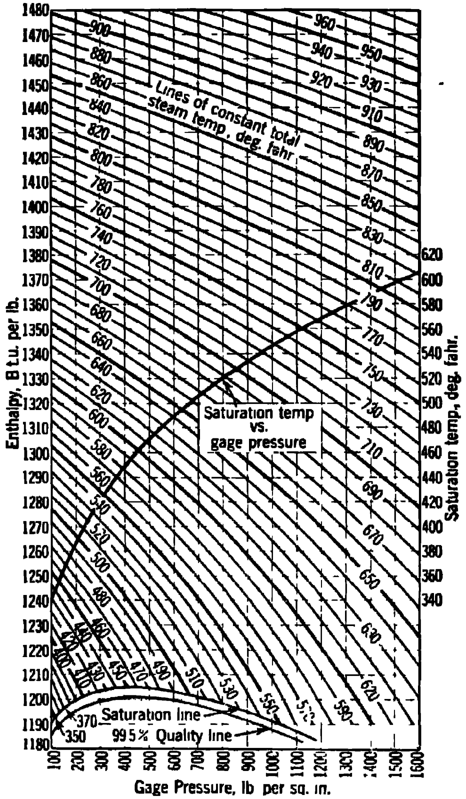

FIG. 8-36. Variation in heat content of steam at different pressures and temperatures. (*Combustion Engineering—Superheater, Inc.*)

surface, located in the path of the flue gas, which receives saturated steam after it has been removed from the generating portion of the boiler.

The properties of saturated steam approximate those of a perfect gas. It contains no moisture, and it cannot condense until it reaches saturated-steam temperature. With saturated steam, the heat available depends entirely on pressure, while with superheated steam there is additional heat proportionate to the degree of superheat. All these factors are illustrated by the series of curves in Fig. 8-36 which show variation in heat content of both saturated and superheated steam at different pressure and up to a temperature of 900°F. Also included is a curve of saturated-steam temperature for various pressures.

The characteristic of a convection-type superheater is rising steam temperature with capacity increase, while that of a radiant type is falling steam temperature. Practically all present-day superheaters are of the convection type in which heat is absorbed from the hot flue gas as it impinges and flows around the tubes. If the superheater is so located that it is not entirely shielded by boiler surface, it will also receive heat through radiation from the furnace. As a result steam-temperature variation over a range in capacity will be less marked than with a straight convection type. By referring to Figs. 8-32 and 8-35 it will be noted that the first bank in each of these superheaters is the combination convection and radiant type because it is located behind the equivalent of one row of boiler tubes. The second bank of these superheaters is of the straight convection type. The superheater shown in Fig. 8-34 is of the convection type as it is located behind a baffle which shields it from the radiant heat. The interdeck superheater shown in Fig. 8-29 is also of the straight convection type. The girth-type superheater shown in Fig. 8-25 employs both radiant and convection absorption and a measure of steam-temperature adjustment may be obtained by changing the clearance between the superheater elements and the boiler shell.

The performance of a superheater is closely related to operation of the turbine to which steam is supplied. Materials used in these prime movers are carefully chosen, and running clearances are limited so as to obtain satisfactory steam rates. Thus it becomes not only desirable but necessary to provide constant steam temperature over a wide range in load for modern h-p high-temperature installation.

By-pass Damper Control. One means available for providing constant steam temperatures from some predetermined partial load to full load is the by-pass damper. With this arrangement the quantity of hot gas passing over the superheater surface is varied to maintain the desired steam temperature. The superheater is designed to provide full steam temperature at the predetermined partial load point, usually not less than 75 per cent of full load, and therefore is oversize at the higher capacities. The steam temperature will rise unless some of the gas is by-passed to reduce effectiveness of the superheater. Automatic controls may be used to make necessary by-pass damper adjustments and thereby easily maintain temperature to within plus or minus 10°F.

Adjustable Burner Control. In the previous discussion we have indicated a number of variable factors which govern superheater performance. The amount of superheater surface and its location in a given steam generating unit are fixed by the steam temperature required at some specific capacity and by the fuel used. The performance of this superheater at capacity points other than that for which it is designed, will depend upon temperature and mass flow of the gas sweeping over its surface, temperature of the feed water, cleanliness of the furnace heat-absorbing surfaces, and output at which the unit is operated. The contour of the temperature-capacity curve is fixed, and it can be shifted bodily upward or downward only by changing either the amount of gas that flows over the superheater or the entering gas temperature. The change in gas flow may be secured by some form of by-pass damper arrangement, as previously discussed. The gas-temperature change can be accomplished by selectively using all or part of the provided furnace volume and area of heat-absorbing surfaces. By combining variable gas flow with variable furnace temperature, it is possible to secure control of steam temperature to within plus or minus 3°F over a wide range in capacity.

Control of furnace outlet temperature is possible with tangential firing. With this arrangement, fuel and air from burner assemblies located in each of the four corners are directed along tangents to a small circle lying at the furnace center. A high degree of radiation and convection transfer is secured through the sweeping action of the flame envelope over the wall surfaces. The regulation of furnace temperature is then secured, as graphically illustrated in Fig. 8-37, through changes in angularity of the burner nozzles. Thus, by directing the incandescent turbulent mass of burning fuel toward the upper zone of the furnace, the resulting outlet temperature will be made higher during light load, or when walls are clean than would be the case with fixed burners which constantly use the entire provided furnace volume. For heavy loads or when walls are coated with dust the entire furnace volume can be used to maintain the desired outlet temperature by tilting the burners downward. Now, by adjusting the burner nozzles to the proper angularity for any given and reasonably varying load condition a by-pass damper can be confined to its most effective operating range. Thus the combination of adjustable burner nozzles with a by-pass damper provides a more positive combination for maintaining steam temperature with less variation from normal, over a far wider operating range than when a by-pass damper and fixed burners are used.

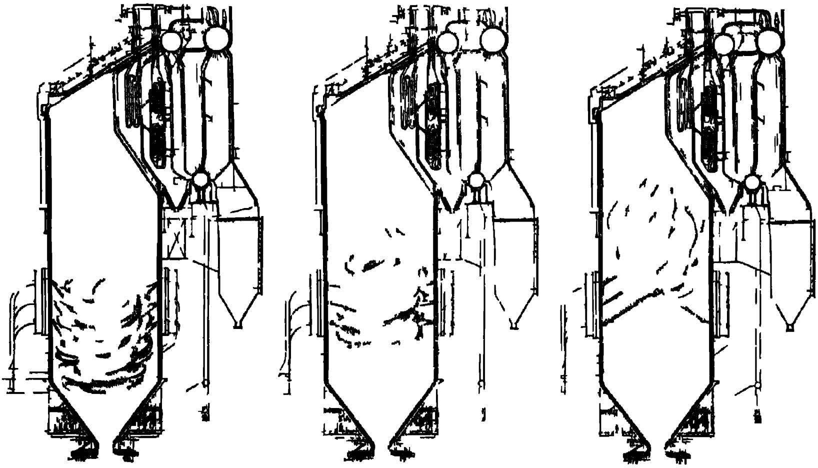

Fig. 8-37. Diagrammatic sketch showing how the zone of turbulent burning may be selectively positioned by adjustment of burner nozzles. (*Combustion Engineering-Superheater Inc.*)

Desuperheaters. An alternate or sometimes supplementary arrangement to the by-pass damper for steam temperature regulation is some form of desuperheater. Desuperheaters are used to effect a reduction in temperature of the superheated steam. They are preferably located between sections of the superheater, especially where the final steam temperature is above 850 F. In these interstage installations, the steam is first passed through a primary superheater where it is raised to some intermediate temperature which may be only slightly below that finally desired. It is next passed through the desuperheater in which its temperature reduction is controlled so that after continuing through the final stage of the superheater the required temperature is maintained at the outlet. Desuperheaters do not necessarily extend the range over which constant steam temperature is obtained except in those cases where the superheater is of considerably increased size.

Desuperheaters may be either of the direct contact or the noncontact type. The selection for any specific installation will be governed principally by the water available for use as the temperature regulating medium. This is not important with the noncontact type where a tubular heat exchanger is used since the steam to be desuperheated is separated from the cooling medium and the heat is transferred through the tubular wall. In some instances noncontact desuperheating coils may be installed in one of the boiler drums. The principal objection to this design is its relatively low capacity because parts must not occupy so much space in the drum that they cannot be assembled in place and must be small enough to handle conveniently through the boiler-drum manhole opening. In the direct-contact type, however, the cooling medium is injected into and mixed with the superheated steam to effect a reduction in temperature. The cooling medium is usually water, and it must therefore be comparatively free of solids. In most installations, condensate or feed water will be of sufficient purity to permit its use for this purpose.

A combination of adjustable burners, together with interstage desuperheating, is effective for maintaining constant steam temperature over a wide capacity range. With this arrangement, the response to changes in operating conditions is rapid, and deviation from present temperature can be held to less than plus or minus 5°F provided the load does not fall below superheater design point.

The use of desuperheaters has not been particularly popular with steam-plant operators even though several designs are available. Perhaps the reasons for this might be the following: With the direct-contact type where water is injected into the steam to effect temperature reduction, there is the ever-present possibility of moisture carry-over into the final superheating stage if the sprays are improperly adjusted or controlled. Also, the noncontact types are large heat-exchanger types of vessels suitable for h-p and high-temperature operation and are, therefore, bulky, heavy, costly, and subject to service interruptions common with equipment of this type

HEAT-RECOVERY EQUIPMENT

Heat-recovery equipment is apparatus forming part of a steam-generating unit but which is not part of the steam-generating or superheating sections. Feed-water-heating and air-heating apparatus, arranged to recover heat from flue gas leaving the steam-generating section of a unit, are known as "economizers" and "air heaters," respectively

Economizers. The most suitable material for h-p economizers, except as regards corrosion, is steel tubing because of the following advantages: thinner tube walls, smaller diameter, closer spacing, more compact arrangement of heating surface, better heat transmission for a given weight of surface, and lower cost.

The temperature of feed water entering any type of economizer should never be less than 125 F if sweating and resulting external corrosion are to be avoided

Internal corrosion in steel economizers is most frequently due to dissolved oxygen in the feed water which must, therefore, be properly conditioned to overcome this difficulty The method employed for this conditioning should be based on recommendations of those who are well qualified to advise on feed-water treatment Some form of deaerating heater may reduce the oxygen content to the required minimum for the specific operating conditions. In some instances supplementary chemical treatment may also be indicated

Steel-tube economizers are arranged in vertical or horizontal banks of tubes The vertical types are known as "integral economizers" and the horizontal types as "separate economizers."

Integral Economizers Integral economizers are used only in conjunction with bent-tube boilers They are arranged like an additional bank of boiler tubes and are located within the boiler setting Either the one- or two-drum type may be used. The one-drum type employs a lower drum from which the economizer tube bank extends upward to terminate in the rear steam-and-water drum of the boiler section In this design the feed water is introduced into the lower drum, makes one pass through the tube bank, and is then discharged directly from these tubes into the boiler drum. The flue gas is directed over the economizer tubes by properly located baffles. The upper drum of a two-drum integral economizer is divided into two compartments by a partition plate which likewise divides the tube bank into two sections. The feed water is introduced on one side of the partition and discharged from the other side through connecting tubes into the upper steam-and-water drum of the boiler. The water therefore makes two passes through the economizer in flowing from one compartment through part of the tube bank to the lower drum and then back through the remainder of the tube bank to the other compartment of the rear drum. An economizer arrangement of this type is shown in Fig 8-38.

Integral economizers have a low water velocity owing to multiplicity of parallel circuits and therefore also have a low waterside pressure drop. This characteristic is not conducive to good distribution or positive circulation but is favorable to internal oxygen corrosion unless suitable water conditions are continuously and carefully maintained.

Separate Economizers. Separate economizers are the most widely used type. They employ horizontal steel tubes arranged in staggered closely spaced rows, with the gas flowing transversely to the axis of the tubes. The water flows progressively through the tubes, upward or downward from row to row, and usually in counterflow relation to the gas. Upward flow of water is preferred by some engineers, particularly if there is any possibility of steam being generated in the economizer. Many downflow applications, however, are in successful use. These economizers are available in either the accessible type or the continuous-tube type which has no provision for access to the interior of the tubes.

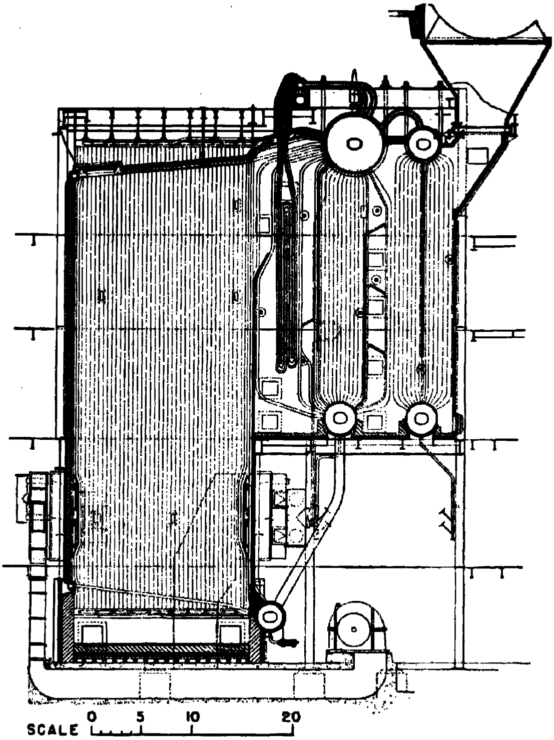

FIG. 8-38. One of several large steam-generating units equipped with two-drum integral economizer and arranged for tangential firing of blast-furnace and coke-oven gas. Capacity 200,000 lb of steam per hr at 325 psi and 700°F. (*Combustion Engineering—Superheater, Inc.*)

The *accessible type* is provided with flanged return bends or other means for ready access to the interior. It is used where feed water is poor, and the economizers, therefore, require periodic inspection and cleaning.

The *continuous-tube type* affords no access to the tube interiors and is used where good feed water makes internal cleaning unnecessary. The recent advent of inhibited hydrochloric acid cleaning for internal tube surfaces may lead to the adoption of this design for many installations where provision for mechanical tube cleaning would otherwise be very necessary.

An economizer is frequently located outside the boiler setting as a separately supported and encased unit. The stoker-fired steam-generating unit shown in Fig.

into which the heating surface of the regenerative mass is fitted. This surface is made up of thin metal sheets which are notched and undulated so that when compactly assembled space remains through which the air or gas can flow. The heating surface may also be arranged in two layers, as shown in Fig. 8-44, to minimize any replacement of the cold end which may become necessary owing to corrosion. Sheets in these two layers may be of different materials, each of which is best suited to the conditions which exist in its particular zone of application. Suitable seals are used to minimize leakage through the heater and thus maintain its performance.

Heaters of this type may be of either the vertical- or horizontal-flow design as best suits the particular application. They are compact since 1 in. in height of the particular heating surface used will recover approximately as much heat as a 1-ft length in a standard plate-type heater or 2 ft in a tubular-type heater with equivalent resistance to flow of gas and air. For the installation shown in Fig. 8-35, two vertical-flow heaters are used for each steam-generating unit, and they are located side by side above the boiler as shown. For the installation shown in Fig. 8-32, two vertical-flow heaters for unit are located in the rear of the boiler as shown.

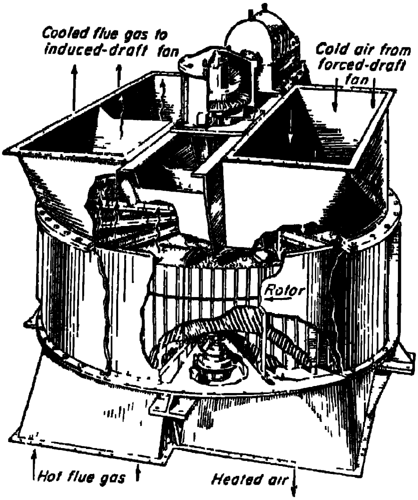
Cooled flue gas to induced-draft fan
Cold air from forced-draft fan
Rotor
Hot flue gas
Heated air

FIG. 8-44. Vertical-flow design of regenerative air preheater. *(Air Preheater Corp.)*

FURNACES FOR STEAM-GENERATING UNITS

To design furnaces for steam-generating units properly requires detailed knowledge of many influencing factors which vary from installation to installation. For this reason it is difficult to establish a complete set of standards which can be applied to cover any and all conditions. In material entitled "Fuel Preparation and Burning Equipment" the preferred limits of fuel-burning and heat-liberation rates for the various type, of equipment available were indicated. There was, however, no discussion as to some of the basic furnace design principles which govern in various types of installations. An abbreviated discussion of these will follow in order to provide a background which should be of some assistance when making comparisons or evaluations.

Single-retort Stokers. Furnaces for most single-retort stokers are of the simplest form because they are devoid of arches and consist essentially of four refractory walls (see Figs. 8-25 and 8-27). If the heat-liberation rate is limited to not more than 45,000 Btu per cu ft per hr, they will have reasonably low maintenance and very infrequent outages due to refractory failure. In some cases water-cooled side-wall clinker chills at the dump-grate line will minimize clinker adhesion. With some of the newer steam-generating units, water-cooled furnaces are a part of the standardized design. In practically all those now available, ample furnace volume is provided so that at full capacity heat liberation is well below that allowed for less favorable construction. The single-retort stoker can be substituted for the spreader type shown with the package-type unit of Fig. 8-33. In all cases the heat-liberation rate should be based on effective primary-furnace volume and not include parts of boiler passes or other similar ineffective zones.

Multiple-retort Stokers. Multiple-retort stoker furnaces, except for size, were originally quite similar in design to those used with the single-retort type. Application and use of water-cooled surface, however, enabled the combustion engineer to develop the full fuel-burning capacity of these machines in removing the restrictions imposed by all refractory-furnace construction. More recent design modifications, made possible through further extended use of water-cooled surface, have improved operation still more by providing added turbulence to the furnace gas to overcome stratification which existed because of slow rate of flow. Regardless of furnace design, the maximum continuous heat-liberation rate should preferably be kept below 35,000 Btu per cu ft per hr even though water cooling is used. The reason for this is that furnaces for stokers of this type are quite large, and, therefore, the ratio of wall area to volume is relatively low which, in turn, means somewhat higher furnace temperatures.

Furnace turbulence to break up gas stratification may be produced in a number of ways. Over-fire air and steam jets are effective, particularly when located in the bridge wall and directed forward to blow above and parallel to the fuel bed. They may be easily and inexpensively applied to existing installations. Their use, however, requires power to drive the over-fire-air fan or exhaust not less than 1 per cent of the steam generated up the stack. For new installations it is possible to build in furnace turbulence so that it may be continually produced without an expenditure of power or steam. The means available for this purpose is the rear-arch type of construction, well illustrated in Fig. 8-45. With this design, an arch is located parallel with and in close proximity to the fuel bed and extends forward from the rear wall to cover approximately fifty per cent of the stoker length. Its action is to produce turbulence and mixing by directing the products of combustion, rising from the rear portion of the fuel bed, forward at relatively high velocity and causing them to intermingle with those rising from the forward part of the fuel bed. This results in elimination of all stratification and provides uniformity of gas composition and temperature at entrance to heat-absorbing surfaces of the boiler because combustion is completed in the lower furnace zone.

Spreader Stokers. Spreader stokers are usually installed in the conventional open-type furnaces where heat-liberation rates, at continuous capacity, should preferably be held below 35,000 Btu per cu ft per hr. The operation of these stokers is sensitive to the amount of water-cooled surface which may be installed. Usually bare tubes on relatively wide centers are used to avoid the tendency toward smoke which is frequently present when too much cooling effect is used. Owing to the open type of furnace, as well as fuel-bed characteristics and fuel sizing, there is always considerable stratification of the furnace gas. This may be effectively broken up through the use of steam jets or high-velocity over-fire-air jets.

Spreader-stoker fuel beds are characteristically sensitive to air flow. They operate best when approximately 90 per cent or more of the air required for combustion is flowing through them. Because of this the amount of over-fire air that can be used for cinder-return systems and turbulence-producing jets is definitely limited in volume and must therefore be supplied at relatively high pressure to be effective. When steam jets are used they should be of the non-air-aspirating turbulence-producing type so that air supply is under control, with the maximum possible amount passing through the fuel bed.

The steam-generating unit shown in Fig. 8-36 is representative of the dumping type spreader-stoker application for the small industrial plant. For larger capacity, the continuous-discharge design as shown in Fig. 8-46 finds increasingly frequent applications, particularly where low-quality fuels are available.

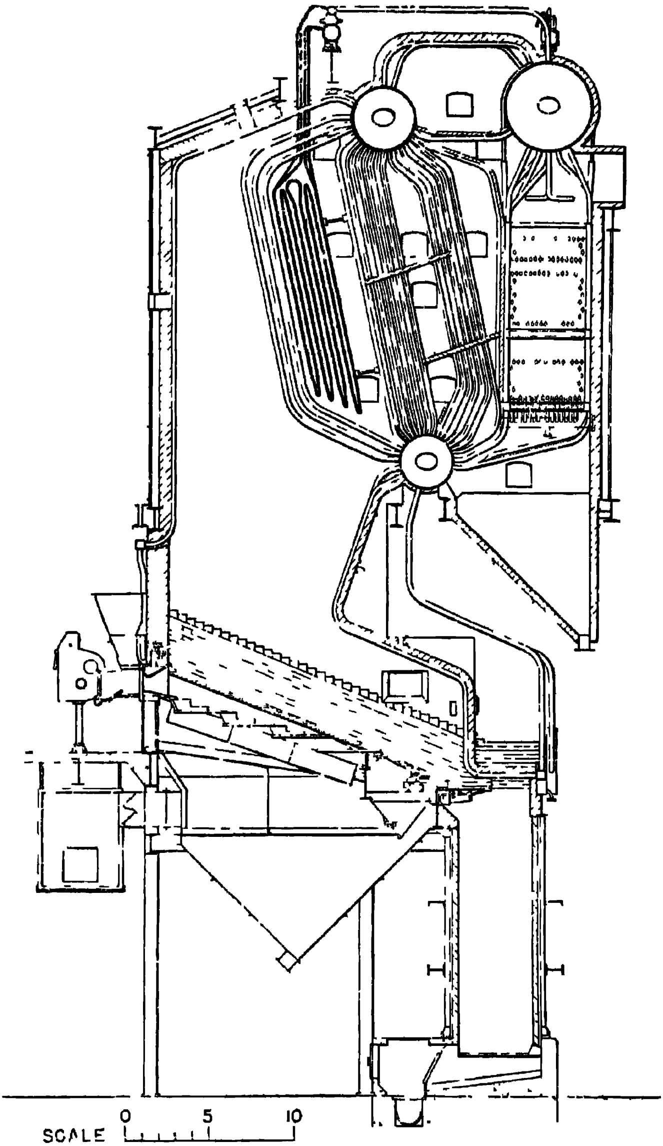

Fig. 8-45. One of several steam-generating units each with a capacity of 150,000 lb of steam per hr at 405 psi and 725°F. A continuous-discharge multiple-retort stoker with rear-arch furnace is used. Water walls are applied to the furnace walls at the sides, rear, and front. (*Combustion Engineering—Superheater, Inc.*)

Traveling- and Chain-grate Stokers. The basic furnace contour for traveling- and chain-grate stokers is practically the same, regardless of fuel burned Modifications in height and length of arches are, of course, necessary when the fuels have such widely varying characteristics as those between anthracite and free-burning bituminous coal or coke breeze and lignite No attempt will be made to fix definite standards for dimensions Several typical furnace arrangements will be shown, and furnace-heat-liberation-rate limits will be presented merely as a guide to what constitutes a good furnace

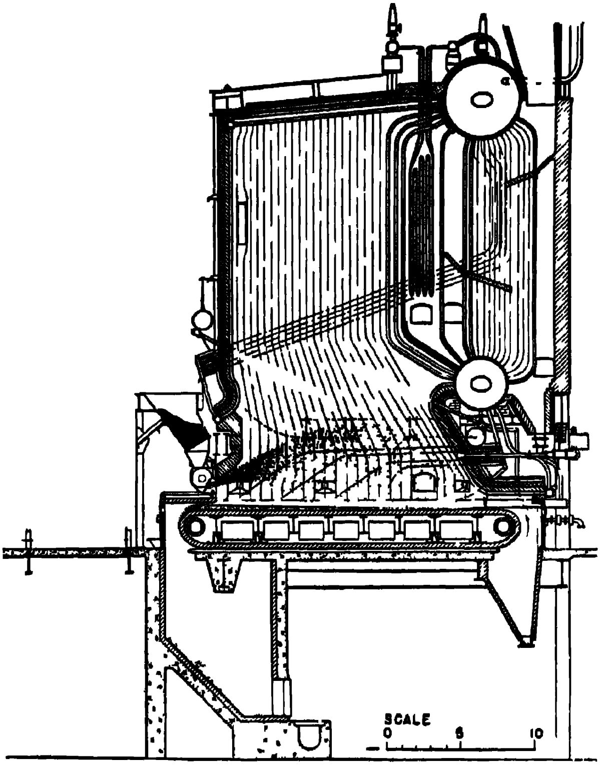

Fig 8-46 A two drum steam-generating unit equipped with continuous-discharge-type spreader stoker Capacity 150 000 lb of steam per hr at 450 psi and 750°F (*Combustion Engineering—Superheater Inc*)

Many of the earlier installations were of the *front arch* type in which the arch extended rearward from the front wall to cover from 30 to 40 per cent of the stoker length The primary purpose was to assure prompt and rapid penetration of ignition by reflecting heat onto the incoming green fuel bed The height of the arch above the fuel bed together with its length were a function of volatile content in the fuel and the maximum rate at which it was burned Front-arch furnaces almost invariably require over-fire air jets to break up gas stratification which results from low rates of flow A front-arch furnace designed for free burning bituminous coal is shown in Fig 8-47 With this installation a side-wall clinker chill and an overhanging water back are used to minimize refractory maintenance Over-fire air jets are directed angularly downward toward the rear of the furnace from a point just above the nose of the arch

At present a number of installations operating with Middle Western bituminous coal have furnaces in which arches cover a portion of both the front and the rear of the stoker This *combination-arch* arrangement did much to further stabilized ignition and improve combustion not only with these Middle Western fuels but also with anthracite, coke breeze, and lignite

Originally, when burning such undersize fuels as anthracite No. 4 buckwheat and river coal in front-arch furnaces, there was considerable carbon loss in fly ash and ashpit, even at low combustion rates The combination-arch furnace provided only a partial remedy Additional experimenting however, finally resulted in development of the *rear-arch* furnace With this design, well illustrated in Fig 8-39, stability of ignition is assured by utilizing the principle of raining particles of incandescent fine coal onto the incoming fuel This is accomplished through a modification in air distribution, coupled with the use of a long, low arch over the rear of the stoker

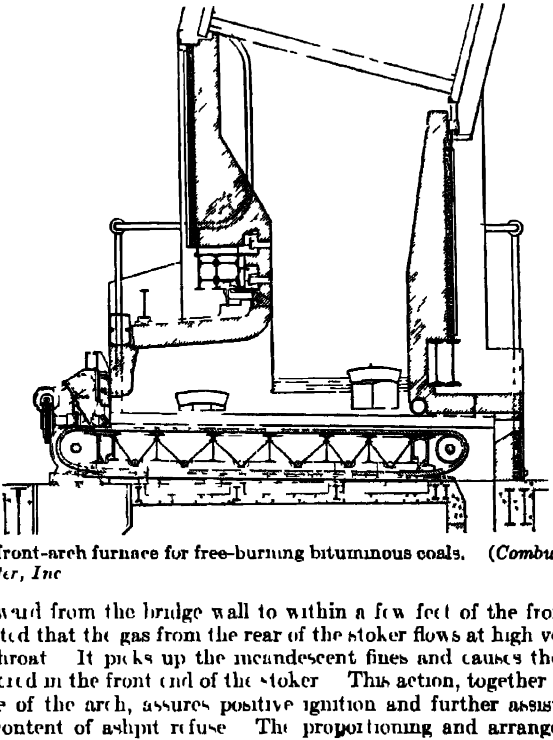

Fig. 8-47 A front-arch furnace for free-burning bituminous coals. (*Combustion Engineering Superheater, Inc*

extending forward from the bridge wall to within a few feet of the front wall The arch is so located that the gas from the rear of the stoker flows at high velocity toward the furnace throat It picks up the incandescent fines and causes them to be continually showered in the front end of the stoker This action, together with radiation from the nose of the arch, assures positive ignition and further assists in reducing combustible content of ashpit refuse The proportioning and arrangement of arch surface are governed by the fuel to be burned, capacity developed, and stoker proportions The length will vary from 60 to 75 per cent of stoker length, the height above the grate from 2 to 4 ft, and the throat opening from 3 to 7 ft Arches of this type lend themselves to the maximum of water cooling, and the amount of surface applied will be governed largely by the operating conditions

Over-fire air may be used to advantage even with rear-arch designs and their built-in furnace turbulence The operation of these jets maintains a high rate of turbulent mixing during low-load operation, when gas volume and velocity fall below their effective values

For the furnace design described the maximum continuous furnace-liberation rates should preferably be held to between 40,000 and 55,000 Btu per cu ft per hr. Combustion rates per pound of dry coal per square foot of grate per hour will vary from 30 for anthracite and coke breeze to 50 for subbituminous coal and lignite.

Pulverized Coal. The present-day pulverized-coal-fired furnace, regardless of firing method employed, is of water-cooled construction. Size of unit together with its pressure and temperature, space available, fuel characteristics, ash-fusion temperature, type of burner, and method of ash removal are all contributing factors. They not only determine furnace volume and extent of water cooling employed but also ultimate design of the steam-generating unit as a whole. The maximum continuous rate of heat liberation should preferably be held to between 15,000 and 22,000 Btu per cu ft per hr.

The type and number of burners used, as well as their arrangement, will determine furnace dimensions. Minimum front-to-rear depth is governed by the distance between burners and front row of tubes in boiler bank, which is required to assure completion of the combustion process. Width is governed by the horizontal spacing plus the distance that must be provided between the burner and furnace side walls to minimize the effect of any flame impingement. The height is selected so that the indicated furnace volume is obtained, but in no case may it be less than the minimum allowable distance between the bottom burner row above the furnace water screen, water-cooled hopper bottom, or slag hearth, plus the vertical spacing of burner rows, plus minimum allowable distance between top burner row and roof.

Furnace volume is usually determined by using the allowable heat-liberation rate for ash-fusion temperature and quality of the coal to be employed. Wherever possible, there should be provided a combination of volume and water-wall cooling that will result in a furnace outlet temperature which is below ash-fusion temperature of the lowest quality fuel it may be necessary to use. With coals having a high percentage of low-fusion ash, it is necessary to use greater furnace volume so as to increase the amount of water-wall surface that can be installed and thereby secure comparatively low gas temperature at the entrance to the boiler bank. With high steam temperature, the requirements of the superheater may govern furnace outlet temperature.

After required volume and minimum dimensions have been determined, the furnace is proportioned so that length, height, and width finally chosen will provide the most suitable design for available space conditions.

Finally to complete the furnace it is necessary to select the type of furnace-bottom construction that will be best suited to the installation

If the ash is to be collected and removed in a dry state, either the bottom water-screen or the hopper-bottom furnace construction should be used With the *water-screen* type of installation a series of tubes, tied into the circulation system of the water walls and boiler, is located above the furnace floor to provide a shielded and cooled zone in which the collected ash is maintained at a temperature below its fusion point. This construction is used for the units shown in Figs. 8-29, 8-31, and 8-42. In the *hopper-bottom* design the front- and rear-wall tubes are bent at their lower end to provide a floor slope greater than the angle of repose for ash. A self-cleaning water-cooled floor is thus formed. A transverse gap is provided in the hopper valley through which the ash is discharged to an enclosed pit This type of construction is particularly well adapted for the application of vertically adjustable burners as shown in Fig 8-37.

When the ash is to be collected and drained from the furnace in molten form, the *continuous-slag-tap* construction is used. With this design a water-cooled floor, covered with a layer of special refractory, forms the hearth on which the molten slag is collected. A slag-drip opening is located in the bottom near the center of the hearth. Around this opening a water-cooled ring is used to form a slag dam and drip ledge so that a relatively constant thickness of slag is maintained over the hearth. Any excess slag accumulation continually drips through this opening and is quickly quenched and shattered into granular form by dropping into a suitably enclosed water-filled chamber. This construction is illustrated in Figs. 8-32 and 8-35.

Liquid and Gaseous Fuels. When consideration is being given to the installation of oil- or gas-burning equipment, it is well to remember that results obtainable depend more on furnace design than on the burner. For best results, however, the two should be properly coordinated. Practically all units for either of these fuels are arranged for the alternate use of the other. While their appearance may therefore be the same, there is a marked difference in heat absorption throughout. With oil firing there is considerable radiant heat transfer to the furnace cooling surfaces owing to flame luminosity. With natural gas the flame is nonluminous, and therefore there is more convection heat transfer in the boiler section. This is particularly significant in superheater design, and if uniform performance is required of either fuel some means must be provided for by-passing a portion of the products of combustion around the superheater.

Furnace designs frequently follow accepted pulverized-fuel practice. The extent of water cooling is governed by temperature requirements at the superheater inlet, and furnace heat-liberation rates are preferably held to between 20,000 to 30,000 Btu per cu ft per hr at maximum continuous load point.

Types of Wall-cooling Surfaces. The most effective types of furnace wall-cooling constructions are ones which employ bare tubes, arranged to satisfy the requirements established by the several variables on which design is to be based.

The *bare-tube type* of construction may be defined as one in which the wall-cooling surface is composed of plain bare tubes or bare tubes having extended metallic surfaces welded to them. The cooling surface is directly exposed to furnace temperature and capable of absorbing heat at an extremely high rate. When plain tubes are used, they may be spaced on uniformly close or wide centers to cover the furnace wall areas that require protection. In the bare-tube types, with welded metallic extended surfaces or fins, there is unit metal construction without heat gaps capable of the same high rate of heat absorption as the plain tube. The two fins, each 1 in. wide, are located on diametrically opposite sides of the tube. The tubes are then spaced so that the gap between adjacent fins is $\frac{1}{8}$ in. Thus there is a continuous metallic surface forming the interior lining of the furnace.

With any type of water-cooled construction, downcomer tubes must be provided to supply the water-wall headers. Riser tubes must also be provided to complete the circuit and return the steam generated in the walls to the boiler. The flow areas of these downcomers and risers depend on the amount of wall exposed, the effective head of water on the wall, and the amount of heat absorbed. Illustrations of the several large steam generating units clearly show the extent of water cooling used as

Table 8-33

1. Fuel characteristics	Types of fuel available to plant: Ash content Ash-fusion temperature Comparative fuel prices Comparative operating efficiency Lower grade secondary fuel which it may be necessary to use if primary fuel source "dries up" Comparative operating costs with the available fuels: For handling fuel For handling refuse For repairs and maintenance For auxiliary power
2. Capacity and steam conditions	Maximum continuous capacity Extent, duration, and frequency of peak loads Design or operating pressure Steam temperature if superheat is required Feed-water temperature
3. Space conditions	Cubical contents of space available or required Shape of space
4. Cost..	Comparative cost of designs considered High- or low load-factor plant Value of reliability and flexibility Value of incremental increase or decrease in efficiency Operating costs for use in economic evaluations
5. Individual preference	Experience or familiarity: In a particular plant or industry With a certain type of fuel or equipment Uniformity with old equipment already installed should not jeopardize value of plant extension or betterment

well as the arrangement of downcomer and riser systems which tie the wall surface into the boiler circuit

EQUIPMENT SELECTION

The ability properly to select and coordinate equipment so as to produce a well-designed steam-generating unit requires not only a broad background of experience which covers each separate element and its relation to the others but also a wide variety of available designs in order that the particular needs of a given project may be satisfied No attempt will be made to discuss the many details that enter into equipment selection The principal factors, however, which exert most influence, together with some of the secondary ones, are outlined in Table 8-33

Table 8-33*a*. Fuel-burning and Steam-generating Equipment Combinations

Boiler types	Continuous capacity ranges Pounds of steam per hour				Design pressure range	Steam temperature range
	1,000 to 15,000	15,000 to 35,000	35,000 to 200,000	150 000 to 1 000 000		
Firetube	Hand fired Single retort Multiple-retort Spreader stoker Traveling-grate Oil Gas	▨	▨	▨	Up to 250 psi	Up to 150 deg F superheat
3 Drum low-head		Single-retort Multiple-retort Spreader stoker Traveling grate Oil Gas	▨	▨	Up to 400 psi	Up to 150 deg F superheat
2-Drum * vertical			Multiple-retort Spreader stoker Traveling grate Pulverized coal Oil Gas	Traveling grate ** Multiple retort ** Continuous ash discharge spreader ** Pulverized coal Oil Gas	Up to 1000 psi	900 F
3 or 4 Drum vertical	▨				Up to 1500 psi	925 F
Sectional header	▨			▨	Up to 2000 psi	900 F
Special utility type	▨	▨	▨	Pulverized coal Oil Gas	Up to 2500 psi	1050 F ***

* Up to 225,000 lb per hr on Coal and 300 000 lb per hr on Oil and Gas
** Occasionally for capacities exceeding 200 000 lb per hr
*** Present maximum used Design and materials available for higher temperature

The most frequently encountered combinations of fuel-burning and steam-generating equipment, together with capacity, pressure, and steam-temperature ranges, are shown in Table 8-33*a* This table is intended to serve no purpose beyond providing a convenient comparison of the various equipment combinations used for the several operating conditions There is no satisfactory short cut to equipment selection, and final decision should, therefore, always reflect the seasoned judgment of engineers who have specialized in steam-plant engineering

LOCAL AND REMOTE GRAPHIC POWER MEASUREMENTS AND TOTALIZING*

BY W. E. PHILLIPS

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Graphic instruments, which make a graph of an electrical quantity as a function of time, have become increasingly important in the study, operation, and control of electrical power systems and also to consumers of electrical power. The development of new principles, the demand for increased speeds of response, and the adaptation of established principles to new designs have resulted in a wide diversity of instruments. It is the purpose of this subsection to describe well-established principles and designs used in the graphic metering of electric power and related quantities and the telemetering of these measurements.

Graphic meters are commonly referred to as "recording instruments" or by the simpler term "recorders."

Telemetering of electric power and related quantities will also be described. The American Institute of Electrical Engineers defines telemetering as the indicating, recording, or integrating of a quantity at a distance by electrical translating means. It is possible to extend current and potential transformer secondary or similar leads so that a reading may be obtained at a considerable distance, but the extension of leads in this manner will not be considered telemetering. Totalizing of electric-power measurements will be defined for purposes of this text as the recording of the algebraic sum of several individual measurements. In general, individual and totalizing recorders will be at the same location.

UTILIZATION OF GRAPHIC POWER MEASUREMENT

Graphic power measurements are found to be *necessary* to the operation of electrical power systems and industrial plants under normal as well as abnormal conditions. Their use can be broadly classified under the headings of electrical generating stations, electrical power systems, and industrial-plant operation.

NOTE: For purposes of this discussion, the term "electrical power system" is defined as a multiplicity of electrical generating stations operated as an integrated system under the supervision of a central-system operator.

Operation of Electrical Generating Station. The utilization of graphic power measurements in individual electrical power plants is widespread, and the following applications are typical:

1. The graphic recorder is used by the operator to ensure that the station is delivering its assigned schedule of electrical load.
2. The graphic recorder is an excellent medium for operating a remote indicating system of which the Selsyn system is typical. A system of this type provides a visual indication for the turbine and boiler rooms of the station load by means of large-dial indicators.
3. Automatic control of unit and station load has become increasingly important, and devices have been added to the graphic instruments to provide the necessary intelligence and limiting features for the automatic control.
4. Graphic power records are helpful in the economic studies of unit and station efficiencies since the graphic record of power output can be compared with the charts of steam flow, temperature, pressure, etc.

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* Superior numbers refer to specific references listed at the end of this subsection.

System Operation. The practice of telemetering electrical power and allied quantities to a central point, usually the system operator's office, is now considered to be necessary for the intelligent and efficient operation of power systems. Telemetering has been termed the "eyes" of the system operator since it enables him at a glance to determine that the electrical power system is functioning as desired.

Some of the advantages of having station generation and major tie-line loads telemetered to the system operator's office and the totalizing of these loads to obtain total system generation,[*] total interchange,[†] system "net" load,[‡] etc., are as follows:

1. With the enormous growth of power systems, both as to the number and the capacity of the generating stations, most of the system operator's time would be consumed in obtaining telephoned data if telemetering were not provided. Telemetering reduces verbal communications by at least 50 per cent, thus relieving both system and station operators for other duties without constant interruption Under extreme conditions, when verbal communication between the station and system operator has been interrupted, it has been possible to operate for more than a day with telemetering as the only link

2. Telemetering provides a means of loading a system on a prearranged basis Such a loading means might conceivably consist of telemetering to each generating station the total system load and assigning to each station a scheduled output for each system load. Thus each station operator can determine immediately that his station load is correct.

3. Telemetering is used to give the system operator a continuous record of the individual tie lines to adjacent systems, and by totalizing the algebraic sum of these tie lines a reading of net interchange is obtained. Thus the system operator is informed at all times as to the interchange loadings

4. The value of telemetering at times of system trouble or under abnormal conditions is not subject to any strict evaluation. It has been found in some cases that major interruptions and trouble have been prevented because the system operator has had prompt and complete information regarding the loads on different parts of the system. With this information it has been possible intelligently to isolate parts of the electrical system and continue operation without serious outages

5 The records of the graphic instruments are found to be of value when economic and load-growth studies are being made of a power system.

6. The trend in automatic load-frequency control at the present time is to locate the master control elements at the system operator's office. Thus telemetering and totalizing perform many functions in connection with the automatic load frequency control.

Industrial-plant Operation. The utilization of graphic meters for measuring electrical power and related quantities in industrial plants is legion, and their use is an absolute necessity if optimum economic and efficient results are to be approached with respect to the cost of electrical power

In general, most electrical rates consist of an "energy" and a "demand" charge In addition, the rate schedule may include other charges based on power factor, fuel costs, etc.

The energy charge is based on kilowatthours consumed during the month, and usually the cost per kilowatthour decreases with increased consumption

The demand charge is usually based on the maximum rate at which kilowatthours are used over a specified period, usually 15 min for industrial loads. This charge is intended to cover the fixed costs of the equipment which the utility company must provide to supply this maximum demand A high maximum demand is often the result of a few peaks caused by abnormal and perhaps avoidable conditions In some cases the monthly demand charge for an entire year may be based on the maximum demand which was registered during a single abnormal period Thus the cost of a single peak may be enormous.

Graphic power and power-factor recorders can be used to study and also provide a continuous record of the plant load and power factor. In many instances a new peak demand may be avoided by disconnecting a portion of the electrical load for only a few minutes. In other cases possibly a rescheduling of work to prevent coincident peaks in different departments may produce the desired result.

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[*] Total system generation is the sum of the loads on all generators on the system

[†] Total interchange is the algebraic sum of power interchange with adjoining power systems

[‡] System net load is the total system generation plus the power being fed into the system minus the power being fed out of the system over its interconnecting tie lines with adjacent systems

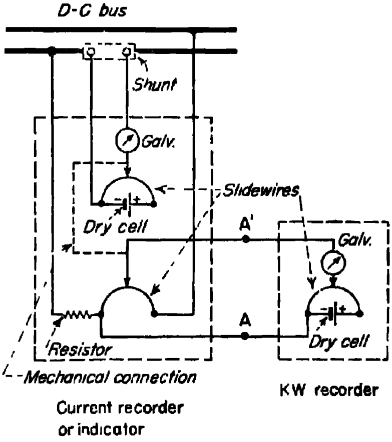
Power factor may be improved by ensuring that motors are not oversized, by installing overexcited synchronous motors, or by installing capacitor banks.

The graphic power meter is also useful in many processes. A typical example is one which occurs in the chemical industry in connection with stirring. As the chemicals are stirred they go through a critical point where the viscosity changes rapidly. A record of the power input to the motor driving the stirring mechanism provides an excellent indication of when this critical point is reached.

PRIMARY ELEMENTS USED FOR GRAPHIC MEASUREMENTS

Direct Current and Voltage. The d'Arsonval (moving coil, permanent-magnet field) measuring element is used almost exclusively for the graphic metering of direct current and voltage. This type of primary element has been described widely in existing literature, and it is not felt to be necessary to describe it in this text.

Direct-current Watts. Two methods are commonly used for the graphic measurement of d-c watts. The first method employs an electrodynamometer as the measuring element. The current coil is connected to the output of a shunt which is in series with the load being measured. The potential coil is connected in the conventional manner. Since current is drawn over the leads from the shunt, it is necessary that the meter calibration take into account the resistance of these leads. Any change in the shunt-lead resistance will introduce an error in the reading. Errors are also introduced because of residuals and stray field effects so that the order of accuracy is lower than when an electrodynamometer is used for a-c measurements.



8-48. Measurement of d-c watts

A second method of measuring d-c watts, which overcomes the objections of having to take into account shunt-lead resistance and results in accuracies of high order, consists of two recorders connected as shown in Fig. 8-48. The instrument that measures current may be an automatically balanced indicator rather than a recorder.

The current recorder or indicator operates on the potentiometer principle from the millivolt drop of the current shunt. Since it is a conventional potentiometric measurement, the accuracy is independent of the lead resistance between the shunt and the recorder or indicator. A retransmitting slide-wire is mounted on the current recorder or indicator and moves with and assumes the same position as the recorder or indicator measuring slide-wire. A potential proportional to the d-c bus voltage is applied to this retransmitting slide-wire.

The output potential from this slide-wire, *i.e.*, the potential between *A-A'* (Fig. 8-48), is proportional to the current recorder or indicator position and the d-c bus voltage. More exactly, it is proportional to the product of the current and voltage which is watts. It should be noted that the voltage across the retransmitting slide-wire is independent for all practical purposes of the resistance of the leads connecting the slide-wire to the bus potential since the slide-wire and its series coil, which are made of extremely low-resistance temperature-coefficient material, have a total resistance of approximately 60,000 ohms.

The output of this retransmitting slide-wire is recorded potentiometrically by the kilowatt recorder (Fig. 8-48).

The measurement of kilowatts is based entirely on the potentiometric principle, except for the supplying of the bus potential to the retransmitting slide-wire, and consequently the accuracy is inherently of high order. As was pointed out previously, the method of supplying the potential to the slide-wire introduces no practical error.

Alternating Current. The primary elements used in connection with graphic measurements of a-c power and related quantities are the thermal wattmeter, the electrodynamometer, the induction-disk watt-hour meter, and the Kelvin balance. The principles and theory of the latter three have been thoroughly explained in previous literature, and it is not necessary that they be repeated.

The thermal wattmeter, which is the newest of the primary elements adapted to graphic metering, produces a d-c millivoltage proportional to input watts. It is in all respects a true wattmeter—the polarity of its output reverses with reversal of watts; it measures the product of voltage, current, and the cosine of the angle between them; it is independent of wave shape to a high degree because of the fact that its output depends on heating (root-mean-square) values.

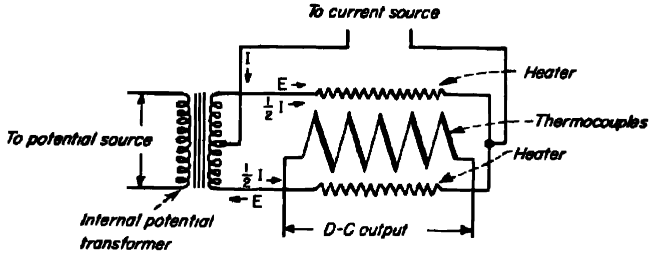


FIG. 8-48a. Thermal wattmeter.

The theory of one type of thermal wattmeter, commonly referred to as a "thermal converter"—which is typical of all—will be readily understood from the following explanation.

A single-element thermal converter, complete with its internal potential transformer, is shown in Fig. 8-48a. Two such elements are used for conventional three-phase, three-wire metering.

The current and potential circuits are connected to the thermal converter as indicated in Fig. 8-48a. The current I divides as shown so that one-half flows through each heater. The potential causes a current designated as E to flow through both heaters in series. It should be noted that these two currents are in the same direction in the top heater and in opposite directions in the lower heater. It will be clear by inspection that the vector sum of the currents in the upper heater will be greater than that in the lower heater when the phase angle between the two currents I and E is less than 90 deg. Thus more heat will be produced in the upper heater than in the lower heater since the heaters have equal and constant resistances. Small thermocouples are located adjacent to the heaters, and the thermocouple junction at the upper heater is raised to a higher temperature than that located at the lower heater. The output emf of the thermocouples is proportional to this difference in heating which in turn is proportional to watts input to the thermal converter. This is demonstrated mathematically as follows:

Using E as a reference vector, $\dot E$ and I may be expressed as follows when the current is leading the voltage by θ°.

$$\text{Vectorial } E = \dot E = E + j0$$
$$\text{Vectorial } I = \dot I = I\cos\theta + jI\sin\theta$$

The current in upper element is

$$\dot E + \tfrac{1}{2}\dot I = \sqrt{(E + \tfrac{1}{2}I\cos\theta)^2 + (j\tfrac{1}{2}I\sin\theta)^2}$$

The heating in the upper element is proportional to the current squared. Thus

$$\text{Heating in upper element} = (E + \tfrac{1}{2}I\cos\theta)^2 + (j\tfrac{1}{2}I\sin\theta)^2 \qquad (1)$$

The current in the lower element is

$$\dot{E} - \tfrac{1}{2}\dot{I} = \sqrt{(E - \tfrac{1}{2}I\cos\theta)^2 + (-j\tfrac{1}{2}I\sin\theta)^2}$$

The heating in the lower element is proportional to the current squared

$$\text{Heating in lower element} = (E - \tfrac{1}{2}I\cos\theta)^2 + (-j\tfrac{1}{2}I\sin\theta)^2 \qquad (2)$$

The difference in heating is proportional to Eq. (1) − Eq. (2) =

$$(E^2 + EI\cos\theta + \tfrac{1}{4}I^2\cos^2\theta - \tfrac{1}{4}I^2\sin^2\theta) - (E^2 - EI\cos\theta + \tfrac{1}{4}I^2\cos^2\theta - \tfrac{1}{4}I^2\sin^2\theta) = 2EI\cos\theta \qquad (3)$$

Since 2 is a constant of calibration, the difference in heating is proportional to the product of E, I, and the cosine of the angle between them, which is power.

CONNECTION OF PRIMARY ELEMENTS

Alternating-current Watts. The connection of the primary elements for measuring polyphase a-c power is subject to considerable latitude on the part of the engineer. In general, the problem resolves itself into the equating of cost against required accuracy, keeping in mind the degree of unbalance between phases which may be experienced.

Totalizing current transformers are sometimes used so that the currents of several circuits may be totalized. This permits the use of only one primary measuring element for the measurement of several circuits. The use of totalizing transformers introduces certain hazards in that several current-transformer secondary circuits are brought into a common piece of equipment, *viz.*, the totalizing transformer. The possibility of trouble spreading from one secondary circuit to another is thus increased. Also the use of totalizing transformers is limited owing to the fact that all circuits must be in parallel, *i.e.*, supplied from a common bus.

Many different and special connections have been used for measuring a-c power. However, the connections employed in most cases are those described in the following paragraphs. Only three-phase metering will be considered since practically all graphic metering is confined to three-phase systems.

The fundamental principle upon which polyphase-power measurement is based is that enunciated by Blondel. This theorem states that for a system of N conductors the total power can be measured by means of $N - 1$ wattmeter elements.

As an illustration of this principle, if a three-phase system having a grounded neutral return is examined, it will be seen that it is in reality a four-wire system, and consequently three elements are required for correctly measuring the power under any condition of load unbalance. If we consider the above system further and state as a premise that no current flows through the ground-return circuit, it can be considered to be a three-wire system, and the power can be measured correctly by two wattmeter elements. As a further step in considering such a system, if we state that the load on all three phases is identical, *i.e.*, balanced voltages and currents and the same value of power factor, one element measuring the power in one phase only has a constant relation to the total three-phase power and can be used as measure of total three-phase power.

The connections and vector diagrams for these types of metering are shown by Figs. 8-49*a*, 8-49*b*, and 8-49*c*.

A modification of a two-element meter is quite often employed when measuring the power in a three-phase circuit with a grounded neutral. This is commonly referred to as $2\tfrac{1}{2}$-element metering and has the advantage that it measures the three-phase power correctly for unbalanced currents.

The connections and vector diagram for this type of metering are shown by Fig. 8-50. The vector diagram shows balanced voltages. The currents are unequal in magnitude and displaced by different phase angles with their respective voltages.

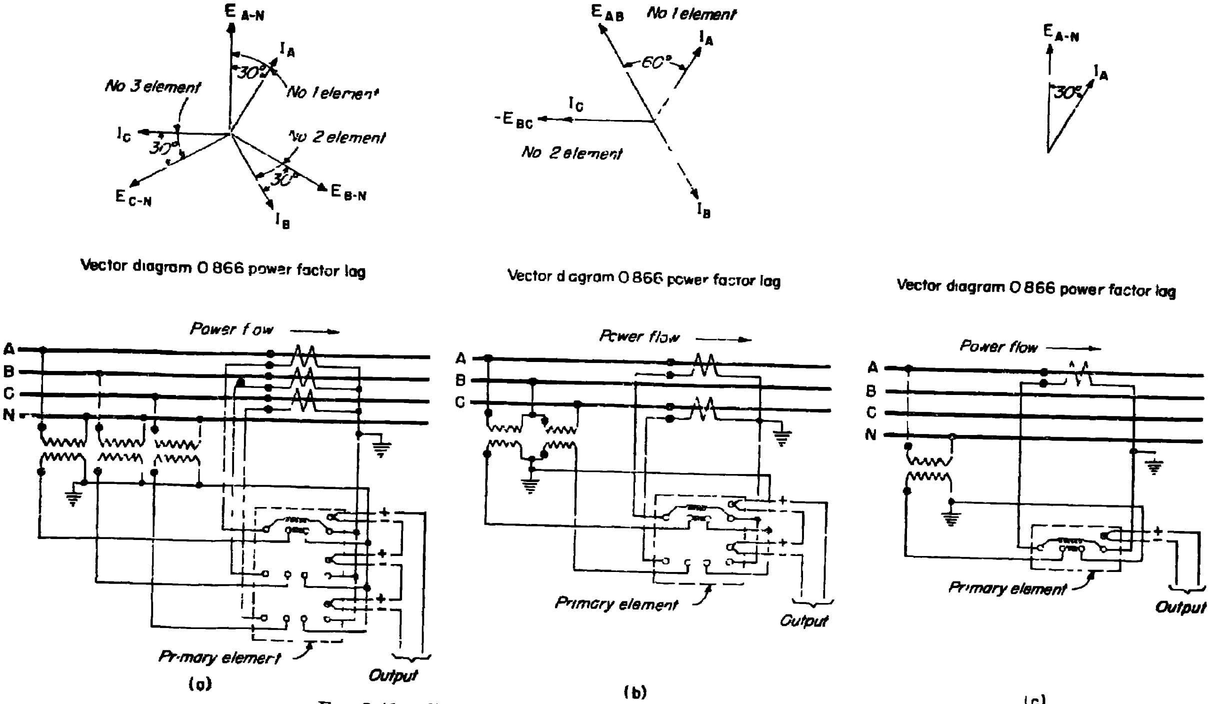


Fig. 8-49 Connections for measurement of three-phase power.

The currents in phases 1 and 3 react with their respective voltages, and the current in phase 2 is reversed and reacts with the potentials of phases 1 and 3.

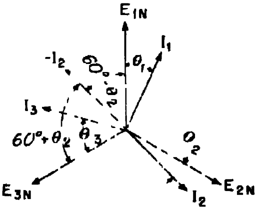
The metering can be expressed as follows:

$$\text{Metered phase 1 power} = E_{1n}I_1 \cos\theta_1 \tag{4}$$

$$\text{Metered phase 3 power} = E_{3n}I_3 \cos\theta_3 \tag{5}$$

$$\text{Metered phase 2 power} = E_{1n}I_2 \cos(60^\circ - \theta_2) + E_{3n}I_2 \cos(60^\circ + \theta_2) \tag{6}$$

If the voltages are equal in magnitude, then as far as magnitude is concerned we can write $E_{1n} = E_{3n} = E_{2n}$, and Eq (6) reduces to $E_{2n}I_2 \cos\theta_2$. The total for the three phases is the sum of the power measured on the individual phases which in this case is correct for total three-phase power.



Vector diagram

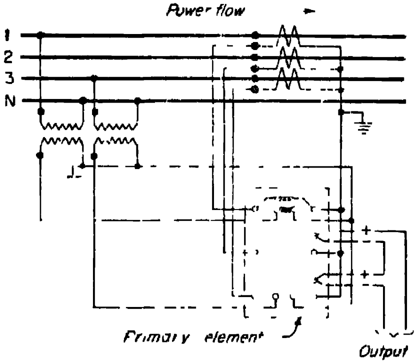
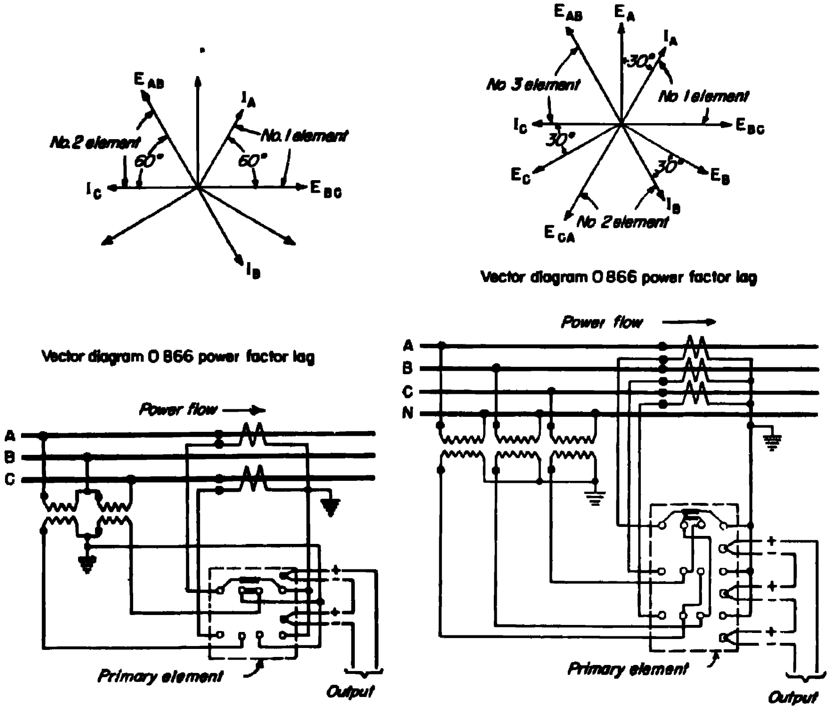


Fig. 8-50. Two and one-half-element metering.

Reactive Volt-ampere Measurement. As in the measurement of a-c watts there is considerable latitude in the connections which may be employed in the measurement of reactive volt-amperes, but the method which is used universally for the measurement of reactive volt-amperes on three-phase circuits is that of cross phasing the potential circuits with respect to the current circuits. This cross phasing may be accomplished merely by connecting the voltage coils across the phases opposite to that in which the current coil is connected as shown by Fig. 8-51*a* and 8-51*b* or by using phasing transformers as shown by Fig. 8-52*a* and 8-52*b*. The latter method, which is most generally employed, has the advantage of greater accuracy but is somewhat more complicated.

Unfortunately, there is no method available for accurately metering reactive volt-amperes of a three-phase circuit when both currents and voltages are unbalanced. A detailed analysis of the errors is beyond the scope of this section. The errors arising from unbalanced load conditions are most readily determined by the use of symmetrical components. This is covered in a very comprehensive manner by A. E. Knowlton.[1]

Power-factor Measurements. Power-factor meters at the present time are most generally used in connection with the metering of industrial loads. (On power systems power-factor meters have been, to a large extent, replaced by kvar meters.) Power-factor meters are (1) accurate only on balanced loads, (2) primarily for use on circuits where the power flow is in one direction, and (3) their sensitivity decreases as the current flow decreases. Crossed-coil electrodynamometer type of meters are used quite generally for measuring power factor. They usually employ two potential circuits and one current circuit.



Note.- Registration must be multiplied by 0.866

Note.- Registration must be multiplied by 0.577

(a) (b)

FIG. 8-51. (a) Reactive volt-ampere measurement of three-phase, three-wire circuit using cross phasing. (b) Reactive volt-ampere measurement of three-phase, four-wire circuit using cross phasing.

A method employing a thermal converter, which is connected in the same manner as a two-element wattmeter, measures the ratio of the millivolt outputs of the two elements. The connections of the primary element and the recorder for measuring the ratio of the millivolt output is shown by Fig. 8-53.

The output of the element 1 is proportional to $EI \cos (\theta + 30°)$, and the output of element 2 is proportional to $EI \cos (\theta - 30°)$. When θ is zero (unity power factor), the outputs of the two elements are equal (balanced load conditions assumed), and the ratio of the two outputs is unity. Thus the recorder balances at its midpoint. This mid-point on the scale is marked unity power factor. When the power factor is 0.5 lag, θ is 60 deg, and the output of element 1 is zero. Thus the recorder moves to the high end of the scale which is marked 0.5 power factor—lag. When the power is 0.5 lead, the conditions of output are reversed, and the recorder moves to the low end of the scale which is calibrated 0.5 power factor—lead. At intermedi-

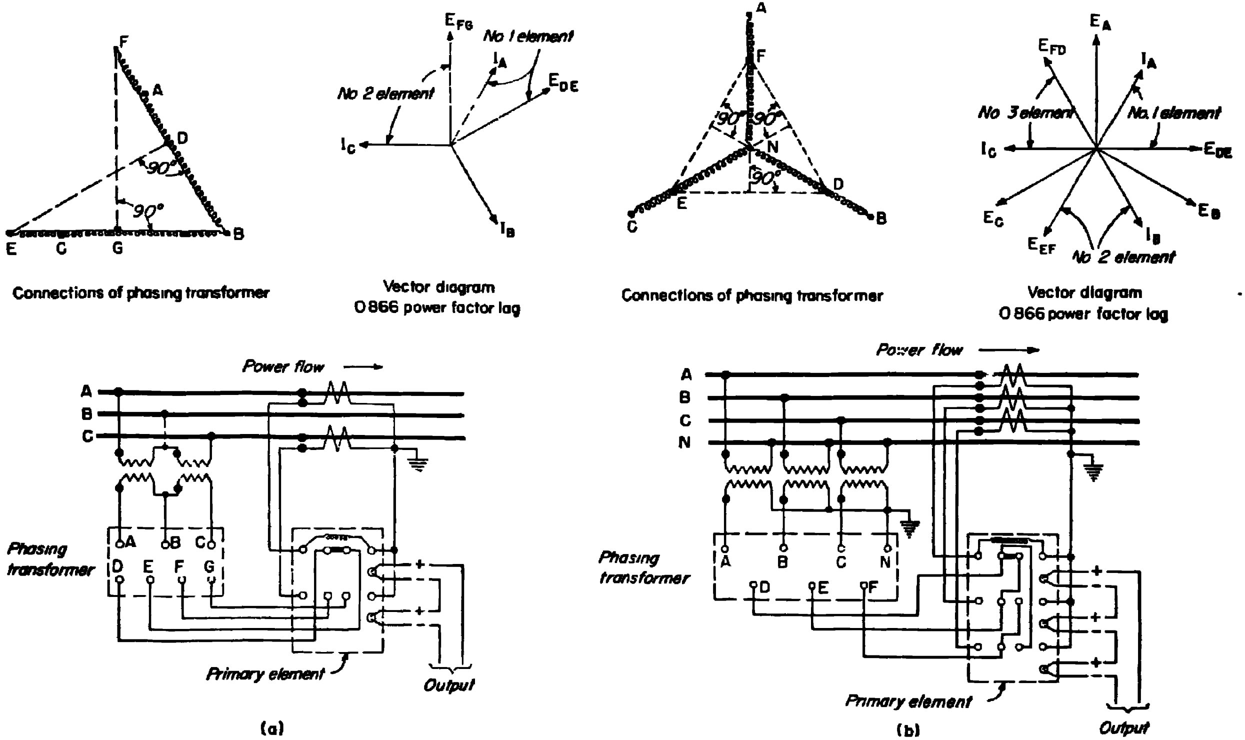
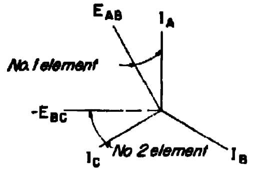


FIG. 8-52. (a) Reactive volt-ampere measurement on three-phase, three-wire circuit using phasing transformers. (b) Reactive volt-ampere measurement on three-phase, four-wire circuit using phasing transformers.

ate values of power factor the recorder balances at intermediate points along the scale.

Alternating-voltage and -current Measurements. For the measurement of alternating voltage or current any of the primary elements listed for a-c measurements under Primary Elements Used for Graphic Measurements can be used. They all produce a squared scale for both voltage and current.

The thermal type of primary element offers certain advantages in that its output can be connected directly to a full metallic telemetering channel without the use of auxiliary equipment. It also offers certain advantages when it is desired to record several voltages and or currents on a single chart which is commonly done by means of a multiple-point recorder.



Vector diagram unity power factor

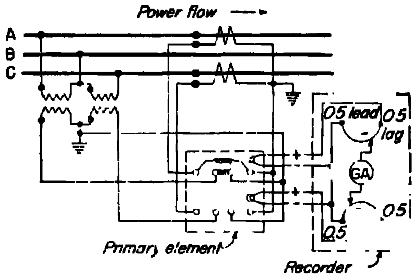


FIG. 8-53. Measurement of power factor by means of a thermal converter.

TELEMETERING PRINCIPLES

The systems most commonly used for the telemetering of electric power and related quantities may be classified as follows:

1. Direct-current Systems
   a. Current
   b. Voltage
2. Impulse Systems
   a. Time duration
   b. Frequency
   c. Impulse matching

Direct-current Systems. The system of telemetering which employs direct current as a translating means may take several different forms. The systems employing this translating means produce the necessary d-c voltage so as to cause a direct current to flow in a telemetering channel. This current is maintained at a definite ratio with the primary quantity being measured. The current corresponding to full-scale reading varies from 5 to 30 ma depending on the manufacturer.

The transmitting principle with all systems is such that changes in the circuit resistance through which the current flows and also changes in supply voltage to the power supply are automatically compensated.

The Esterline-Angus long-distance recorder, the General Electric Company torque-balance transmitter, and the Leeds and Northrup Company current-balance transmitter are typical of current-balance transmitters.

The first two enumerated above employ grid-controlled vacuum tubes to produce the required telemetering current. The torque-balance transmitter (Fig. 8-54) consists of an optical system, a lamp, two photoelectric tubes, and two three-element vacuum tubes. The restraining element, through which the telemetering current flows, sets up a torque opposing that of the operating element. If these two torques are not equal, the suspension on which the mirror is mounted will turn to a new position and decrease the light on one photoelectric tube and increase the light on the other. The photoelectric tubes by means of their light-sensitive characteristics will bias one three-element tube to conduct less current and will bias the other to conduct more. The difference in these two currents is the current which flows in the telemetering channel and also the restraining element. When the torque of the restraining element equals that of the operating element, the mirror remains stationary in the position required. The system is suitable for operating with a zero-center wattmeter. The polarity of the telemetering current will reverse with a reversal of power flow through the wattmeter.

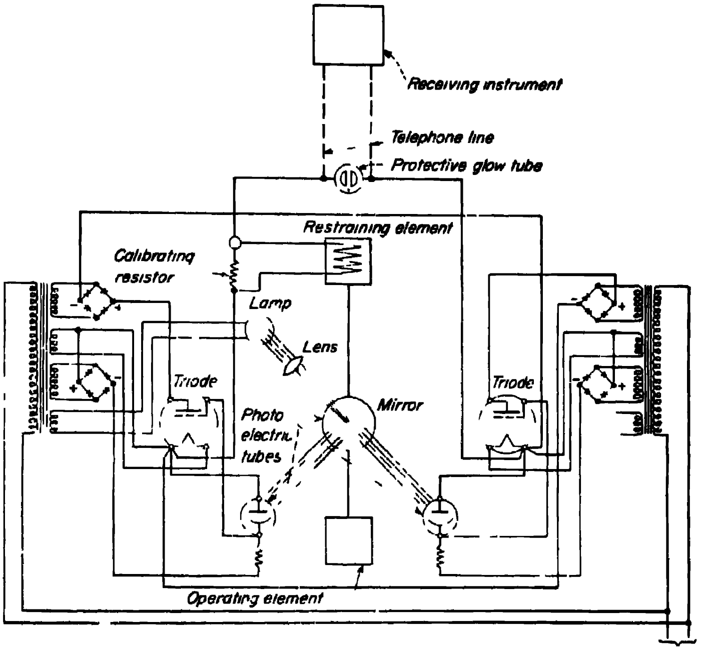


Fig. 8-54. Torque-balance transmitter.

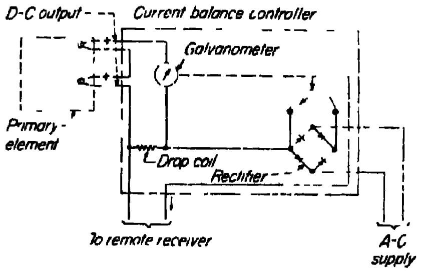


Fig. 8-55. Current-balance transmitter.

The Leeds and Northrup Company current-balance transmitter (Fig. 8-55) uses a Lincoln thermal converter as a primary element. The d-c output of the thermal converter is compared with the voltage across a "drop" coil which is connected in series with the telemetering circuit so that the d-c voltage across this coil is proportional to the current flowing in the telemetering channel.

Any difference between the thermal converter and "drop"-coil voltages will be detected by the galvanometer and by means of the mechanical (Micromax) amplifying system will vary the voltage, which sets up the telemetering current, until a balance is achieved. Thus, a definite relation is maintained between the thermal-converter output and the telemetering current.

Any number of receivers may be connected in series with the d-c system so that the same reading may be obtained at widely separated locations. The receivers may be recorders of the milliampere or potentiometer type, or an indicating milliammeter may be used when a record is not desired.

The d-c system is used with metallic circuits up to 100 miles in length and has certain advantages when the noise level of the circuit is high. It is readily apparent that the noise voltage across the receiver terminals is the product of the current produced in the loop telemetering channel by the noise voltage and the receiver resistance. The receiver resistance is usually in the order of a few ohms so that the noise voltage across the receiver is only a small fraction of the total noise voltage in the circuit. For example, if the total telemetering-channel resistance is 1,000 ohms and the receiver resistance is 10 ohms, only 1 per cent of the noise voltage appears across the receiver terminals.

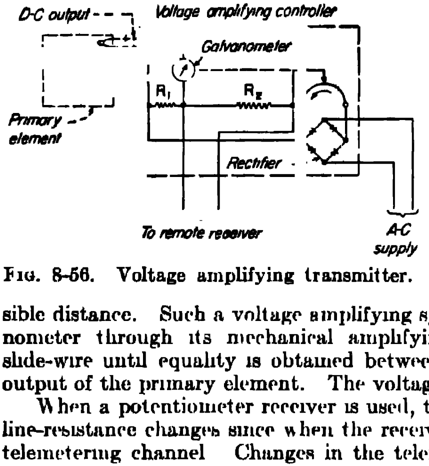


FIG. 8-56. Voltage amplifying transmitter.

Direct-current Voltage Systems. Direct-current voltage systems use a d-c voltage, usually of small magnitude, as a translating means. The d-c voltage system usually employed, where the distance does not exceed 20 to 30 miles and no unusual noise voltages are encountered, is that employing a thermal converter as the primary element and a potentiometer recorder as a receiver.

When distances are greater than noted above, a voltage amplifying system can be used to extend the permissible distance. Such a voltage amplifying system is shown in Fig. 8-56. The galvanometer through its mechanical amplifying system adjusts the contact of the slide-wire until equality is obtained between the voltage across R_1 and the voltage output of the primary element. The voltage amplification obtained is R_2/R_1.

When a potentiometer receiver is used, the voltage system is also independent of line-resistance changes since when the receiver is balanced no current flows over the telemetering channel Changes in the telemetering-channel resistance due to temperature variations affect only the sensitivity and not the inherent accuracy of the system.

Slow-rate Impulse Systems. *Time Duration.* Several telemetering systems utilizing the length of impulse in a cycle repeating at constant intervals are now in general use.

These are typified by the Builder's Iron Foundry Chronoflo system, The Leeds and Northrup Company Micromax Ballistic system, and the Bristol Company Metameter system.

The principle employed in the Chronoflo and Metameter systems utilizes the time between successive impulses as well as the length of the impulse. It is evident that, for the constant telemetering cycle, if the impulse become longer, the time between impulses becomes shorter and vice versa. The transmitter (Fig. 8-57) consists of cam, synchronous motor, rocker bar and magnet, and cam shoe and arm.

In operation, the cam rotates at a constant speed making a complete revolution in 15 sec (1-min or 4-sec cycles are used for some services). The cam-shoe arm swings in a plane parallel to the face of the cam and rides between the cam and the rocker bar. As the leading edge of the cam engages the cam shoe, the rocker bar is moved forward, carrying the magnet away from the mercury switch and causing the switch contacts to close.

Since the cam-shoe arm is positioned by the measuring element, the length of time that the cam shoe rides on the cam, and consequently the length of time that the switch is closed, will vary in accordance with the quantity measured.

The receiver (Fig. 8-57) consists of a relay, magnetic "up" and "down" clutches, synchronous-motor clutch drive, and pen-arm positioner. The motor, running continuously, drives the clutch gears in opposite directions. When the two-wire signal circuit from the transmitter is closed, the relay is energized and in turn closes the circuit which energizes the "up" clutch, and the pen-arm positioner travels outward as long as this clutch is engaged. Conversely, when the circuit from the transmitter is open, the relay is deenergized, thus energizing the "down" clutch, and the pen arm positioner moves inward. Once each cycle, therefore, the pen (or pointer) is accurately and positively positioned to the correct reading.

If the frequency of the supply to the transmitter and receiver are the same value, there is no inherent error in the telemetering system. If the frequency applied at the transmitter is higher than that of the receiver, the receiver will be proportionately insensitive. When the frequency at the receiver is higher than that at the transmitter, the receiver will oscillate or hunt around the correct reading proportional to the difference of the frequencies.

The Leeds and Northrup Company Micromax Ballistic system of telemetering operates on a 2.4-sec cycle. The transmitter consists of Micromax potentiometer-type load recorder which generates impulses proportional in length to the transmitter reading.

The receiver consists of a standard Micromax balancing mechanism operating in conjunction with a ballistic galvanometer, hence the name Micromax Ballistic system of telemetering.

The receiver measuring circuit (Fig. 8-58) takes the form of a bridge which is supplied from a rectified d-c source. The bridge form of circuit makes the measurement independent of drifts in the voltage supply.

The relay contact F_1 is closed by the transmitter and remains closed for a time equal to the length of the transmitter impulse. The closure of F_1 charges capacitor C through the high resistance R.

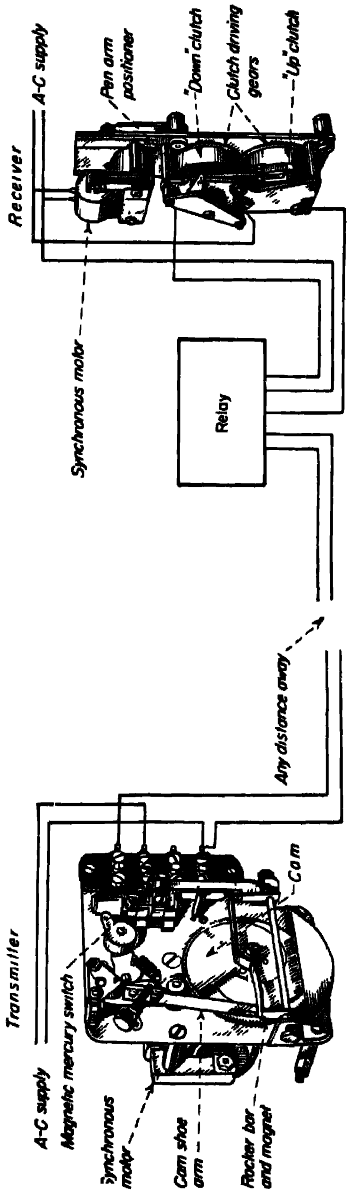


Fig. 8-57. Chronoflo telemetering system.

The voltage across the capacitor at the end of the transmitted impulse, which bears an exponential relation to the length of the impulse, is measured ballistically by the closure of interrupter I_1. If the capacitor voltage differs from that obtained from the slide-wire at the point of contact, the galvanometer receives a ballistic "kick" when interrupter I_1 closes. Immediately after the galvanometer deflects, the Micromax mechanism clamps the galvanometer pointer when it has deflected to its maximum position and moves the recorder slide-wire—pen moves with the slide-wire—to a position where the two voltages are equal. If there is no further change in the length of the transmitter impulse, subsequent closures of I_1 will not cause the galvanometer to deflect since the capacitor and the slide-wire voltage at the contact point are equal.

After the voltage of the capacitor has been measured, the capacitor is discharged by the shorting interrupter I_2 so that the capacitor starts the next charging cycle from zero voltage. The time constant of this RC circuit is approximately ten seconds so that for the maximum length of impulse—approximately two seconds - the capacitor voltage is still on the more linear part of the exponential charging curve The effect of this small departure from linearity is eliminated by shunting a portion of the slide-wire by means of resistor R_1.

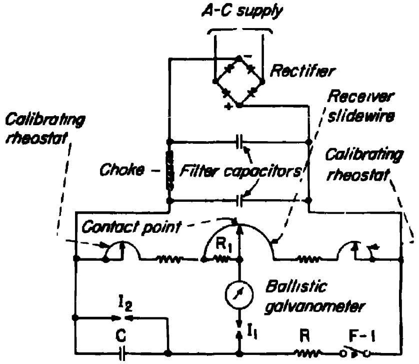


FIG. 8-58. Micromax Ballistic telemetering receiver circuit.

Means are provided for automatically synchronizing the receiver with the transmitter so that the interrupters close at the correct point in the telemetering cycle. This synchronizing means also permits the operation of this telemetering system when the frequency at the transmitter and receiver differ by as much as 2.4 cycles. Under these abnormal frequency conditions the receiver record will be displaced with respect to the transmitter but will otherwise faithfully reproduce the transmitted quantity.

Impulse Matching. A type of telemetering in which both the transmitter and receiver generate impulses proportional to their respective readings is typified by the Bailey Company Synchro-meter and the Leeds and Northrup Company type-MI telemetering.

The "Synchro-meter" (Fig. 8-59) consists of a transmitter and a receiver which are operated from power sources having the same frequency so that the cams C_1 and C_2 have the same time cycle. The contacts at the transmitter are pivoted at points C and M so that, as the quantity being measured increases, the upper contact moves upward, and the lower contact must consequently move up farther to close these contacts. Thus the time of contact is later in the cam cycle. The receiver operates in essentially the same manner except that its upper contact is positioned with the pointer of the receiver. If the transmitter and receiver have the same reading contacts, M_r and M_t will close simultaneously it is assumed for the moment that the cams are synchronized as regards phasing in the cycle—and close both shading-coil circuits of the induction motor with the result that the receiver does not move. As the receiver cam advances in its cycle, the mercury switch is opened before contacts M_r and M_t so that it is not necessary for these contacts to break any current. If the transmitter reading decreases, contacts M_t will close before contacts M_r and close the circuit of one shading coil of the induction motor, causing the receiver to decrease its reading. When contacts M_r close, they will close the circuit of the second shading coil of the induction motor, and the motor stops. The movement of the receiver is thus proportional to the difference in the readings of the transmitter and receiver. When the transmitter reading increases, contacts M_r close first, causing the receiver reading to increase. As noted in the foregoing, the cams must be correctly phased in the cycle. This is accomplished by means of an auxiliary circuit which, by interrupting the feed to the receiver cam motor, causes the receiver cycle to be retarded until it coincides in place with the transmitter cycle. The auxiliary circuit requires a third wire between the transmitter and receiver.

The type MI telemetering of the Leeds and Northrup Company operates on the same principle, but only a single circuit is required between the transmitter and receiver. The difference in the start of the impulses is used for synchronizing, and the difference at the end of the impulses is utilized for balancing the receiver.

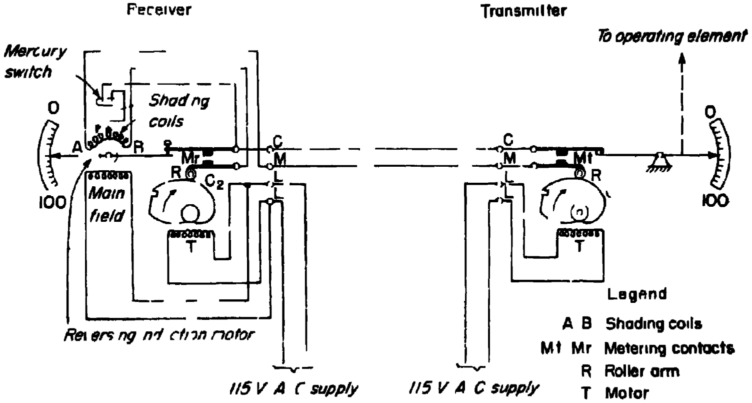


Fig. 8-[illegible]. Synchro-meter telemetering system.

These systems of telemetering are not affected by frequency variations when the frequency is the same at both the transmitter and receiver. The type MI telemetering system is arranged to produce a faithful record of the transmitted quantity—although slightly displaced—when the frequencies differ by an amount as great as 2.5 cycles.

High-rate Impulse Systems. High-rate impulse systems are generally understood to be telemetering systems which send from ten to approximately two hundred impulses per minute.

The Westinghouse system of impulse telemetering which operates over a range of 40 to 200 impulses per minute is an example of this type of telemetering.

The transmitter consists of a conventional polyphase induction disk watt-hour meter carrying on the upper end of its shaft a commutator similar to that used on a d-c watt-hour meter. Alternate bars of the commutator are connected to a slip ring. In order that the receiving equipment does not have to operate down to zero impulses and also to take care of installations where power flow may be in either direction, a synchronous motor drives a brush around the commutator, making and breaking in the circuit which sends out the impulses. When the watt-hour meter is running, the rate of impulses is proportional to the algebraic sum of the speeds of the brush and meter. When the watt-hour meter is standing still, a so-called "base rate" of impulses is generated which is proportional to the synchronous motor speed.

The receiver uses the principle of releasing a constant quantity of electricity for each impulse. These charges are smoothed out into a steady meter reading by a smoothing circuit. The receiver is constructed as two units, one containing the glow-tube constant-voltage device, relay, and capacitors and the other the indicating or recording instrument. The schematic diagram is shown in Fig. 8-60.

The impulse operates the relay which transposes the leads to the 0.2-uf capacitor on a 90-volt d-c circuit. Since the capacitor is discharged and recharged in the opposite direction, the total charge represented by a change in relay position is 36 microcoulombs. The smoothing circuit consists of a choke coil in series with the meter and a condenser and copper-oxide rectifier in series, shunting the coil and meter. The impulse of current from the operation of reversing the condenser passes through the rectifier in the forward direction and is stored in the large capacitor from which it flows into the meter through the rectifier in the backward direction. The action of the rectifier in the charging of the capacitor is to stretch out the discharging time over what can be obtained with an ohmic resistor.

The choke coil serves mainly to reduce the ballistic kick given the meter pointer when an impulse is received. The receiving meter is designed with a very strong field which permits a high cemf to be developed with small motion and further serves to reduce pointer pulsations. The glow tube UX-874 is used to keep the voltage constant for charging the capacitor.

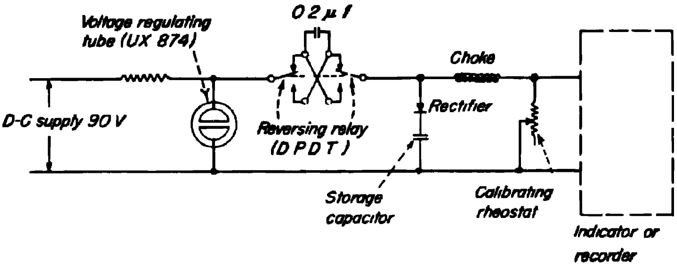


Fig. 8-60. High-rate impulse telemetering receiver.

Telemetering Systems Using Frequency as a Translating Medium. Frequency as a translating medium for telemetered data offers excellent possibilities for high-speed telemetering systems. A frequency system can be considered as an extremely high-rate impulse system. The use of an oscillator as a frequency source permits the rate of impulsing to be changed smoothly and, for the most practical considerations, instantaneously to a new value. If the primary measuring element which changes the oscillator frequency is fast, there is little delay in getting the change in frequency to the remote receiver.

With a continuously balancing type of receiver it is entirely possible to reduce the over-all response of the telemetering system to less than ½ sec.

The idealized system should have the following characteristics:

1. The frequency used should be between 20 and 100 cycles, and the band width should be in the order of 20 per cent of the base frequency used. For example, a range of 20 to 25 cycles or 80 to 100 cycles would be ideal from the standpoint of carrier-current transmission. Frequencies of this order take up very little space in the frequency spectrum, which is becoming more crowded as time goes on. Frequencies in this range also have the advantage of low attenuation when used on telephone circuits.

2. The receiver should, to a high degree, be independent of changes in the wave form which appears at its input terminals.

3. Variations of 40 db (100:1) in the amplitude of the voltage input to the receiver should not cause appreciable errors.

4. The speed of response of the components which constitute the telemetering system should be such that the over-all response does not exceed ½ sec. The use of a frequency lower than 20 cycles means that delay must be incorporated into the receiver, which shows up the response of the system.

5. The telemetering system should be independent of ambient temperature changes.

Some of the disadvantages of frequency telemetering systems employed in the past were:

1. The use of too high a frequency for translation
2. The use of too wide a band of frequencies
3. Insufficient stability as regards ambient temperature changes
4. Errors introduced by wave-form changes
5. Errors due to voltage variations

DIRECT-CURRENT TEIEMETERING CHANNELS

All telemetering systems which employ either direct current or voltage as a translating medium require a full metallic circuit. In special cases the earth may form the return conductor. This, however, is the equivalent of a metallic conductor. In all cases the insulation of the telemetering channel should be as good as the standards required in telephone practice

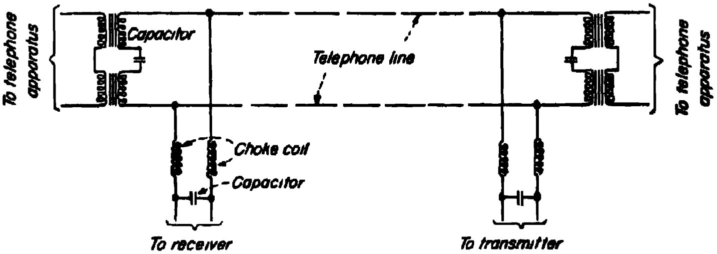


FIG. 8-61 Composite telephone circuit for telemetering.

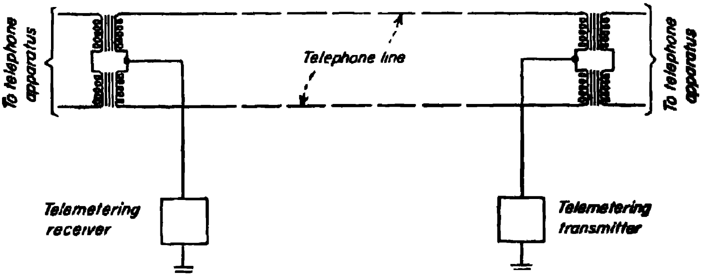


FIG. 8-62. Simplexed telephone circuit for telemetering.

The distance over which a reading may be telemetered is limited, from a practical standpoint, only by the cost of the telemetering channel. Telemetering readings for a distance of 100 miles is common practice.

Direct-current Voltage Systems. For all d-c *voltage* systems of telemetering which operate either on a few millivolts or several volts, a full metallic, ungrounded, circuit is essential. The circuit should also be balanced within commonly accepted limits.

Under special conditions it may be possible to use a composite circuit. The circuit as shown in Fig. 8-61 is typical.

Direct-current Systems. For direct-*current* systems the circuits described under Direct-current Voltage Systems may be used, and in addition, under certain conditions, a single wire with ground return or a simplexed telephone circuit as shown in Fig. 8-62 may be used.

In connection with d-c telemetering circuits it should be understood that the *most* *reliable and interference-free readings* are obtained when a twisted pair in lead cable, which is not simplexed or composited, is used as the telemetering channel.

IMPULSE-TELEMETERING CHANNELS

Impulse telemetering may be used with any circuit which will pass an alternating current. The circuit may be of the metallic type completed through a magnetic circuit or a dielectric circuit. This means that, in addition to the circuits which are used for d-c telemetering, lower grade metallic circuits as well as carrier-current circuits may be used for transmitting the telemetering impulses.

With carrier-current channels, which in most cases use existing high-tension lines for transmission, the cost of the transmission circuit is not a factor. Consequently, the transmission of telemetering signals a distance of 300 or 400 miles is not uncommon.

FACTORS AFFECTING CHOICE OF TELEMETERING SYSTEMS

The factors of reliability and over-all cost (channel plus telemetering equipment) determine largely whether direct current or impulse telemetering is used.

Reliability. Direct-current systems of telemetering are inherently more reliable than impulse-telemetering systems. In its simplest form a d-c telemetering system may have as its transmitter a thermal wattmeter whose d-c millivoltage output is connected directly to the telemetering channel Thus there are no moving parts and no auxiliary supplies required at the transmitter. The receiver consists of a standard potentiometer recorder.

An impulse system of telemetering, in its simplest form, consists of a primary measuring element which actuates a contacting device with the consequent wear on the contacting device and relays. Also one or more auxiliary power supplies is required for the operation of the system. In general, the receiving recorder for impulse systems is more complicated than that used with the d-c systems.

Since the d-c system has fewer moving parts than the impulse system, the order of reliability is higher.

Telemetering Equipment Costs. The cost of equipment required for d-c telemetering systems is less than that for a comparable impulse system so that, if the telemetering equipment were the primary cost, the d-c system would be used exclusively This cost, however, is usually not a very great item. The cost of the telemetering channel usually dictates the type of telemetering system used

Telemetering Channels. In metropolitan areas where the distances possibly do not exceed 20 to 30 miles, it is impractical to use anything but telephone lines to bring the telemetering signals to the system operator's office since this is usually located centrally in the area. In general, the generating stations and interchange points radiate in all directions from the system operator's office so that a separate telemetering channel is required for each point. Consequently, the d-c system of telemetering is usually used because of lower first cost and higher reliability. If it were desired to transmit a multiplicity of readings between a generating station and the system operator's office, the use of carrier frequency on the telephone cables, in conjunction with impulse telemetering, might be economically justified depending on the basis used for computing the channel rental.

When the telemetering is not in a congested area, the problem generall, resolves into equating the cost of the carrier frequency and the impulse-telemetering equipment to be used over existing high-tension circuits to that of the rental of a telephone channel and the cost of the d-c telemetering equipment. The factor of reliability should be given proper weighting in this comparison.

AUTOMATIC INTEGRATION OF GRAPHIC METER RECORDS

In many instances an integration of the quantity being recorded at the receiving end of a telemetering system is required. Also in some installations the integration of the total of several of the telemetered quantities as recorded on a totalizing recorder is necessary. Several methods, both mechanical and electrical, have been extensively used for this purpose.

An electrical method of integration commonly employed consists of a d-c retransmitting slide wire mounted on the recorder (Fig. 8-63) which operates in conjunction with an equivalent watt controller to produce an a-c wattage which is proportional to the recorder reading. As the d-c millivoltage of the retransmitter changes, it causes the galvanometer of the equivalent watt controller to deflect, and by means of the mechanical amplifying system of the controller the alternating current to the current element of the thermal wattmeter is varied until its d-c output balances that set up by the retransmitting slide wire. A conventional single-phase induction-disk watt-hour meter has its current coils in series and potential coils in parallel with the thermal wattmeter. Thus the watts imposed on the watt-hour meter are the same as those on the thermal wattmeter, and in turn the watt-hour meter load is proportional to the position of the retransmitting slide wire. The watt-hour-meter registration is thus an integration of the recorder chart or, to be more exact, is an integration of the electrical load being telemetered. The watt-hour meter, when equipped with contacts, operates a demand meter in the conventional manner.

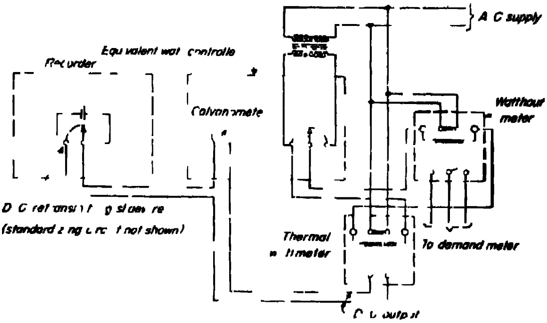

A C supply
Equivalent watt controller
Recorder
Galvanometer
Watthour meter
D C retransmitting slide wire
(standardizing circuit not shown)
Thermal wattmeter
To demand meter
D C output

Fig. 8-63. Integrating circuit for telemetered reading.

A typical mechanical type of integrator (Fig. 8-64) has a pivotally mounted member 1 which is rotated at constant speed around 6 as a center. Member 1, when in its lowered position as shown in Fig. 8-64, permits knife edge 11 to engage the toothed gear 8 and causes it to rotate with 1 in a clockwise direction until the projection 12 touches a fixed pin (not shown) and lifts assembly 1. When the assembly is lifted, the flipper 10, owing to its spring action, assumes a position whereby assembly 1 and the knife edge 11 are lifted and the toothed gear 8 is disengaged, and there is no further rotation of 8. It should be noted that the flipper 10 rotates with assembly 1. As the flipper rotates further it comes in contact with notch 14 on the bakelite disk 7. When the flipper drops into notch 14, it assumes the position shown, and the toothed gear 8 is engaged. The position of notch 14 is governed by the reading of the recorder. The movements of toothed gear 8 are summed up on a dial (not shown),

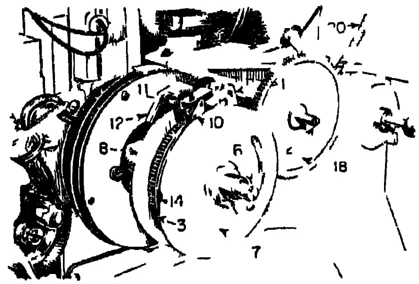


Fig. 8-64. Mechanical integrator for telemetered reading.

and thus the dial reading is an integration of the quantity being measured. Contacts for the operation of demand meters are mounted on an extension from the dial shaft.

TELEMETERING APPLIED TO AN INTERCONNECTED SYSTEM

Local and remote graphic power metering for a typical interconnected electric-power system may incorporate several types of primary elements and methods of transmission (Fig. 8-65).

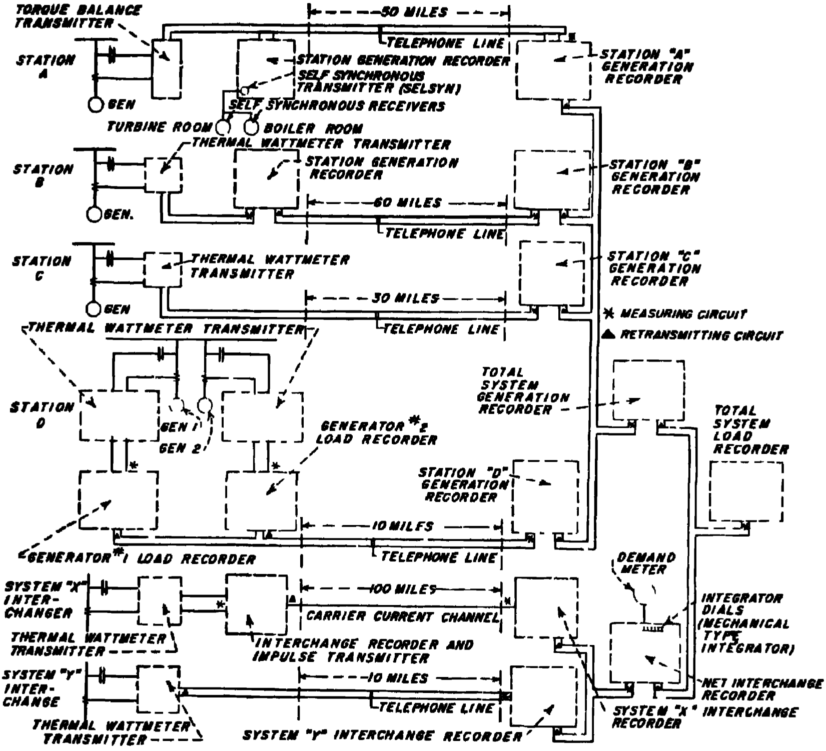

TORQUE BALANCE TRANSMITTER
50 MILES
STATION A
TELEPHONE LINE
STATION GENERATION RECORDER
SELF SYNCHRONOUS TRANSMITTER (SELSYN)
STATION "A" GENERATION RECORDER
GEN
SELF SYNCHRONOUS RECEIVERS
TURBINE ROOM
BOILER ROOM
THERMAL WATTMETER TRANSMITTER
STATION B
STATION GENERATION RECORDER
STATION "B" GENERATION RECORDER
GEN.
60 MILES
TELEPHONE LINE
STATION C
THERMAL WATTMETER TRANSMITTER
STATION "C" GENERATION RECORDER
GEN
30 MILES
TELEPHONE LINE
* MEASURING CIRCUIT
▲ RETRANSMITTING CIRCUIT
THERMAL WATTMETER TRANSMITTER
TOTAL SYSTEM GENERATION RECORDER
STATION D
GEN 1
GEN 2
GENERATOR #2 LOAD RECORDER
TOTAL SYSTEM LOAD RECORDER
STATION "D" GENERATION RECORDER
10 MILFS
GENERATOR #1 LOAD RECORDER
TELEPHONE LINE
DEMAND METER
SYSTEM "X" INTER-CHANGER
100 MILES
CARRIER CURRENT CHANNEL
INTEGRATOR DIALS (MECHANICAL TYPE INTEGRATOR)
THERMAL WATTMETER TRANSMITTER
INTERCHANGE RECORDER AND IMPULSE TRANSMITTER
SYSTEM "Y" INTER-CHANGE
10 MILES
TELEPHONE LINE
NET INTERCHANGE RECORDER
SYSTEM "X" INTERCHANGE RECORDER
THERMAL WATTMETER TRANSMITTER
SYSTEM "Y" INTERCHANGE RECORDER

FIG. 8-65. Typical installation of telemetering on an electric power system. System operator's office not designated.

The equipment at generating station A consists of a torque-balance transmitter whose d-c output current operates a local "station generation recorder," and in series with this is the 50-mile telephone line and a "station A generation recorder" located at the system operator's office. The station generation recorder at station A operates a self-synchronous transmitter (Selsyn) which in turn causes the large dial indicators driven by the self-synchronous receivers to assume the same positions as the transmitter. The use of these indicators gives the boiler and turbine rooms an instantaneous indication of any change in the station load-recorder reading.

The telemetering equipment for generating station B performs the same functions as that of station A. The difference is in the method of making the measurements. The thermal wattmeter develops a d-c potential which is measured by the local recorder. This local recorder retransmits a d-c potential—of the order of 1 volt—to the station B generation recorder located at the system operator's office.

The telemetering equipment for station C does not provide any record at the generating station. In this instance the output of the thermal wattmeter is connected directly to the telephone line and is measured by the station C generation recorder at the system operator's office.

Station D is shown to consist of two generators with the output of the thermal wattmeter of each generator recorded on a separate instrument, and thus a record of each generator output is provided. The d-c outputs of the retransmitting circuits of these two recorders are connected in series so that the station D generation recorder at the system operator's office records the sum of the two generator loads.

The interchange with system X is transmitted to the system operator's office by impulse telemetering over a carrier-current channel. The output of the thermal wattmeter is recorded at the point of interchange, and the impulses generated by this recorder are received over the carrier-current channel for the operation of "X interchange recorder" located at the system operator's office.

The method of the measurement of the interchange with system Y is the same as described for station C.

At the system operator's office a record of the output of each generating station is provided as well as a separate record of the interchange of systems X and Y. Thus the dispatcher knows at all times the conditions prevailing at the several generating stations and interchange points.

The summation of the outputs of the several generating stations by means of retransmitters on the individual recorders provides a record of total system generation.

The algebraic summation of the two interchange recorders furnishes a record of the net interchange with adjacent power systems. The net-interchange recorder is provided with a mechanical-type integrator with demand contacts. From this information the system operator can determine not only instantaneous values but integrated values as well. He also has an indication, by means of the demand meter, as to how well he is holding the demand to contract values.

The algebraic sum of the total system generation and net-interchange recorders supplies the system operator with a record of the total load on his system.

CONCLUSIONS

The ramifications of local and remote power measurements are many and varied as pointed out in this discussion. It will suffice to state that the many methods available permit the metering and combining of the quantities to solve almost any problem which exists in the field of local and remote graphic power metering

Reference

1 KNOWLTON, A. E., "Electric Power Metering," McGraw-Hill Book Company, Inc., 1934.

STEAM-TURBINE GENERATORS*

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INTRODUCTION

Of all the electrical generating capacity which exists in the United States (over 70 million kva) approximately 65 per cent consists of steam turbine generators of which Figs. 8-66 and 8-67 are representative. Of the remainder approximately 33 per cent is represented by hydraulic power using water-wheel turbines, and but 2 per cent is available from the use of diesels, gas engines and reciprocating steam engines. As a consequence, power-plant engineers are almost certain to come into contact with steam turbine generators and a knowledge of the basic principles for the application, operation and maintenance of these units will simplify the engineer's task and assist him in maintaining efficient and continuous operation of his plant. This subsection is devoted to a presentation of these fundamentals primarily as they

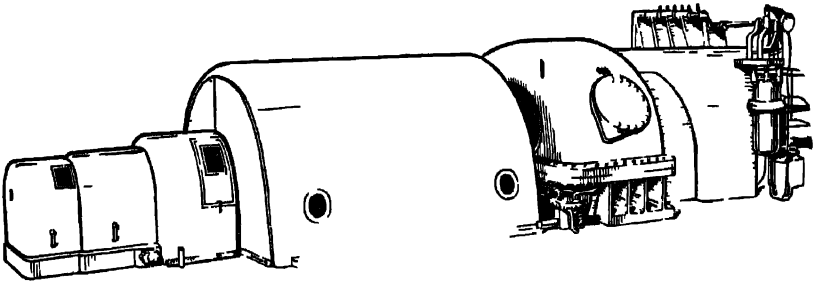

Fig. 8-66. A 7,500-kw steam turbine generator.

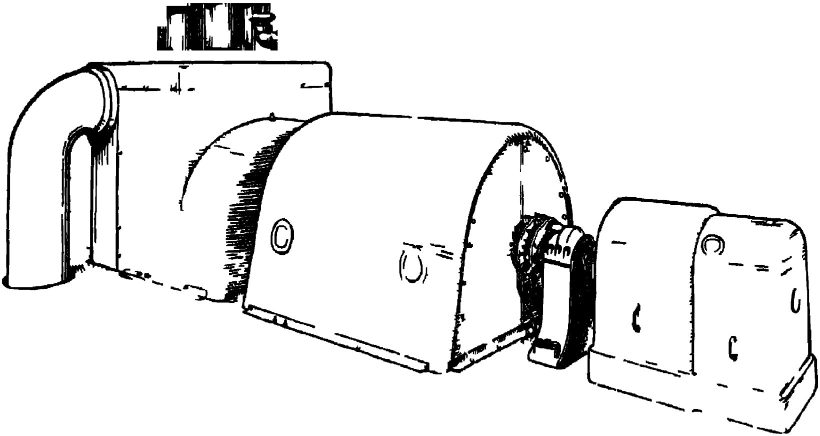

Fig. 8-67. A 1,000-kw noncondensing automatic extraction turbine generator.

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* Superior numbers refer to specific references listed at the end of this subsection.

apply to units ranging in capacity from 75 to 15,000 kw, but they apply in general to steam-turbine generators of any capacity

APPLICATION OF STEAM-TURBINE GENERATORS

Turbines

Types of Turbines. Steam turbines may be broadly classified into two types:

1 Condensing (exhaust pressure less than atmospheric)

2 Noncondensing (exhaust pressure atmospheric or higher)

Condensing turbines, illustrated by Fig 8-68*a*, convert a higher percentage of the available energy in the steam into useful work and thus for the same initial steam conditions give greater power output than a noncondensing turbine per pound of fuel consumed Condensing turbines, however, usually furnish no steam for heating or other industrial use. The heat remaining in the exhaust steam is ordinarily discarded to the condenser cooling water and serves no useful purpose

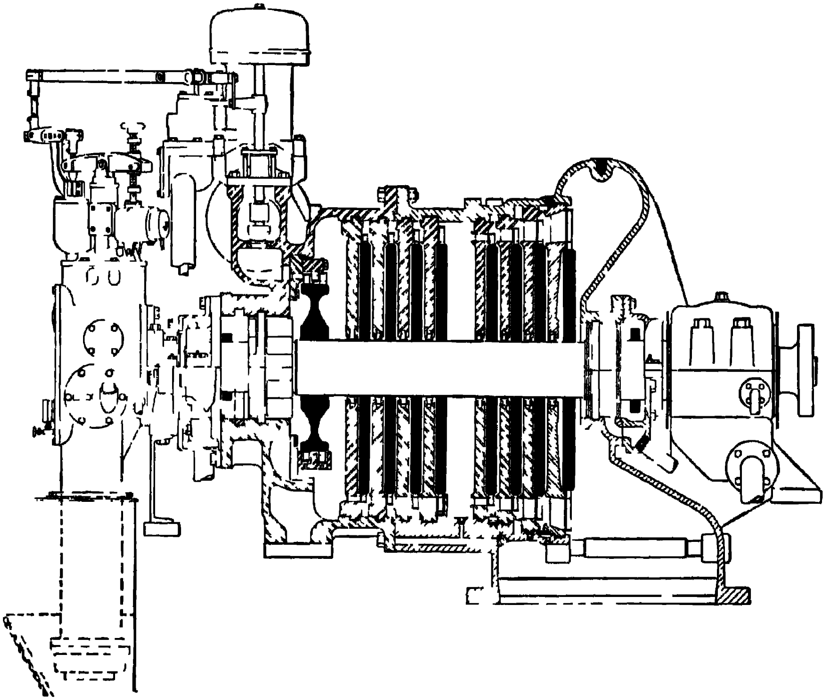

FIG 8-68*a* Straight condensing turbine.

Noncondensing turbines, represented by Fig 8-68*b*, although making less use of the available energy in the steam for electrical power generation, are generally designed so that the exhaust steam is at a pressure and temperature suitable for heating or other industrial use. Thus, the heat remaining in the exhaust steam serves a useful purpose and is not charged to the turbine as a loss In such a system, the electrical power generated may be considered a by-product of the industrial process for which steam is supplied

Condensing and noncondensing turbines may be subdivided further into extraction turbines of either the nonautomatic or automatic type Turbines of the nonautomatic-extraction type similar to Fig 8-68*c* provide steam at pressures varying with

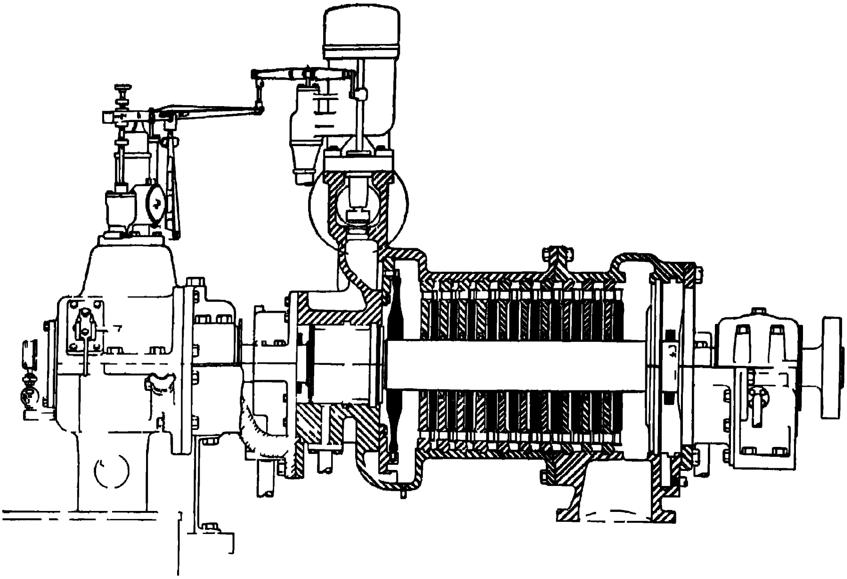

FIG. 8-68b Straight noncondensing turbine

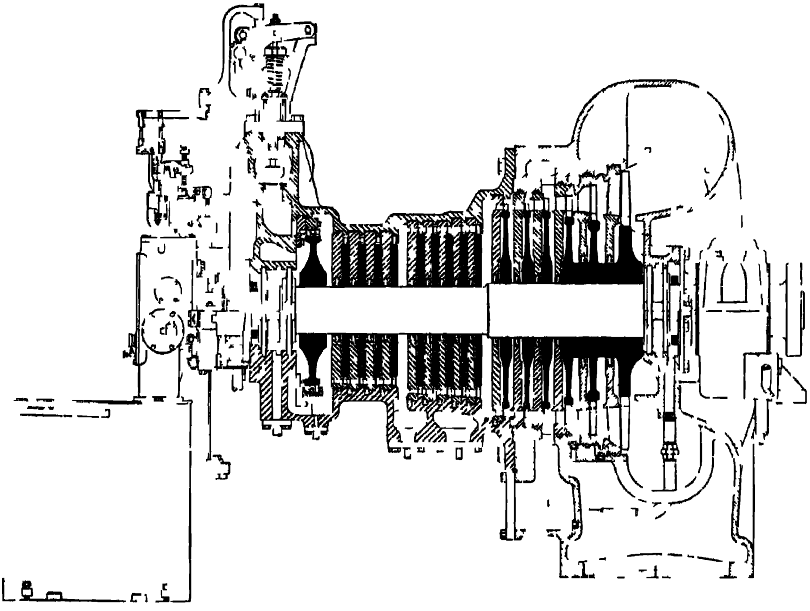

FIG. 8-68c Nonautomatic-extraction condensing turbine.

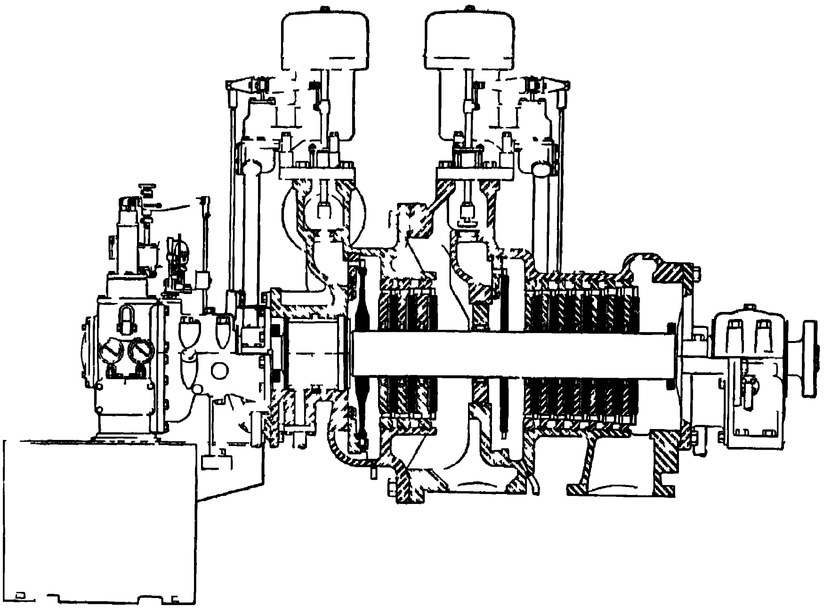

Fig. 8-68d. Single automatic-extraction noncondensing turbine.

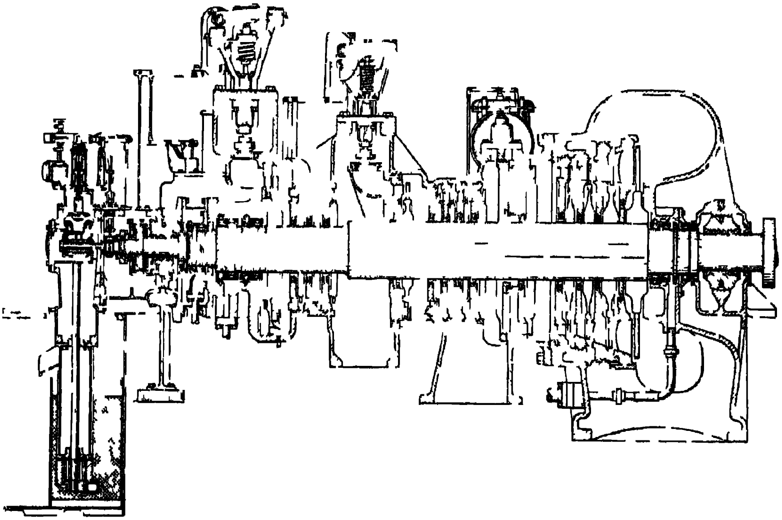

Fig. 8-68e. Double-automatic-extraction condensing turbine.

load, usually for feed-water heating as described later, but also for other industrial purposes where the quantity of steam required is comparatively small. Automatic-extraction units with either one or two extraction openings, as shown in Figs. 8-68d and 8-68e, usually furnish process steam at a constant pressure over the rated load range of the set. If the unit is noncondensing, the exhaust flow may also be used for process steam. If, however, the extraction-process steam flow is not sufficient to carry the electrical load requirements of the plant, the exhaust end of the turbine may be made condensing, and the additional steam required to carry the electrical load is passed to the condenser.

Standard Ratings. Standard ratings, in kilowatts, for both a-c and d-c turbine-generator sets are listed in Table 8-34. Units of larger rating are also available.

Table 8-34. Standard Kilowatt Ratings for Turbine-generator Sets

A-c sets			D-c sets		
10	125	1,500	10	125	1,500
15	150	2,000	15	150	2,000
20	175	2,500	20	200	2,500
25	200	3,000	25	250	*
35	250	3,500	30	300	
50	300	4,000	40	400	
60	350	5,000	50	500	
75	400	6,000	60	750	
100	500	7,500	75	1,000	
	700	10,000	100		
	1,000	11,500			
		12,500			
		15,000			

* Larger ratings can be obtained by coupling generators in tandem.

Pressure and Temperature Ranges. Turbine prices are generally increased in steps as the initial pressure and temperature are increased. For turbines rated 10 to 400 kw, the standard pressure and temperature steps are usually as shown in Table 8-35.

Table 8-35. 10- to 400-kw Turbines

Standard inlet pressure, psig	*Standard inlet temperature, deg F*
250 and lower	500 and lower
251 400	501–600
401 600	601 750
	751–825

Table 8-36. 500- to 7,500-kw Turbines

Standard inlet pressure, psig	*Standard inlet temperature, deg F*
101 250	500 and lower
251 400	501 750
401–600	751-825
601 900	826 900
901 1 500	901-950

For turbines rated at 500 to 7,500 kw, the standard pressure and temperature steps are usually as shown in Table 8-36.

For turbines rated at 10,000 kw and larger, the pressure and temperature steps (condensing and noncondensing) are usually as shown in Table 8-37.

In general, pressure and temperature are considered independently for pricing purposes, and the price of the turbine is based on the upper limit of the inlet pressure and temperature group in which it falls, *i.e.*, a turbine designed and constructed for normal operation and most efficient performance with a given inlet pressure and temperature will also be designed and constructed so as to be satisfactory for operation with minor changes at the maximum pressure and temperature of the group above in which it falls and will be priced accordingly. Thus, a 4,000-kw turbine to be purchased for operation with 375 psig 800°F total temperature steam at the throttle would be priced the same as a turbine purchased for operation with 400 psig 825°F total temperature steam at the throttle and would be constructed so as to be satisfactory for continuous operation at these upper limits with but minor changes to the steam-path parts such as new valves, valve seats, and first-stage nozzle plate.

Table 8-37. 10,000-kw and Larger Turbines

Standard inlet pressure, psig	*Standard inlet temperature, deg F*
450 and lower	750 and lower
451 650	751 825
651-850	826 900
851 1,450	901 950
	951-1,000

Steam Performance. Turbine-generator-set performance is generally quoted in terms of steam rate. The steam rate is the pounds of steam that must be supplied to the turbine inlet per kilowatthour of generator electrical output. The throttle flow, in pounds per hour, equals the product of the electrical output being delivered by the generator and the steam rate at that load.

The approximate full-load and half-load steam rates of condensing and noncondensing, nonautomatic-extraction, 60-cycle, a-c turbine-generator sets may be obtained by multiplying the theoretical steam rate[1] by factors obtained from Figs. 8-69 and 8-70.

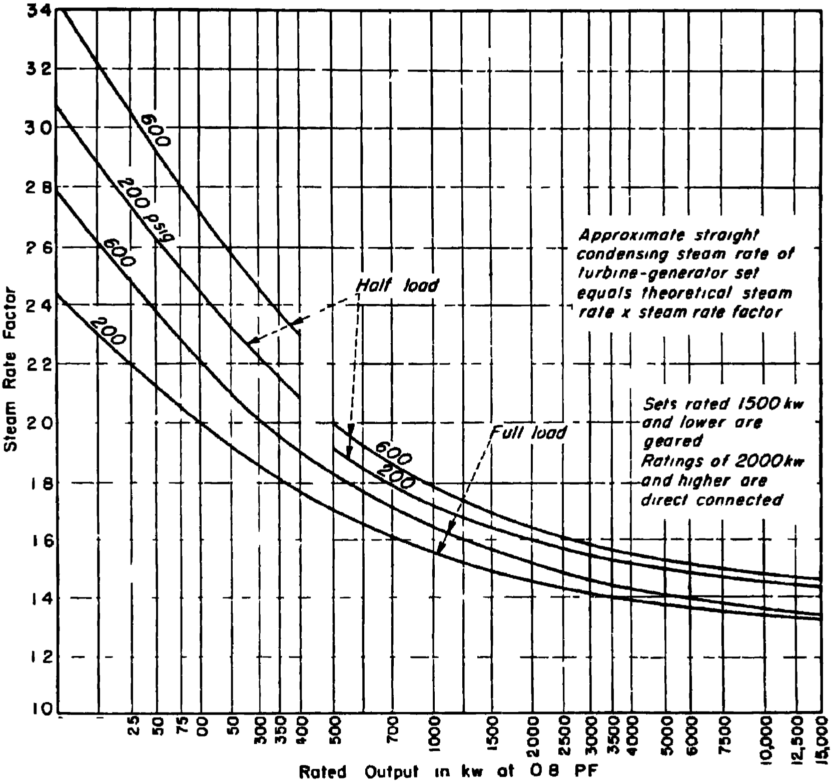

Fig. 8-69. Approximate steam-rate factors for condensing, nonautomatic-extraction, 60 cycle, a-c turbine-generator sets.

The steam rates of 50-cycle and 25-cycle a-c sets and of d-c sets will be approximately 3 to 10 per cent greater than those for the correspondingly rated 60-cycle a-c sets

Feed-water Heating. When required, turbines, either condensing or noncondensing, will be furnished with openings for the nonautomatic extraction of steam for feed-water heating or for any required industrial use such as the operation of an evaporator.

The number of openings which will be furnished without a price increase is as shown in Table 8-38.

Table 8-38

Rating of set, kw	*Number of openings*
10–400	None
500–1 500	1
2,000–3,500	2
4,000–7,500	3
10,000–15 000	4

The flow which may be extracted is limited to 15 per cent of the nonextraction throttle flow at any given load from one opening or a total of 30 per cent of the nonextraction throttle flow at any given load from two or more openings, except for machines rated 10,000 kw and larger where the uncontrolled extraction flow may equal 35 per cent of the throttle flow at any given load. A price addition will be made for quantities of extraction steam in excess of these amounts or for more openings than are shown by Table 8-38.

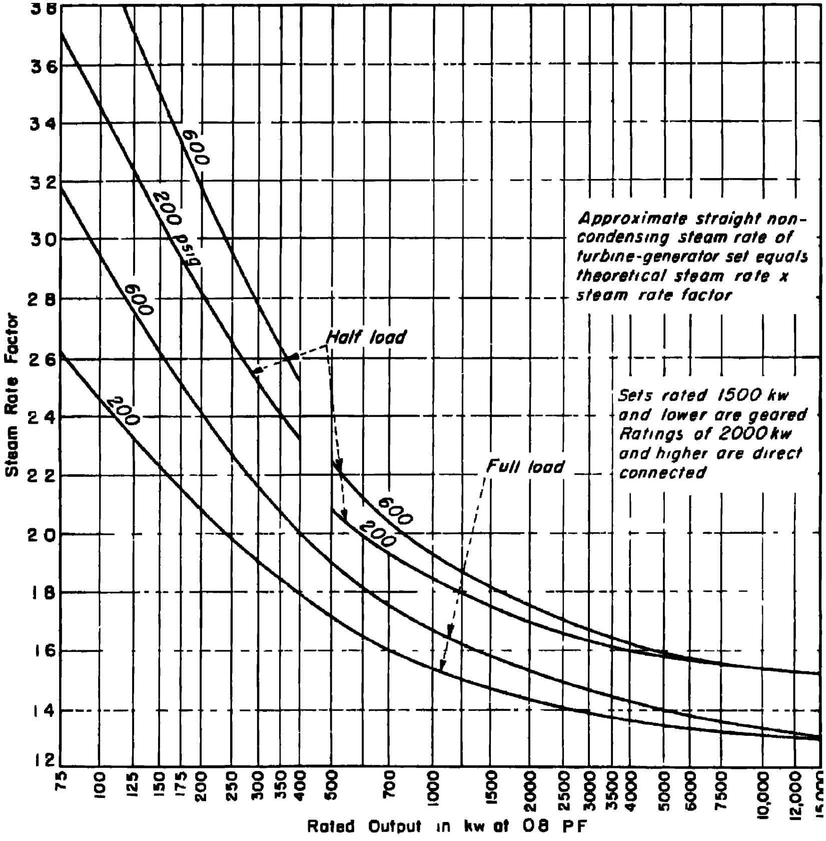

FIG. 8-70. Approximate steam-rate factors for noncondensing, nonautomatic-extraction, 60-cycle, a-c turbine-generator sets.

When steam from power-plant auxiliaries is not available for feed-water heating, the approximate fuel saving which may be made by extracting from the main turbine will be approximately as given by Fig 8-71.

Figure 8-72 gives a means of finding the approximate optimum turbine-flange pressures for feed-water heating extraction with any turbine inlet pressure and any number of heaters.

To use this chart, (1) enter with the turbine inlet pressure as the horizontal coordinate and go vertically to the line labeled with the number of feed-water heaters desired. (2) Draw a straight line from the point obtained in (1) to the ¾ load condenser pressure in inches of mercury absolute. (3) Where this line crosses the heaters designated on the coordinate, read the respective approximate ideal turbine-flange pressures for extraction to the feed-water heaters.

The turbine manufacturer may be allowed to deviate several per cent (say 10 to

20 per cent or more) from the ideal extraction pressures with little effect on the fuel savings resulting from feed-water heating extraction.

Turbine Governing. The control arrangement with which a turbine generator is equipped is generally dictated by the requirements of the application, i.e., by the service and results intended. Fundamental safety devices such as an overspeed trip (Fig. 8-73) are common to all types.

The control of speed and the control of steam pressure are the two most common functions of turbine controls.

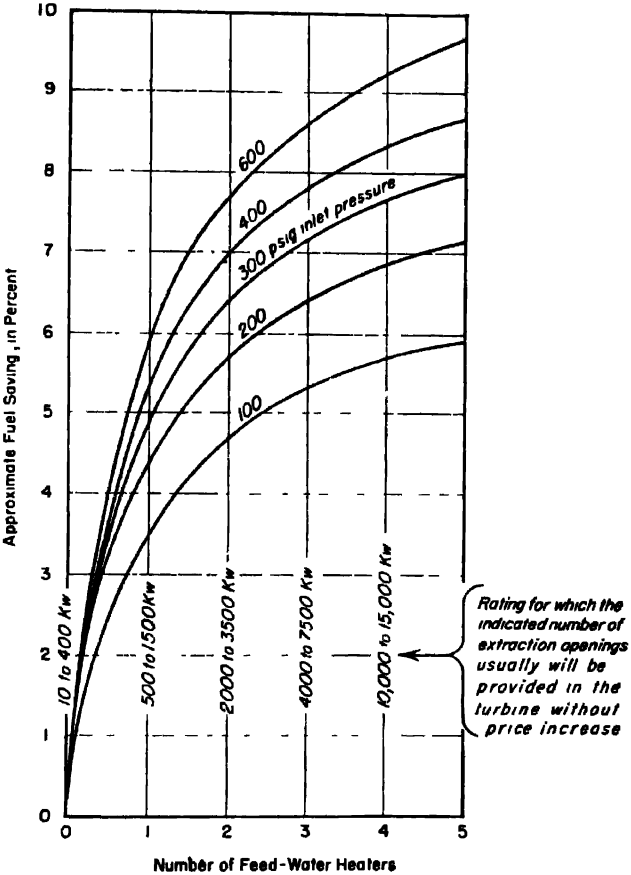

Fig. 8-71. Approximate fuel savings resulting from the use of feed-water heaters.

On very small turbines, particularly if exceptionally accurate control is not required, the speed or the pressure-sensitive element itself may be used to operate the steam-control valves as illustrated in Fig 8-74. On larger turbines and where more accurate control is required, the relatively small forces available in the speed or pressure-sensitive elements must be amplified to provide positive operation of the steam control valves. Amplification of forces is generally accomplished by the use of relays comprised of a pilot valve which regulates fluid-pressure supply and discharge on a power piston of ample capacity to operate the control valves. Figure 8-75 is representative of such a control arrangement. The governor element supplies only

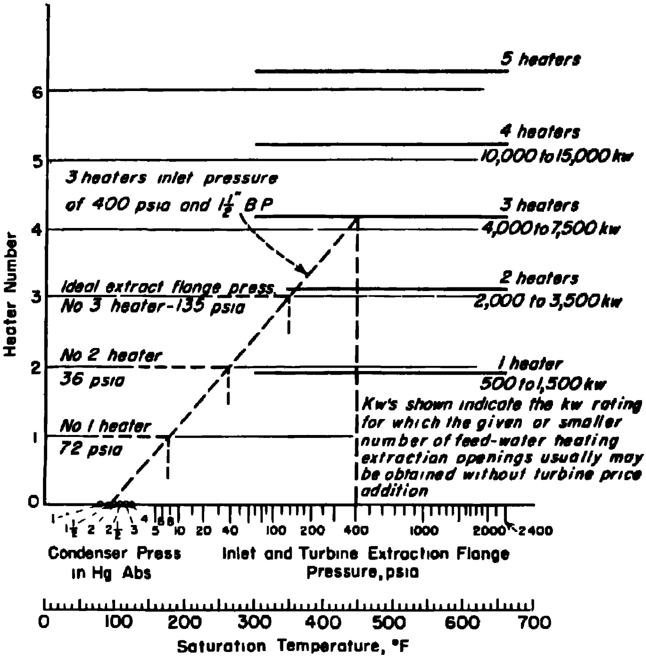

FIG. 8-72 Chart for obtaining approximate ideal-extraction turbine-flange pressures for feed-water heating extraction.

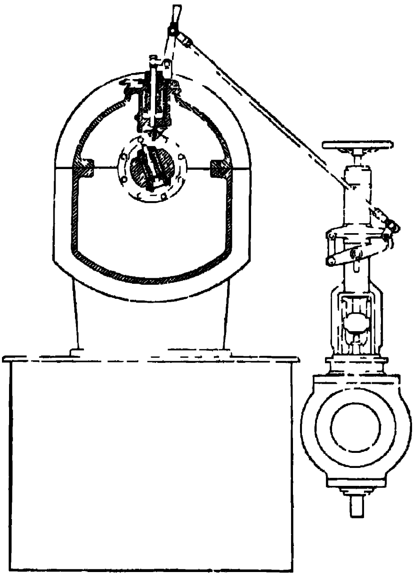

FIG. 8-73. Emergency-governor and throttle-trip arrangement.

the small forces required to position the pilot valve, and the large forces are obtained from the fluid-pressure supply. Oil at the proper pressure from the turbine's own oil system is most generally used as the operating medium. Other operating mediums, such as steam, are used successfully.

Governor Regulation. The speed or pressure (whichever is being controlled) variation permitted by the governor, as load is added or subtracted, is known as the "governor regulation."

Speed-governor Regulation. Speed-governor regulation is generally not over 2 per cent speed change for one-half rated load change or not over 4 per cent for rated load change.

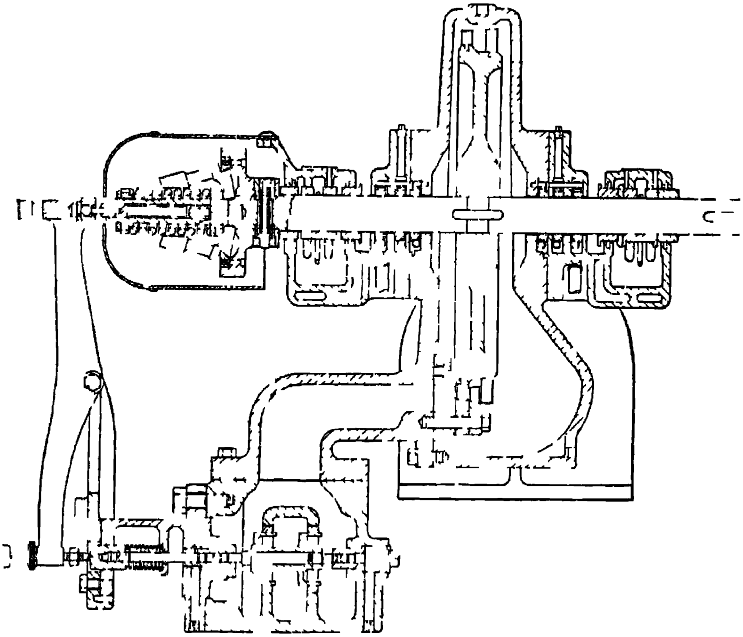

FIG. 8-74. Direct-acting speed-governor control of valve gear.

Speed governors are occasionally equipped with correctors or compensators (Fig. 8-76) which, as the name implies, correct for the speed change inherent in the regulation and give in actual practice substantially zero regulation. It is possible to operate only one such unit on a system at a time with its corrector in service. Such a unit absorbs all system-settled load changes except as load may be redistributed among units by the operators, and all other speed-controlled units operate at substantially base load.

Pressure-governor Regulation. Pressure-governor regulation is generally of the order of not over 4 per cent pressure change for a total change in the controlled steam flow. These governors (Fig. 8-77) also are frequently equipped with correctors or compensators. When so equipped, stability is enhanced by a governor with a broad basic regulation and with sufficiently fast corrector action to produce over-all control within acceptable limits.

If the entire steam flow to a turbine is under the control of a pressure governor, the generated load is determined only by steam demands, and the speed must be controlled by other a-c generating units electrically in parallel. The speed governor would generally be set as a preemergency governor and would interfere with the steam flow only on an abnormal rise in system speed.

Pressure governors are used to control exhaust pressure, extraction pressure, or inlet pressure, as the case may be Exhaust-pressure governors are generally used to balance turbine steam flow against process-heating steam demand, thus passing through the turbine only that steam required for process and thereby generating whatever power may result Likewise, initial pressure governors may be used to pass through a turbine whatever steam may be generated in a waste heat process Initial pressure governors are also frequently used to unload turbines automatically and thereby reduce boiler flow on high pressure boilers upon dropping of the boiler pressure for any reason

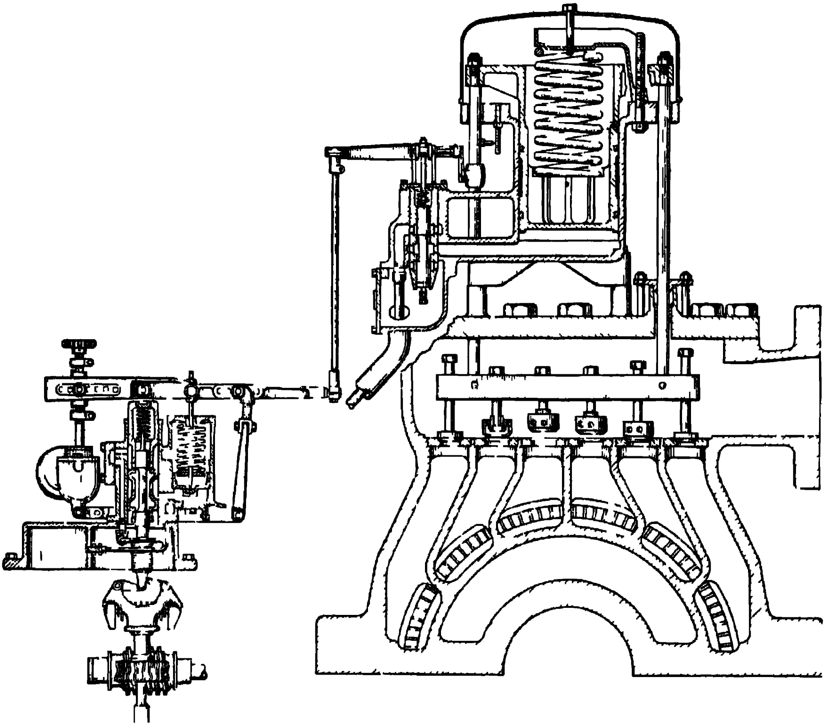

Fig. 8-75 Oil relayed speed-governor control of valve gear

Combined Speed and Pressure Governing On automatic-extraction type turbines many combinations of the functions of speed and pressure control occur depending on the service intended

When speed and extraction pressure are controlled simultaneously, the speed and pressure controls are frequently interconnected in such a way that changes caused by one do not require that major corrections be made by the other Thus, Fig 8-78 diagrammatically indicates how a change in the position of the speed governor will automatically reposition the three control valves in the same direction (towards the wide open or the closed position depending upon whether electrical load is being added or subtracted) and thus prevent a major disturbance in the pressure at the automatic-extraction openings due to the change in total steam flow through the turbine In this way, variations within the governor regulation are reduced to a minimum

Likewise, a decrease in the extraction-flow requirements at either of the extraction

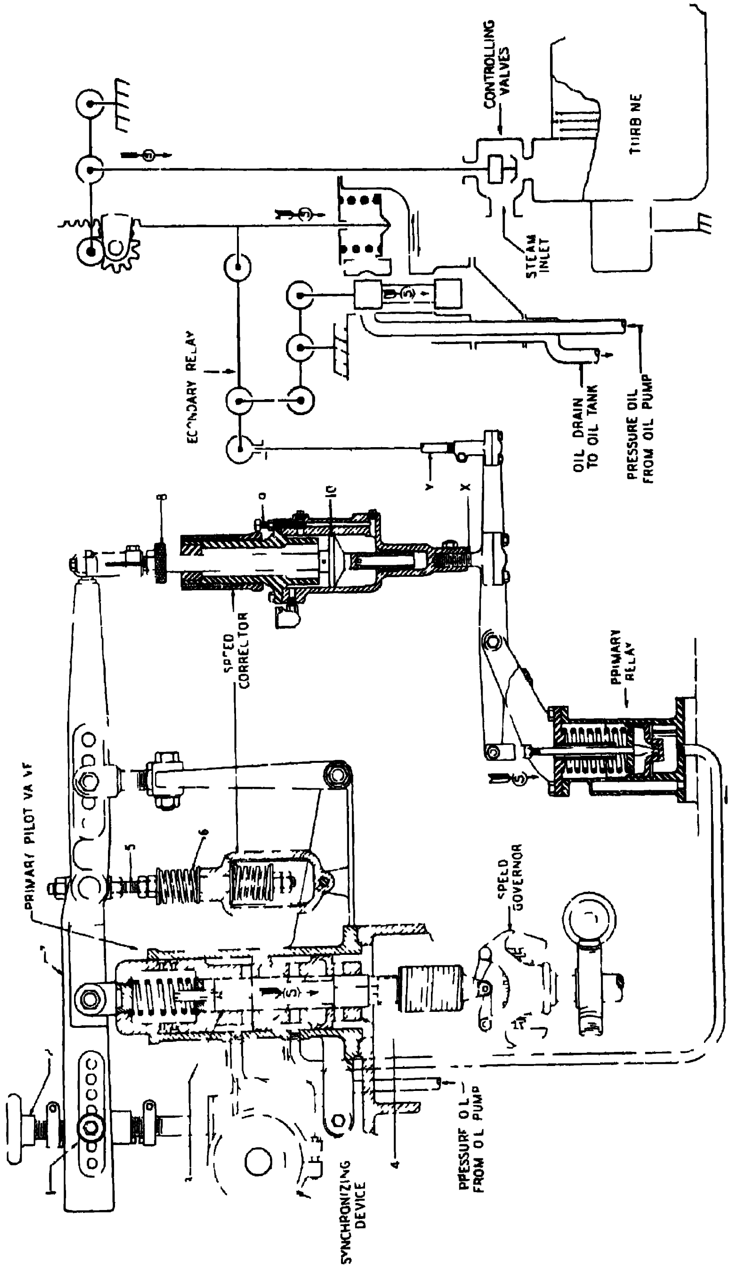

FIG. 8-76. Application of speed corrector to an oil-relayed speed-governing system.

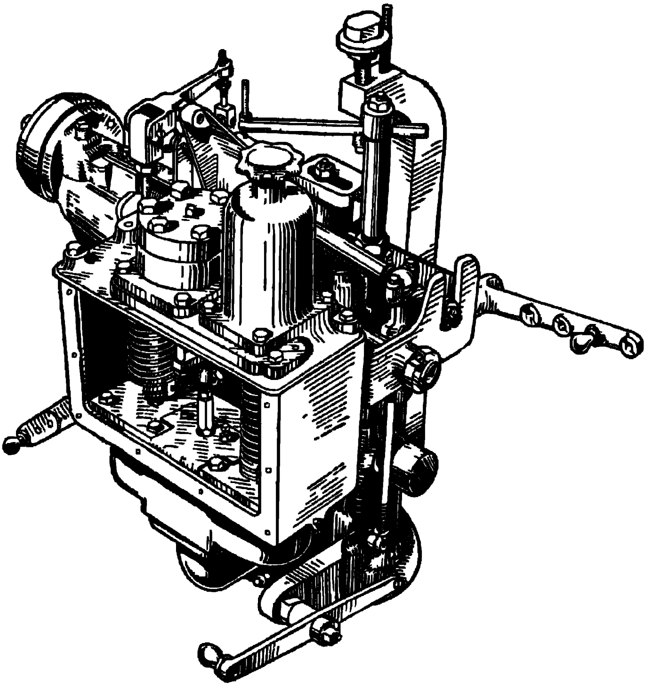

Fig. 8-77 Pressure governor

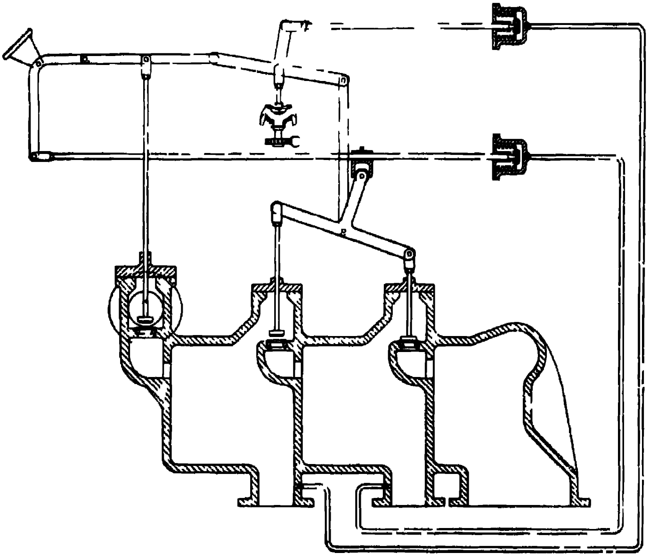

Fig. 8-78 Schematic arrangement for the simultaneous control of speed and extraction pressures

points will tend to raise the pressure at the automatic-extraction opening effected and thus activate the pressure governor which, as indicated in Fig. 8-78, will move the control valve (or valves) ahead of the extraction point in the closed direction and move the control valve (or valves) after the extraction point in the open direction. Thus, although the amount of steam passing through the turbine up to the extraction opening involved is decreased and hence the power output of that section is less, additional steam is passed through the turbine below the extraction point, and the power output in this section is increased correspondingly. Thus the loss of power in one section is offset by the gain of power in the other, and as a consequence little or no change in speed results. Upon an increase in extraction-flow requirements, the operation is similar but in the opposite order to that indicated.

The simultaneous control of speed and the extraction pressure at two automatic-extraction or admission openings is not uncommon.

Generators

Turbine-driven generators (Fig. 8-79) are available in sizes from less than 10 to over 100,000 kw, and single units have been built for ratings as high as 165,000 kw.

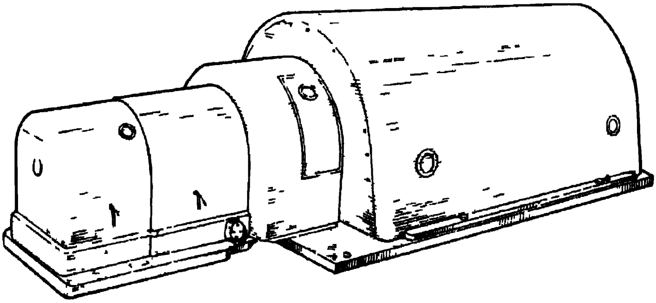

Fig. 8-79. Sixty-cycle, synchronous, a-c turbine generator.

The following description applies to units from 75 to 15,000 kw, inclusive, representing the sizes encountered in most industrial plants and in all but the larger central stations.

Types of Generators. In the size range from 75 to 15,000 kw, a-c generators are all of the revolving-field type and are generally furnished in either of two general types, as regards generator speed, as shown in Table 8-39.

Table 8-39

Kw rating	Type	Poles	Generator speed	
			60 cycles	50 cycles
75–1,500	Geared	6	1,200 rpm	1,000 rpm
2,000–15,000	Direct connected	2	3,600 rpm	3,000 rpm

The six-pole generators are of the so-called "salient-pole" type (Fig. 8-80), while the two-pole units are of the "round rotor" or nonsalient pole type (Fig. 8-81) required by the higher speed of operation. A six-pole generator can be built for ratings greater than 1,500 kw if necessary.

Geared-type six-pole generators are air-cooled and are generally furnished with open-type ventilation. Two-pole direct-connected generators from 2,000 to 11,500 kw are air-cooled and are generally made totally enclosed with air coolers mounted in a recirculating ventilating system. Air coolers may be mounted above or below the floor line as desired (Fig. 8-82). Generators 15,000 kw and above are generally hydrogen-cooled.

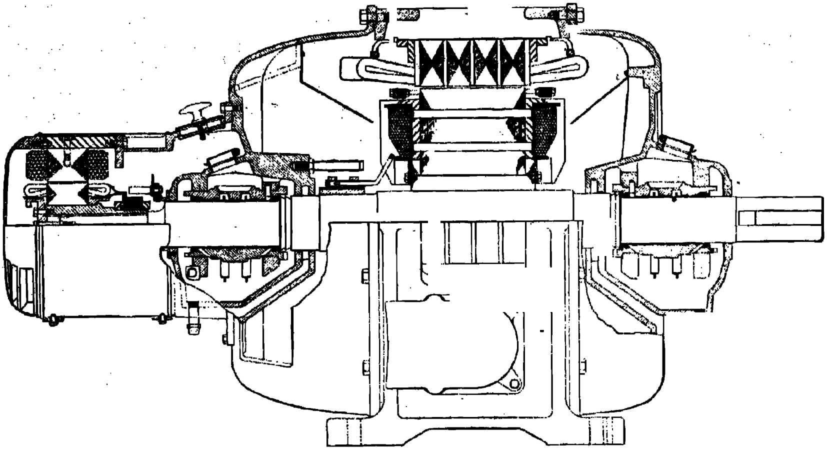

FIG. 8-80. Six-pole, "salient-pole," totally enclosed generator.

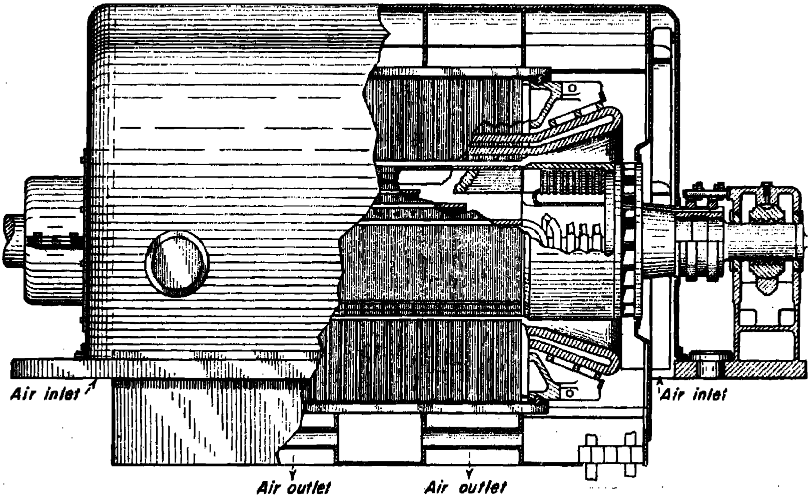

FIG. 8-81. Two-pole, "round-rotor," totally enclosed generator.

Standard Voltages. Standard terminal voltages of a-c generators in the size range from 75 to 15,000 kw are as shown in Table 8-40.

Table 8-40

Kw rating	*Standard voltages*
75–400	240; 480; 600; 2,400
500–1,500	480; 600; 2,400
2,000–7,500	2,400; 4,160; 5,000
10,000–15,000	6,900; 11,500; 12,500; 13,800; 14,400

Turbine-driven d-c generators are always of the geared type with standard speeds and voltages as listed in Table 8-41.

Table 8-41

Kw	Speeds, rpm		
	125 v	250 v	600 v
150 400	1 200	1 200	
500	750	900	900
750		900	900
1,000		750	750
1,500		514	514
2,000		750*	750*
2,000		360	514
2,500		750*	750*
2,500		. ...	514
3,000		514*	514*
4 000		360*	514*
5,000			514*

* Two generators coupled in tandem.

These generators are of the open type, air-cooled. They are also available for $^{125}/_{250}$ volts three-wire in each rating where 250 volts is standard.

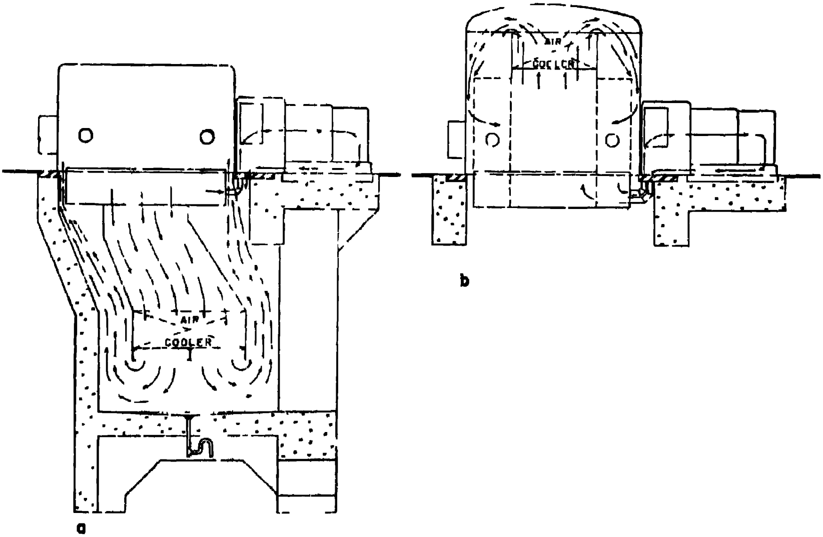

FIG. 8-82. Air-cooler arrangements: (a) cooler mounted in duct work below generator; (b) cooler mounted in top of generator housing.

Standard Characteristics. Present accepted standards for turbine-driven generators are contained in the following publications:

General Standards for Rotating Electrical Machinery—ASA, C50 (1943).[2]

Preferred Standards for Large 3,600-RPM Turbo-generators—AIEE, 601 (1945);[3] AIEE, 602 (1945).

NEMA Turbine-generator Recommended Practices NEMA Publication[4] No. 41-66 (1941).

Some of the main features of these standards are listed in Table 8-42 for convenience. For further details, refer to the standards themselves.

These standards apply to generators operating at altitudes of 3,300 ft or less. Generators designed for standard temperature rises at altitudes greater than 3,300 ft are generally built in oversize frames. Generators designed for 0.70 power factor or for short-circuit ratios higher than listed in Table 8-42 are also generally built in oversize frames.

Table 8-42

Kw	Power factor	Type	Ventilation	Short-circuit ratio	Excitation voltage	Temp. rise above 40°C	
						Armature	Field
75–400	0.80	Salient pole	Open	No standard	125	50°C thermometer	60°C (resistance)
500–1,500	0.80	Salient pole	Open	0.80–0.85	125	50°C thermometer	80°C (resistance)
2,000–7,500	0.80	Round rotor	Totally enclosed	0.80–0.85	125	60°C resistance-temp. detector	80°C (resistance)
10,000–15,000	0.80	Round rotor	Totally enclosed	0.90	250	60°C resistance-temp. detector	80°C (resistance)
11,500* and 15,000*	0.85	Round rotor	Totally enclosed	0.80	250	60°C resistance-temp. detector	85°C (resistance)

* AIEE-ASME preferred standard units.

Exciters

Direct-connected shunt-wound, self-excited exciters (Fig. 8-83) with direct-acting voltage regulators in the exciter field are standard for generators from 100 to 10,000

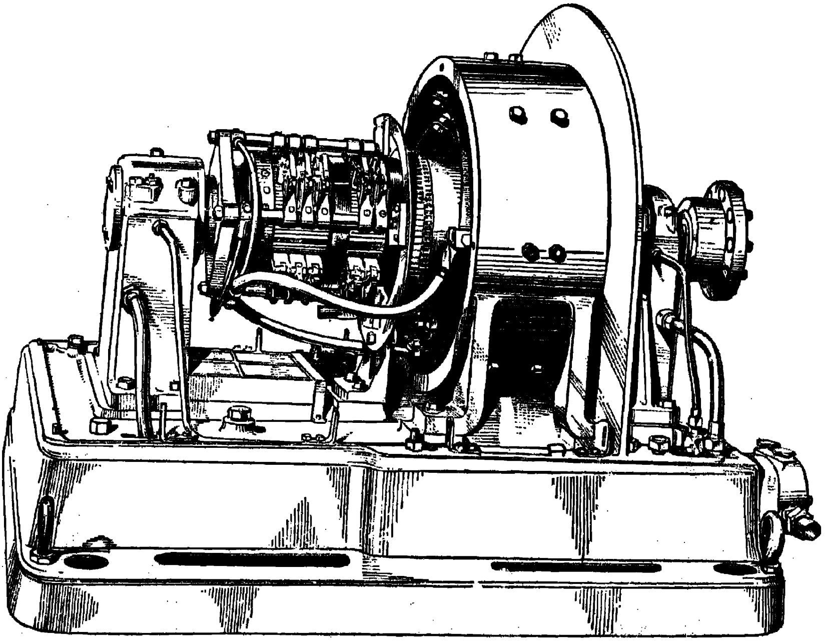

FIG. 8-83. Direct-current, shunt-wound, self-excited, 3,600-rpm exciter.

kw Above 10,000 kw the standards call for a direct-connected main exciter, shunt-wound and separately excited and a compound-wound pilot exciter (Fig 8-84) with hand or automatic control of the main exciter field

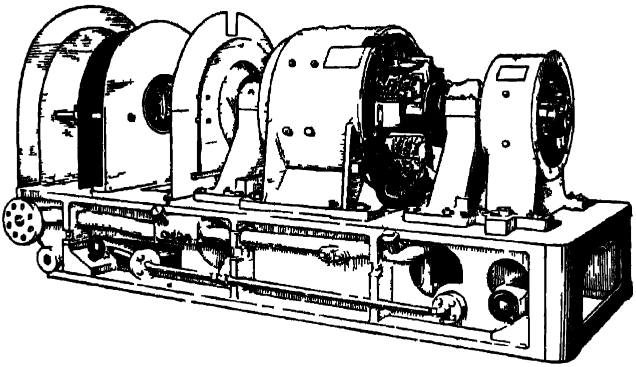

Fig 8 84 Main and pilot exciters

Accessories

For efficient and reliable operation a steam turbine generator unit is supplied with certain standard features and accessories These items vary somewhat with unit rating but the following tabulation is representative of the equipment furnished

Condensing and Noncondensing Steam Turbines (except low pressure—those with 20 psig or lower inlet pressure)

1 Steam strainer with removable element

2 Combined trip and throttle or stop valve actuated by an overspeed governor The action of the emergency governor is transmitted to the valve either through a mechanical linkage or a fluid oil relay system (Fig 8 85)

3 Overspeed governor entirely independent of the normal speed-regulating governor

4 Multiple automatic normal speed governing valves (see Fig 8-75 under Governing)

5 Speed regulating governor to control the position of the normal-speed governing valves through the medium of a relay-controlled hydraulic-valve-operating mechanism (see Fig 8 75 under Governing)

6 Oil pump mounted upon or positively driven from the unit to provide oil for the valve-operating mechanism and lubricating oil to each bearing of the unit except ring-oiled generator and exciter bearings on geared units To protect the pump from damage, a strainer is usually located at the suction passage of the pump

7 Oil reservoir of sufficient capacity for the unit requirements

8 Oil cooler of sufficient size to remove the heat absorbed in the oil and keep the bearing discharge oil temperatures at approximately 140 to 160°F when the cooler is supplied with cooling water at a temperature specified by the purchaser but not exceeding 95 F These are either of the single- or double-pass type with the water flowing through the tubes and the oil directed by baffle plates around the tubes

9 Steam-driven auxiliary oil pump (Fig 8-86) with an automatic starting device actuated by a reduction from normal oil pressure in the system This control device is usually an oil pressure positioned diaphragm which regulates the inlet valve to the driving turbine

10 Oil-pressure trip mechanism or its equivalent to stop the flow of steam into the turbine should the oil pressure in the system drop to too low a value

11 Oil piping connecting all integral parts of the turbine-generator-unit oiling system except piping between the turbine unit and the oil cooler (and between the

turbine unit and auxiliary oil pump) Oil piping will be of steel and will have welded or flanged connections

12 Vibrating-reed or electric indicating tachometer or frequency meter calibrated in rpm to indicate the turbine speed

13 Necessary operating gauges with gauge board or other suitable mounting (Fig 8-87) Only when gauges or gauge boards are mounted on the unit will gauge pipings be included

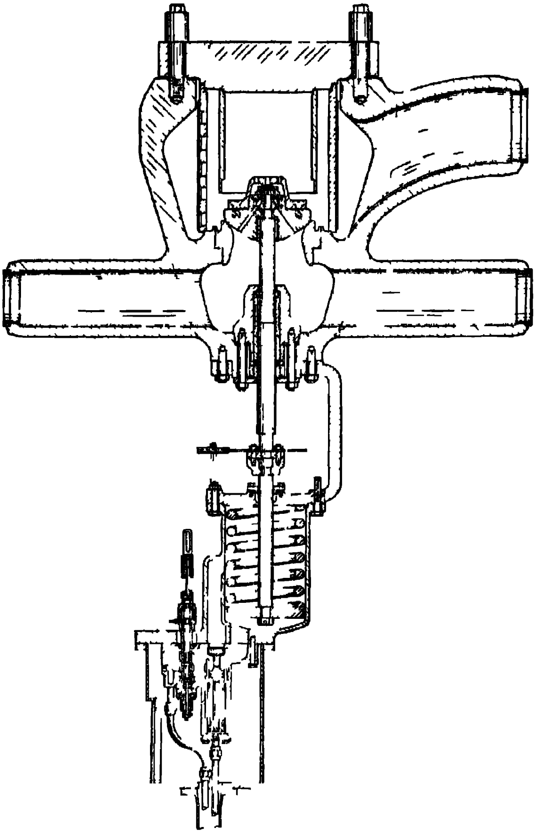

Fig 8-85 Trip stop valve

Standard gauge equipment includes

a Steam gauges for determining
Initial pressure
First-stage pressure
Exhaust pressure

b Oil-pressure gauges—usually one reading bearing pressure and the other governor oil pressure

c Gland-water pressure gauges if the glands are of the water seal type

14 Gland-steam condenser or condensers (when the gland design requires their use) but not including piping between the unit and such gland condensers Sometimes a wet vacuum-pump arrangement is provided These systems serve to prevent gland-steam leakage into the bearing housings and the turbine room To eliminate

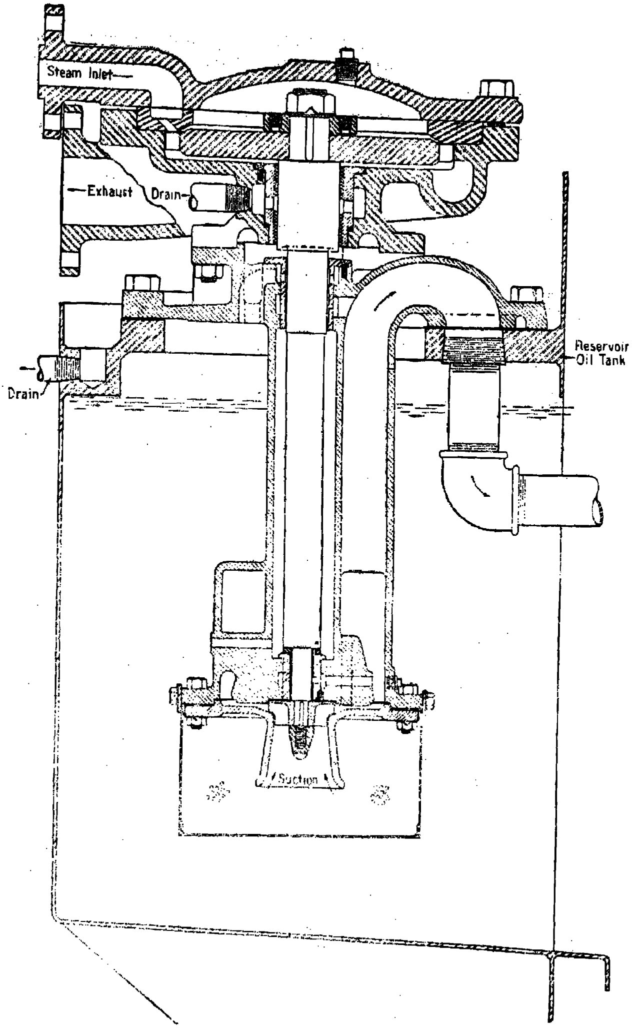

FIG. 8-86. Steam-turbine-driven auxiliary oil pump.

manual adjustment of the sealing steam with changes of load, some manufacturers also provide automatic steam-seal regulators.

Automatic-extraction Turbines. Automatic-extraction turbines normally include all the features and accessories listed for condensing and noncondensing steam turbines together with the following additional features and accessories:

1. Extraction-pressure gauge
2. Automatic device for controlling the pressure of the extracted steam by varying the flow of steam to the lower pressure stages of the turbine
3. Interconnection between the speed governor and extraction control, either mechanical or hydraulic
4. Automatic nonreturn valve for each automatic-extraction outlet, with trip actuated from the overspeed governor, usually through an oil relay

Low-pressure Turbines (those with 20 psig or lower inlet pressure). Low-pressure turbines usually include all the features and accessories listed for condensing and noncondensing steam turbines with the following exceptions and additions:

1. Omit steam strainer.
2. Instead of combined trip and throttle valve, substitute a simple trip valve in low-pressure steam supply line, actuated by overspeed governor (gate or throttle valve not included).
3. A single governor valve may be substituted for multiple automatic valves.

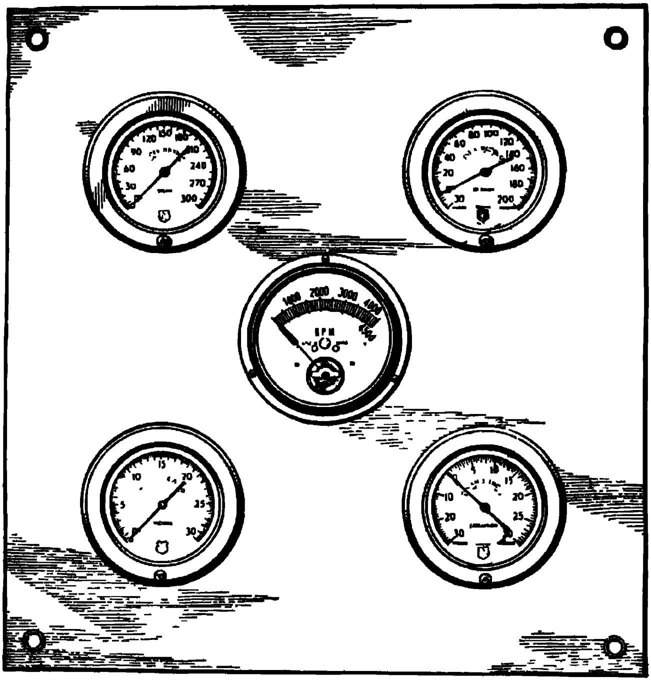

FIG. 8-87. Turbine gauge board.

Mixed-pressure Turbines. In general, mixed-pressure turbines include all the features listed for condensing and noncondensing steam turbines, with the exception of the steam strainer on l-p inlet. Furthermore, they have the following additions:

1. Low-pressure steam gauge
2. Device to control the pressure in the lower pressure steam line by varying the low-pressure steam flow through the turbine
3. Interconnection of l-p governor valve to speed governor and to high-pressure governor valves.
4. Simple trip valve in low-pressure steam line actuated by overspeed governor (gate or throttle valve not included when lower pressure is 20-lb gauge or less)

Reduction Gears. Gears are of the helical type and specially designed for high-speed continuous service. They are of the totally enclosed, forced-flood lubricated type and include the following features and accessories:

1. Complete lubricating system as outlined under Condensing and Noncondensing Steam Turbines, Item 11.

Generators and Exciters. 1. Field-discharge resistor.

2. Embedded temperature detectors in stator windings for 1,500 kw and above.

3. Field wound for 125 volts up to 2,500 kw. Above 2,500 kw, field wound for either 125- or 250-volt excitation.

4. Exciter-field rheostat, hand operated up to 7,500 kw and motor operated above 7,500 kw.

General Accessories—Unit as a Whole. In accordance with present practice, the following unit features and accessories are usually furnished:

1. Wrenches for all special bolts and nuts (standard wrenches not included)
2. Soleplates or base plates as necessitated by the particular turbine and generator design
3. Couplings
4. Turbine-rotor lifting gear, if necessitated by the particular turbine design

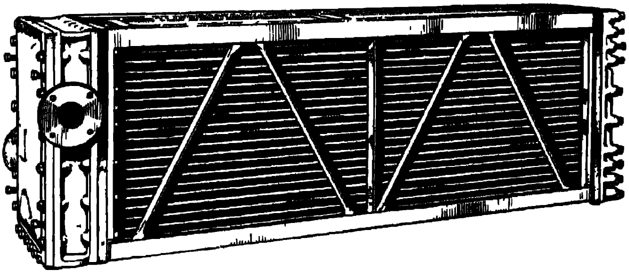

Fig. 8-88. Generator-surface air cooler.

Special Accessories. Certain other accessories, not normally furnished as standard equipment, are desirable for safe and efficient operation. These include generator air coolers (Fig. 8-88), designed for mounting either in duct work below the generator or on top of the generator within the housing of the generator, generator-temperature indicator, thermometers, and positive-type nonreturn valves for uncontrolled extraction lines where there is a possibility of reverse steam flow of sufficient volume to overspeed the turbine. If required, most turbine manufacturers can furnish this equipment.

Over-all Dimensions and Weights

Some concept of the approximate over-all size and weight of a turbine-generator set is usually of value for preliminary station layouts Table 8-43, together with Figs 8-89 *a*, *b*, and *c*, indicates such data for straight condensing units and is sufficiently accurate for estimating purposes. Automatic-extraction units will tend to be somewhat larger - approximately one foot in length—and noncondensing units will be slightly shorter.

Foundations

This material outlines a few general suggestions which may prove helpful in the design and construction of the foundation for a turbine-generator set and which experience proves are very frequently overlooked in the initial design of the structure.

Types of Foundations. Of the several different foundation types from which to choose, ranging from plain massive concrete construction to the other extreme of an entirely structural-steel construction, general practice seems to indicate that foundation construction of the reinforced-concrete type fills the requirements of simplicity in design and construction, necessary amount of rigidity, fair accessibility, and economy of space at a reasonable cost. As a consequence, the following comments are confined mainly to this type of construction, a typical example of which is shown in Fig. 8-90.

Provision for Proper Clearance. Although it appears self-evident, too much attention cannot be given to providing clearances necessary to prevent interference between the turbine parts, especially piping and flange connections and the foundation. For the sake of appearance such parts are usually located below the level of the operating floor. Chipping and burning operations on a completed foundation are too frequently unpleasant and costly indications that this detail was overlooked.

Table 8-43

Rating, kw	Approximate dimensions: Length, With exciter	Approximate dimensions: Length, Without exciter	Approximate dimensions: Width	Approximate dimensions: Above-floor height	Generator, rpm	Outline sketch	Weight, lb
75	12 ft 0 in.	10 ft 9 in.	5 ft 6 in.	5 ft 6 in.	1,200	Fig. 8-89*A*	10,000
100	12 ft 6 in.	11 ft 3 in.	5 ft 6 in.	5 ft 6 in.	1,200	Fig. 8-89*A*	11,000
125	13 ft 0 in.	11 ft 9 in.	5 ft 6 in.	5 ft 6 in.	1,200	Fig. 8-89*A*	11,500
150	13 ft 6 in.	12 ft 3 in.	6 ft 0 in.	5 ft 6 in.	1,200	Fig. 8-89*A*	12,500
200	14 ft 0 in.	12 ft 9 in.	6 ft 0 in.	6 ft 6 in.	1,200	Fig. 8-89*A*	14,000
250	15 ft 0 in.	13 ft 9 in.	6 ft 6 in.	6 ft 6 in.	1,200	Fig. 8-89*A*	15,500
300	16 ft 0 in.	14 ft 9 in.	6 ft 6 in.	7 ft 0 in.	1,200	Fig. 8-89*A*	17,000
350	16 ft 6 in.	15 ft 3 in.	7 ft 0 in.	7 ft 0 in.	1,200	Fig. 8-89*A*	18,500
400	17 ft 0 in.	15 ft 9 in.	7 ft 0 in.	7 ft 6 in.	1,200	Fig. 8-89*A*	20,000
500	17 ft 0 in.	15 ft 9 in.	7 ft 0 in.	7 ft 6 in.	1,200	Fig. 8-89*B*	24,000
700	18 ft 0 in.	16 ft 9 in.	7 ft 0 in.	7 ft 6 in.	1,200	Fig. 8-89*B*	35,000
1,000	19 ft 6 in.	17 ft 6 in.	8 ft 0 in.	7 ft 6 in.	1,200	Fig. 8-89*B*	47,000
1,500	21 ft 0 in.	19 ft 6 in.	8 ft 0 in.	7 ft 6 in.	1,200	Fig. 8-89*B*	60,000
2,000	23 ft 0 in.	20 ft 6 in.	8 ft 6 in.	7 ft 6 in.	3,600	Fig. 8-89*C*	70,000
2,500	25 ft 0 in.	21 ft 6 in.	9 ft 6 in.	7 ft 6 in.	3,600	Fig. 8-89*C*	78,000
3,000	26 ft 0 in.	22 ft 6 in.	10 ft 6 in.	7 ft 6 in.	3,600	Fig. 8-89*C*	86,000
3,500	27 ft 0 in.	23 ft 6 in.	10 ft 6 in.	7 ft 6 in.	3,600	Fig. 8-89*C*	94,000
4,000	28 ft 0 in.	24 ft 6 in.	11 ft 0 in.	8 ft 0 in.	3,600	Fig. 8-89*C*	102,000
5,000	30 ft 0 in.	26 ft 0 in.	11 ft 0 in.	8 ft 0 in.	3,600	Fig. 8-89*C*	118,000
6,000	31 ft 0 in.	27 ft 0 in.	11 ft 0 in.	8 ft 0 in.	3,600	Fig. 8-89*C*	133,000
7,500	32 ft 0 in.	28 ft 0 in	11 ft 0 in.	8 ft 0 in.	3,600	Fig. 8-89*C*	142,000
10,000	35 ft 0 in.	29 ft 6 in.	11 ft 6 in.	8 ft 0 in.	3,600	Fig. 8-89*C*	152,000
12,500	36 ft 6 in.	11 ft 0 in.	11 ft 6 in.	8 ft 0 in.	3,600	Fig. 8-89*C*	165,000
15,000	38 ft 0 in.	32 ft 5 in.	11 ft 6 in.	8 ft 0 in.	3,600	Fig. 8-89*C*	175,000

Loads and Deflections. Turbine manufacturers' outline prints usually indicate the dead load and the vacuum load which must be supported by the foundation,

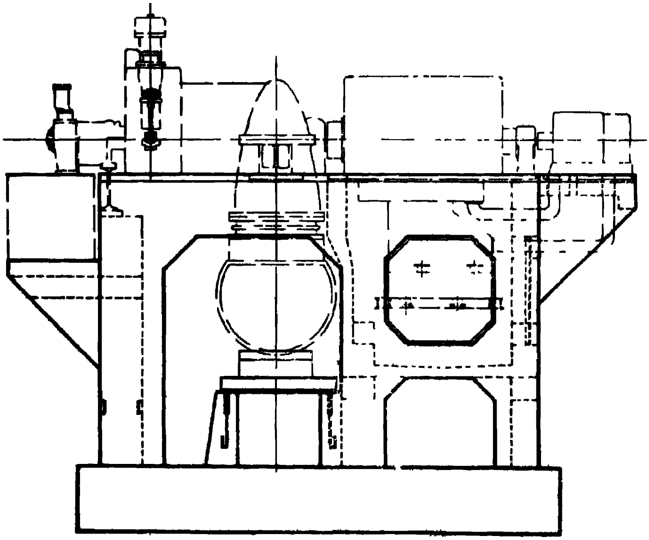

Fig. 8-89. Turbine-generator foundation.

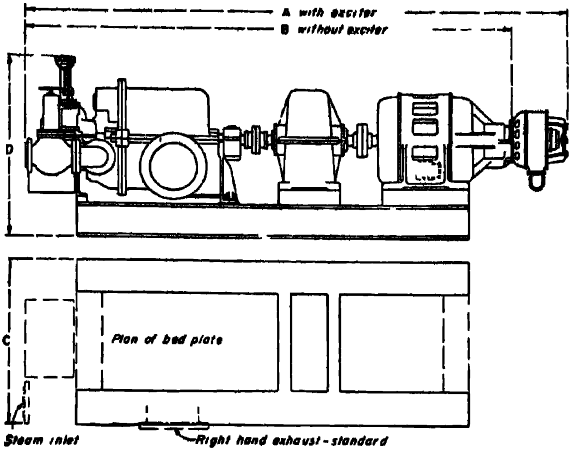

Fig 8-90a. Outline, 60-cycle geared units.

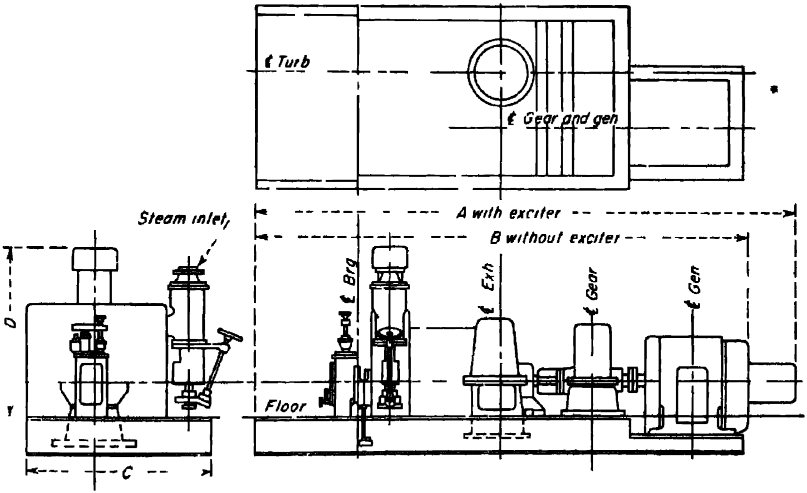

Fig. 8-90b. Outline, 60-cycle geared units.

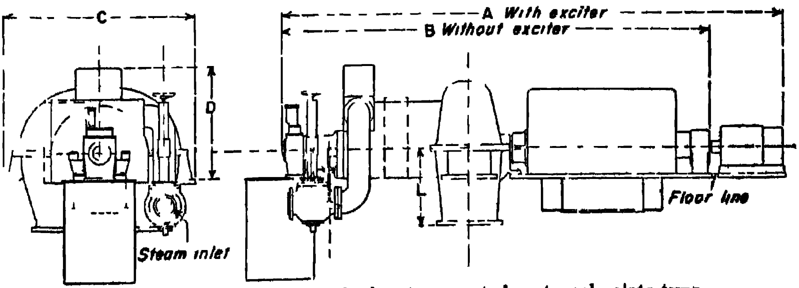

Fig. 8-90c Outline, 60-cycle direct-connected units, sole-plate type.

together with the points of application of these loads. However high-speed machines such as turbines have a live-load characteristic which should be allowed for in the design of a foundation. It is good practice to consider this factor as an impact load and figure it as an additional load which is equivalent to one-half the dead and vacuum load as applied at the various loading points on the foundation and which should be added to the dead and vacuum load to determine the total foundation loading.

Since serious operating difficulties result when a turbine-generator set is misaligned, deflections rather than unit stress are selected as the criterion of strength for the structure. For main supporting beams, the maximum deflection based on the above total loading consisting of dead load, vacuum load, impact load, and the weight of the beam itself should not exceed 0 020 in., and the closer this deflection approaches zero, the better. To provide horizontal rigidity, cross bracing and column reinforcement should limit the horizontal deflection at the foundation deck to not over 0.020 in., based on a force equivalent to one-half the total dead load acting in a horizontal direction at the center of the turbine shaft and at right angles to the axis of the shaft, parallel to the foundation deck.

Vibration. Since the vibration of the unit will be transmitted through the entire structure on which the set rests, it is recommended, as a means of limiting the transmission of the unit's vibration, that its foundation be separated from the station floor or any other part of the building structure. Because of resonant frequencies, almost imperceptible vibrations in the foundation structure itself may be amplified to decidedly objectionable proportions in interconnected adjacent structures. This is especially true for galleries or any other overhanging construction supported from the foundation.

***Subbase.** The subbase under the foundation should be sufficiently rigid to prevent local settling of any part of the foundation structure. Such settlement causes misalignment of the turbine, and vibration and rubbing difficulties may result. Separate footings for each column should be considered only where the foundation footings can be extended down to bedrock. Where this is not possible, a study of the soil may indicate that a reinforced-concrete mat will give the desired support.

Concrete Mixture. A concrete mixture of 1:2:4, based on using only the best material throughout, is usually adequate for these foundations. If poured during cold weather, necessary protection against freezing should be provided, and in all cases, proper setting time should be allowed before applying load. To secure the benefits of a first-class design, careful workmanship is a necessity in the construction of the foundation.

OPERATION OF STEAM-TURBINE GENERATORS

The installation of a turbine-generator unit represents a large plant investment To protect this financial outlay and to avoid unnecessary unit outages, care and precaution should be exercised in the operation of these sets Although turbine construction is inherently sturdy, efficient, and reliable, operation of these units depends on the maintenance of comparatively small clearances between the stationary and rotating parts. As a consequence, definite precautions should be observed in the starting, stopping, and normal operation of these sets.

Turbine manufacturers usually provide instruction books for the turbines which they build. When such data are available, reference should be made to them for detailed operating instructions. However, in the absence of such specific information, the procedures enumerated below can be used. While the instructions tabulated apply primarily to condensing units, they can also be used for noncondensing and automatic-extraction turbines by including several minor variations.

Turbine-generator Starting Procedure— Condensing Turbines (Fig. 8-91)

1. Open all casing drain valves.
2. Open drain ahead of throttle valve to drain condensate from the main, and then slowly open stop valve ahead of the throttle valve.
3. Start auxiliary oil pump, check oil pressure, and look for oil flow through bearing-sight flows.

4. Check level in oil reservoir.
5. Establish circulating-water flow through the condenser.
6. Start condenser air pumps.
7. Start condensate pump.
8. When drain ahead of throttle valve blows clear, open the throttle valve sufficiently to start the turbine rotor revolving, and then quickly shut off by
9. Tripping the emergency hand control to check its operation.
10. Listen for rubbing or internal noise within casing, using a metal rod or other listening device.
11. If no rubs are evident, reestablish steam flow and bring rotor speed up to about 10 to 15 per cent of the normal speed rating. Maintain about one hour to warm up rotor and casing evenly as soon as rotor is rolling steadily.
12. Close all casing drains or transfer to traps to increase vacuum as much as possible.

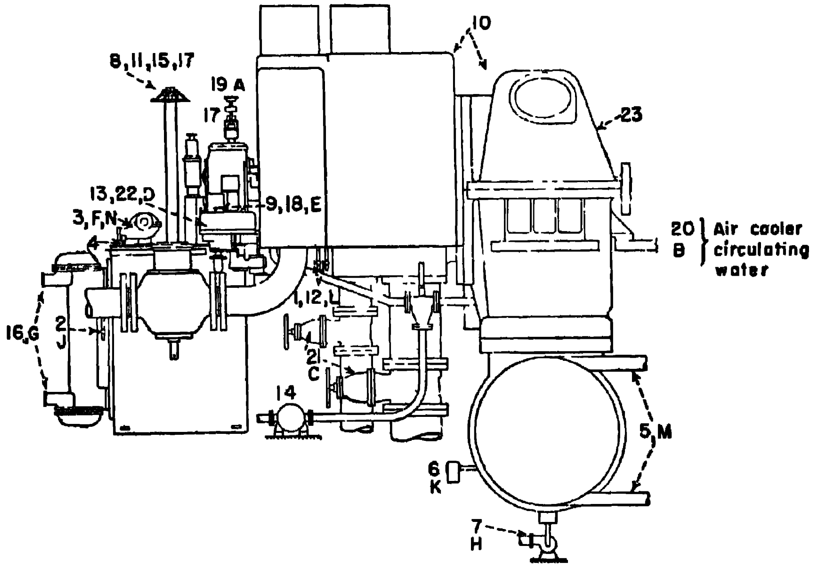

FIG. 8-91. Turbine starting and shutdown procedure.

13. Open packing-drain valves and admit sealing steam to the high-pressure and low-pressure shaft packing boxes using steam-seal regulator if turbine is so equipped. If water-sealed, admit water as required.
14. Start packing-drain condensers or vacuum pumps.
15. Gradually increase the speed to normal rated speed. If any abnormal vibration or noise is experienced at any time during acceleration, decrease the speed to 10 per cent of normal and continue the warm-up period until no difficulty is encountered during acceleration. Do not force a turbine to speed.
16. When the temperature of the oil leaving the bearings reaches 140°F, start the circulation of water to the oil cooler so as to maintain the temperature of the oil leaving the bearings between 140 and 160°F.
17. As the turbine approaches rated speed, observe to see that the governor takes control. When the turbine is operating under the control of the governor, open the throttle valve wide.
18. Trip the emergency device and see that the throttle valve closes promptly. Then reset the tripping device, open the throttle valve, and again bring the turbine up to rated speed.

19. Excite the field, synchronize the generator, connect it to the system, and apply about 20 per cent of normal load.

20. Start the water circulating in the generator air cooler if installed.

21. Open bleed-line valves and place heaters in operation.

22. Adjust high-pressure and low-pressure seals for operating condition, if necessary.

23. After exhaust hood has become cool, apply additional load as desired.

Turbine-generator Shutdown Procedure—Condensing Turbines (Fig. 8-91)

A. Gradually diminish the load to zero, disconnect the unit from the line, and remove the generator field excitation.

B. Shut off the water to the generator air coolers.

C. Close bleed-line valves, and take heaters out of service.

D. Close steam or water supply to high-pressure and low-pressure shaft packing.

E. Close the throttle valve by manually tripping the emergency device.

F. Check to see that the auxiliary oil pump starts as oil pressure falls

G. Shut off the water to the oil coolers.

H. Shut down condensate pump.

J. Close the stop valve ahead of the throttle valve, and open the drain ahead of the throttle valve.

K. To protect against internal corrosion, keep the condenser air pumps operating for about twenty minutes. Then shut down.

L. Open all atmospheric drains

M. Shut down condenser circulating-water pumps.

N. Keep auxiliary oil pump in operation until bearings are cool.

Special Procedures

Noncondensing Turbines. For noncondensing turbines, the same procedure applies except that condenser operations are naturally omitted. Turbines designed to exhaust against high back pressure may be brought to normal speed at no load when exhausting against atmospheric pressure. Before placing the unit on the line, the back pressure is raised to normal by gradually closing the atmospheric-relief valve and gradually opening the exhaust-line valve. As the back pressure is increased, the throttle valve is opened further to maintain speed until ultimately the turbine governs on the speed governor and the throttle is open wide.

Where it is necessary to start the turbine on high back pressure, the following starting procedure is recommended so as to avoid drawing an excessive amount of steam from the system.

1. Have oil pressure on the governing apparatus and bearings.
2. Turn the synchronizing handwheel so as to close the controlling valves.
3. Have the back pressure on the turbine.
4. Open the throttle valve wide.
5. Gradually open the controlling valves by means of the synchronizing handwheel until the rotor of the turbine starts turning.
6. As soon as the rotor starts to turn, regulate the rate of acceleration by opening or closing the controlling valves by means of the synchronizing handwheel.
7. When the governing speed is reached, synchronize the generator and connect it to the system.
8. If the unit is governed by a back-pressure governor, turn the synchronizing handwheel to the high-speed stop position.

Automatic-extraction Turbines. In placing automatic-extraction turbines on the line, they are brought up to speed and loaded first as straight condensing or noncondensing machines, as the case may be, and then the extraction valves are put in operation. Order of procedure is reversed for shutting down.

Routine Operation

Once a turbine-generator set has been connected into the system, normally very little attention is required during its operation. However, as precautionary measures, the following routine operational checks are recommended.

As with any other high-speed rotating equipment, bearing-discharge-oil temperatures and shaft vibration should be checked. Changes from normal values should be investigated for the cause of the variation. Higher oil temperatures may mean clogged oil passages or loss of cooling water.

Inlet- and exhaust-steam conditions should be observed and maintained as close as possible to normal. Stage-shell pressure should be noted in connection with electrical load. Any marked variation in these pressures,[6] if it cannot be explained by a change in operating conditions, may be an indication of plugging in the nozzles or buckets. The various safety devices should be operated occasionally to ensure their being available for use in an emergency. Generator temperatures should be read to ascertain whether or not the generator is being cooled properly.

These routine checks are quickly and easily made and serve to inform the operator as to the proper functioning of his equipment.

MAINTENANCE OF STEAM-TURBINE GENERATORS

Every turbine operator is confronted with the necessity of having his equipment available for use for the maximum percentage of time. To obtain this goal, it is desirable not only to follow established operating procedures as outlined under Routine Operation but also to follow a systemized method of planned inspection and overhaul. Such maintenance reduces outages, guards against hazardous operation, and aids in securing maximum possible steam economy from the unit. The following material contains suggestions which may be helpful in expediting the inspection and overhaul of steam-turbine generators.

Planning Preliminary to Overhaul

To reduce the outage time to a minimum, careful planning prior to the overhaul should be undertaken. Inspection data such as instruction books, drawings, governor-setting diagrams, and previous inspection records should be reviewed and studied. A tabulation of past operation difficulties to be corrected during the overhaul should be made up. Tables or charts on which can be recorded any measurements or notes giving information concerning the condition of the various elements should be prepared in advance. Any repair parts to be installed should be cleaned, inspected, and checked to see that no parts are damaged or missing. Rigging, blocking, tools, and special equipment required for the overhaul should be obtained and placed in first-class condition. Also, to secure the maximum possible benefit from the overhaul program, it is recommended that a competent field-service representative be obtained from the turbine manufacturer to supervise the work. This preliminary planning, if carefully done, will effectively decrease the time the unit will be out of service for overhaul.

Accurate Records

If the records of the parts inspected are to be useful in determining the deterioration rate of the various parts, and this is an important factor in the scheduling of future overhauls and in the ordering of replacement parts, the necessity of making careful and accurate records cannot be overemphasized. Photographs, wax impressions, templates, and sketches are all useful in obtaining accurate and lasting records. Such data when sent in to the turbine manufacturer assist in determining the proper replacement material and also aid in making recommendations for repairs. Many inspection reports contain only the inspector's opinions or conclusions and fail to include the basic facts. Such reports fail to provide valuable records of actual conditions. There is no substitute for accurate facts.

Turbine

Experience has indicated the potential troubles to be looked for in inspecting turbine generators and their accessories. Some suggestions which may prove helpful are enumerated below. Maintenance of the turbine is discussed first.

Steam-path Parts. The steam-path parts consisting of the nozzles and buckets or the blading should be examined carefully for the accumulation of deposits, erosion, corrosion, and cracks. An accurate record of the condition of these parts can be obtained by dimensional measurements and by the use of templates, wax impressions, and photographs. The location of plugged or corroded sections should be recorded. Samples of the deposits should be taken for laboratory analysis which may indicate means of eliminating further deposits or the development of a method for removing the deposits without having to open the turbine. The use of coarse sand blast for cleaning the steam-path parts is not recommended as this method produces rough surfaces which increase steam consumption Coarse sand blasting also produces sharp edges from which cracks may start, and there is also the danger of cutting away valuable metal in an attempt to remove hard deposits. Cleaning with a fly ash blast or fine sand (24 mesh or finer) (Fig. 8-92) or hand scraping is preferable to the use of coarse sand blast. Cleaning pays dividends in improved steam economy and in the reduction of the thrust load.

After the steam-path parts are cleaned, they should be examined minutely for erosion and small cracks. Erosion of nozzles and buckets or blading produces sharp corners or notches having high stress concentration which may result in the starting of cracks. Therefore, these sharp eroded portions should be removed and the edges smoothed and polished. Cracks or flaws in the blading may be discovered by a visual inspection with the aid of a small mirror similar to a dental mirror. Imperfections can be indicated by chalk marks and also recorded in writing. Another method is to apply penetrating oil such as kerosene to the cleaned blades, wipe off the excess oil, and then paint them with whiting If any cracks exist, the kerosene usually stays in the cracks and as soon as the whiting is dry the penetrating oil in the cracks will show through the whiting coat as a yellow streak. A magnetic particle test[9] may also be used to examine the buckets and nozzle blading when these parts are made of a magnetic material.

FIG. 8-92. Cleaning turbine blading with a fly ash blast.

Wheels, Rotors, and Shafts. Since the rotating parts of a steam turbine are relatively highly stressed at speed, it is important that these parts be cleaned down to bare metal so that any defects such as cracks will be disclosed. Tests for cracks should be made by either the "kerosene-whiting" method or the magnetic particle test previously mentioned under steam-path parts Wheels should be checked for movement on the shaft, cracks at the keyway, and sharp edges at the equalizer holes. The sound of each wheel when tapped with a hammer should be noted. A dead-sounding wheel is not necessarily a defective one, but a careful investigation should be made of any wheel having a dead sound if it previously had a bell-clear sound when tapped with a hammer, as such a change indicates a change in the solidity of the rotor assembly. Loose or cracked wheels often sound dead. Before any welding is attempted on any rotating parts of a turbine, the manufacturer should be consulted. Welding, improperly done, can set up internal stresses and also change the structure of the metal and cause cracks which may progress to failure.

Rough or pitted journals should be stoned, ground, or turned, depending upon the extent of the surface damage. If a suitable lathe is not available, this work can be done with the rotor in the unit. Figure 8-93 shows a No. 2 journal being ground with the rotor in place, using temporary bearings at the oil-deflector fits and driving the rotor by means of a motor belt drive on the shaft coupling. Figure 8-94 shows the setup for grinding a No. 1 bearing journal. If it is found necessary to true up a coupling face, great care should be taken in preparing the setup for doing this work. The rotor should be blocked so that it cannot move axially. To check the coupling face after machining or grinding, an indicator should be placed at a point directly opposite the point where the cutting tool is mounted. If the indicator is placed in the tool post where the cut is taken, it will usually indicate zero under any condition, whereas if the indicator is mounted directly opposite the tool post, it will indicate a runout double the actual runout if there was any axial movement of the rotor during the machining. This check should be made on all coupling faces being faced either with the rotor in its bearings or with the rotor in a lathe. Very often coupling faces are improperly machined in a lathe because the lathe-spindle thrust runs out, which causes a slight axial movement of the rotor. Coupling faces should be machined flat to 0.001 in. concave. Coupling faces must not be convex if good operation is expected.

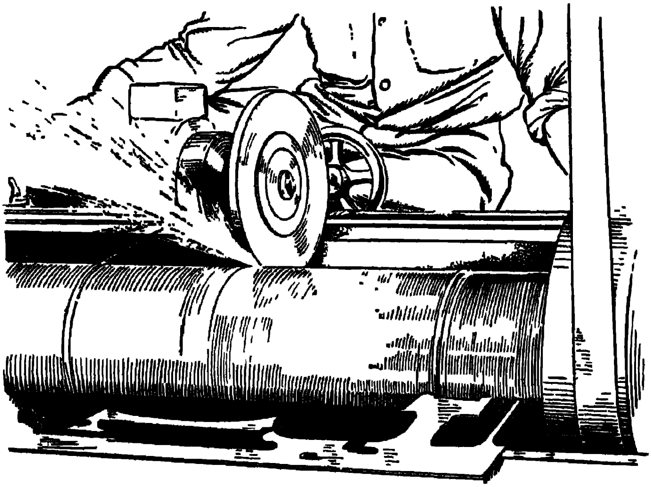

FIG. 8-93. Grinding a main journal.

Bearing Linings and Thrust Bearing. The bearing linings should be examined for wear. The clearances between the journal and the bore of the bearing should be measured and recorded. The clearance in the bearing can be measured either by shims or by lead wire. Bearing linings are usually manufactured with a vertical clearance over the shaft of 0.0013 in. per in. of shaft diameter. If the rotor is in good balance with no tendency to vibrate, rebabbitting may not be necessary until the vertical clearance has become 0.0025 in. per in. of shaft diameter. Records of past performance will usually indicate the allowable bearing clearance.

The babbitt surfaces should be inspected for cracks, high spots, and separation from the bearing shell. Cracking and separation may be caused by vibration in the rotating parts and if found will necessitate rebabbitting. High spots in the babbitt surface should be scraped down lightly to form a uniform bearing surface.

The thrust bearing maintaining the axial position of the rotor in relation to the casing should always be examined. The condition of the babbitt surface should be noted, and the clearance of the bearing should be readjusted to that specified by the manufacturer. Usually this clearance runs from about 0.006 to 0.012 in. depending upon the size and type of bearing.

Shaft Packing. The shaft packings of steam turbines usually fall into one of three classes—the labyrinth type, carbon ring, or water seal, or in some cases combinations of these types. These shaft seals should be examined for wear. The packing clearances recommended by the manufacturer should not be reduced, or vibration trouble may be encountered. By careful machining, worn packing can often be reconditioned. If the shaft-packing fits are found to be pitted or eroded, turning or grinding will recondition the surface. The packing must then be refitted. Where metal-spray equipment is available, shaft-packing fits can sometimes be satisfactorily repaired thereby maintaining the original shaft diameter at the packing fit, but the manufacturer should be consulted for his recommendations on this procedure.

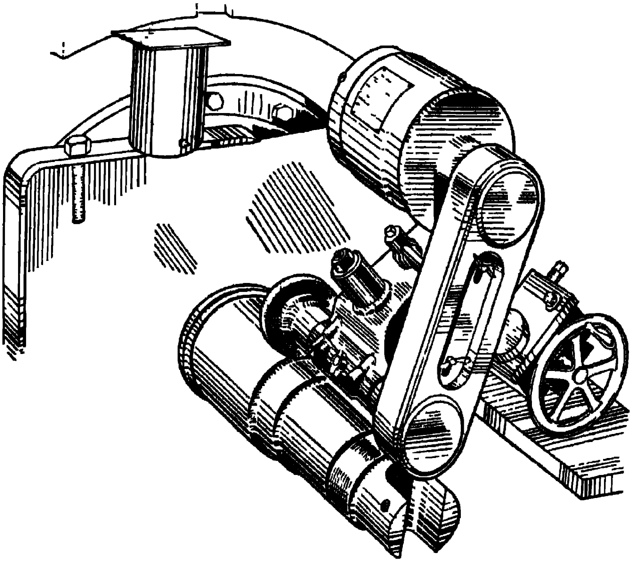

FIG. 8-94. Setup for reconditioning a No. 1 bearing journal by grinding without removing the rotor from the casing.

Casing Erosion. Where casing erosion is experienced, it is usually caused by the accumulation of a high moisture content in the expanding steam. The tiny droplets of water hurled from the rotating blades with a high velocity impinge against the casing and wear away the surface. If possible, the initial superheat of the steam should be increased or the turbine manufacturer should be consulted as to the possibility of adding moisture separators to the casing to remove this moisture which not only causes casing erosion but also reduces the steam economy of the unit.

Repair of the eroded surface can be made either by welding or by the insertion of filler strips or patch plates. Welding, if possible, is usually the less costly process, but caution must be exercised to avoid casing distortion during welding. Where the casing is steel, repair can be made by welding up the eroded portion. On a cast-iron casing it is usually necessary to machine down the surface and install patch plates to repair steam joints, although it may be possible to effect repairs by bronze or monel welding. Erosion elsewhere can be restored with metal spray of a noncorrosive material or by covering the worn spot with patch plates so that future wear will be on the plate which is replaceable. In all cases of serious erosion or corrosion it is advisable to determine the casing thickness by making a template of the contour which can be checked against the drawing dimensions by the manufacturer.

Lubrication System. Cleaning of the lubrication system is an important item in the overhaul work. It is a hard, tedious task to clean an oil system properly, but it is worth while. A clean lubrication system results in longer oil life, reduces sludge, and helps prevent corrosion or rusting of parts. If any part of the lubrication system requires painting, be sure the proper paint is used. And, equally important, the surface must be properly prepared, or the paint may come off with disastrous results. The oil supplier should be consulted for information on cleaning solvents and painting of lubrication systems. A partly cleaned system is a hazard as sludge or dirt from the dirty section may be dislodged and block lubrication.

The main oil pump usually of the positive-displacement gear type and driven at reduced speed through a worm-gear drive should be carefully inspected for proper alignment and wear of the moving parts. The bearing and guide-bushing clearances should be checked against the manufacturer's recommendation. The pump-casing clearance and the wear of the pump gears should be recorded. The condition of the driving worm on the turbine shaft and of the driven gear on the pump shaft should be noted and a careful alignment made of these parts.

The auxiliary oil pump, usually of the centrifugal-type driven by a steam turbine should be given an inspection similar to that described for the main oil pump.

Governor and Valve Gear. The various bushings and pins on the governor and the connecting valve gear linkage should be examined for excess clearance and the condition of the parts recorded. Those showing abnormal wear should be replaced so as to ensure proper functioning of the governor system.

Generator

During the overhaul of a turbine generator the most important points to check on the generator are the condition of the insulation, cleanliness of the whole unit, and mechanical condition of the rotating parts.

Insulation. After disassembling the generator the exposed parts of the windings should be inspected for dirt, oil, grease, carbon dust, or other deposits. Dry dirt should be removed with a vacuum cleaner or by carefully blowing out with dry compressed air. Oil or other gummy deposits should be cleaned off with carbon tetrachloride. If the varnish is damaged, the windings should be revarnished with air-drying varnish.

Before reassembling, the insulation resistance of all windings should be measured with a Megger. Readings of insulation resistance should also be taken at regular intervals between overhauls, using the same or a duplicate Megger each time and recording the temperature and atmospheric conditions under which the readings are taken each time. A record should be kept of these readings to note any unexplained sudden drop in resistance or a gradual decrease with time. When the resistance drops to one-tenth or less of the original value when the machine was clean and new, it is time to consider taking the machine out of service for repair.

The insulation of rotating parts should be inspected as far as possible to note any cracks or mechanical breaks in the tape, collars, etc. In the case of round-rotor generators, it may be advisable to remove the retaining rings, or end bells, if there is any suspicion of mechanical breaking of the coil insulation underneath.

It will not be possible to get any direct visual inspection of the slot portions of the rotor coils on round-rotor generators or of the slot portions of armature coils on any machines. Reliance must be placed in the insulation-resistance readings to check the condition of these parts of the generator.

Air Passages. In addition to checking the cleanliness of the windings themselves, the air passages in the generator should be carefully inspected. These include the air ducts in the stator core, air passages in the frame, holes in the centering rings on round-rotor machine, air spaces under the rotor-coil ends, ventilating slots or holes in the rotor core, etc. All such passages should be blown out carefully with compressed air (free from moisture), and oil or gummy deposits wiped off the surfaces with carbon tetrachloride where they can be reached. The same attention should be given to fan blades and rings.

Any unusual amounts of dirt or unusual oil or other deposits should be investigated to see if air seals, oil deflectors, and brushes are properly fitted during operation.

Rotor Parts. Rotors should be examined for tightness of wedges, bolts, and screws. Any surface cracks or flaws should be noted. Risers on d-c armatures should be checked for breakage or other damage such as overheating. Signs of burning at the wedges on round-rotor a-c generators should be looked for as an indication of unbalanced current loading.

Retaining rings and fans should be examined carefully for any signs of cracks or flaws. Retaining rings that fit over wedges or cores should be particularly inspected for signs of localized heating or burning in the tips due to unbalanced circuit loading Retaining rings and fan fits should be checked for tightness as well as for tightness of locking screws, etc.

Collector Rings and Brush Rigging. The active surfaces of collector rings should be inspected, and any evidence of threading or grooving, unusual wear, or unusual deposits or coatings should be noted. These may give an indication of poor brush fitting or operation at wrong brush pressure or with wrong brushes. The diameters of the rings should be checked to make sure the safe wearing depth is not exceeded. The rings should be trued up, cleaned, and polished if necessary.

The collector insulation should be cleaned off, checked for cracks or other damage, and revarnished if necessary. Leads should be inspected for tightness of joints and for cracks or signs of overheating

Brush rigging should be checked for tightness of bolts, studs, connections, etc. The condition of stud insulation, washers, etc , should be examined and these parts replaced if necessary. Dirt and gummy deposits should be removed Brushes should be inspected for fit on rings or commutator, chipped or cracked brushes should be replaced, and the freedom of movement of brushes in holders checked. Brush pressure should be set at the value recommended by the manufacturer.

Exciter

In general the same points should be checked on a d-c generator or direct-connected exciter during overhaul as on a-c generators.

Insulation. The cleanliness, insulation resistance, and mechanical condition of the armature and field windings of an exciter should be checked and repaired in the same way as for an a-c generator. A more thorough visual inspection of the windings can usually be made on d-c machines, and full advantage should be taken of this fact.

Most modern turbine generators are equipped with direct-connected exciters which are totally enclosed and ventilated by air from the main unit. A filter is provided to prevent the return air from the exciter from carrying carbon dust from the brushes into the main generator air circuits. The condition of this filter should be periodically checked, and it should be replaced when the resistance to air flow through it becomes double that of a new clean filter. Any unusual deposits or collections of dirt or oil inside the housing should be investigated. The exciter should be carefully cleaned out after sanding or stoning the commutator or fitting brushes.

Windings and Connections. It should be made certain that all electrical connections are tight. The windings, risers, bolts, etc., should be checked for evidences of overheating or mechanical failures, such as cracks, etc. In particular, it should be made sure that the binding wire on the armature is not disturbed or damaged.

Commutator. The commutator surface should be examined for blackening or rough spots, high mica between bars, amount of wear, etc. Ordinarily, the commutator will require only occasional cleaning off with a piece of canvas or other nonlinting substance. Oil should not be used. In severe cases of burning or roughening, it will be necessary to stone or even to turn down and regrind the commutator.

Brushes and Brush Rigging. The brush rigging and holders should be cleaned off. Defective insulating studs, washers, or yokes should be replaced. Bolts, screws, connections, etc., should be tightened if necessary. Brushes should be checked for fit on the commutator. Chipped, cracked, or worn brushes should be replaced. Free movement of the brushes in the holders should be checked. Position of the brush yoke should be checked to see that it has not shifted from the original position. Brush pressure should be set on individual brushes at the value recommended by the manufacturer.

Accessories

The various protective devices such as relief valves, nonreturn valves, and oil-pressure regulator should be given an operation test before being placed in service, and a record should be made of the test.

The throttle trip valve should be tested for smooth and easy operation. If any sluggishness is noted, the parts should be cleaned thoroughly and the defect remedied. A sticky throttle valve is a hazardous operating condition.

The tubes of the oil cooler and the generator-surface air cooler should be inspected for plugging and leaks. The oil-cooler tube bundle is usually removable and may be cleaned by slushing in a commercial metal-cleaning solution. The manufacturers of such cleansing solutions furnish comprehensive instruction for their use. Normally only the water side of the air cooler requires cleaning, and in the majority of instances, such cleaning can be done with rubber plugs and compressed air or other standard methods used for cleaning condenser tubes. If the air side of the tubes becomes dirty, it is necessary to remove the sections and clean them with water or a solution of washing soda as previously mentioned. An alternative method is to apply through the tube bundle an air blast saturated with carbon tetrachloride.

During the inspection period, it is also desirable to clean and recalibrate all flow meters, pressure gages, and thermometers. Replacements, if necessary, should be obtained to ensure safe operation.

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GAS TURBINES*

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HISTORICAL

The gas turbine has a history traceable to the period preceding the birth of Christ. Hero[1,4] of Alexandria, in 130 B.C., is credited with the first device that could be included in the category of gas turbines. The medieval "smokejack,"[2,5] similar to a windmill, could also be considered a form of gas turbine.

In 1791, Barber[3] of England obtained a patent on a machine which, while of primitive design, contained the essential features of a modern gas turbine. In 1808, Dumbell,[4] also of England, patented a device which was a prototype of the "explosion" type of gas turbine.

The first serious attempt at achieving a practical gas turbine occurred in 1872 when Stolze[6] of Germany built a machine which appeared to be of advanced design. Tests conducted on the unit were unsuccessful, however.

During the years 1903 to 1906, the Société des Turbo-moteurs,[6,7] in Paris, experimented with gas turbines and produced a unit that operated under its own power. While the machine obtained a thermal efficiency of only less than 3 per cent, it was probably the first combustion gas turbine to produce useful output.

In 1908, Karavodine,[5] in Paris, built a small gas turbine which operated on an explosion cycle with open combustion chamber, similar to the German V-1 "buzz bomb" used during the Second World War. The thermal efficiency of the Karavodine machine was very low, although the turbine allegedly operated satisfactorily.

Also in 1908, Holzwarth[8] began the development of his explosion type of gas turbine and is reputedly continuing this work at the present time. Notwithstanding the long period of experimentation, reports infer that there are only several Holzwarth turbines operating in the world today. Stodola has concluded that a thermal efficiency of 13 per cent is the highest obtained up to 1927 on tests of turbines of this type.

Despite a development period extending over two millenniums, the gas turbine did not achieve commercial significance until the third decade of the twentieth century. The main obstacles confronting the early investigators were twofold. (1) The then-existent materials were incapable of withstanding the elevated temperatures necessary to produce suitable cycle efficiencies. (2) The compressors available at the time did not possess adequate efficiencies so as to make the cycle feasible.

These difficulties have been surmounted by recent metallurgical and aerodynamic advances. Utilization of the required high temperatures is accomplished with the superior materials now available, and a highly efficient axial-flow compressor has resulted from progress in fluid research.

OPERATING PRINCIPLES

Practically all present-day gas turbines operate on a cycle wherein combustion occurs essentially at constant pressure instead of at constant volume, as in the case of the Holzwarth turbine. Those operating on the former cycle are referred to as "combustion gas turbines." Of these, the majority are of the "open"-cycle type wherein air is continuously taken in from the atmosphere and exhaust gases from the turbine are discharged back to the atmosphere. There are other gas-turbine cycles in existence which are referred to as "closed" cycles because the working medium is not expelled from the system but rather is continuously recirculated. This type will be discussed under Closed Cycles for Gas Turbines. All subsequent comments will refer to open-cycle units unless specifically designated otherwise.

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* Superior numbers refer to specific references listed at the end of this subsection.

Simplicity is one of the distinguishing features of open-cycle machines. Absence of the many diverse appendages characteristic of previous types has undoubtedly contributed to the commercial success which they have attained. Basic constituents of a combustion-turbine unit are a gas turbine, compressor, combustion chamber, and fuel equipment. Supplementing these with a generator, lubrication system, control apparatus, and a means for starting completes a power plant significant primarily because of its lack of the veritable myriad of auxiliaries necessary for steam-plant operation. Furthermore, since its working medium, air, is universally available and because water is not required in its operation, the basic gas-turbine plant enjoys the added unique advantage of being independent of geographical location.

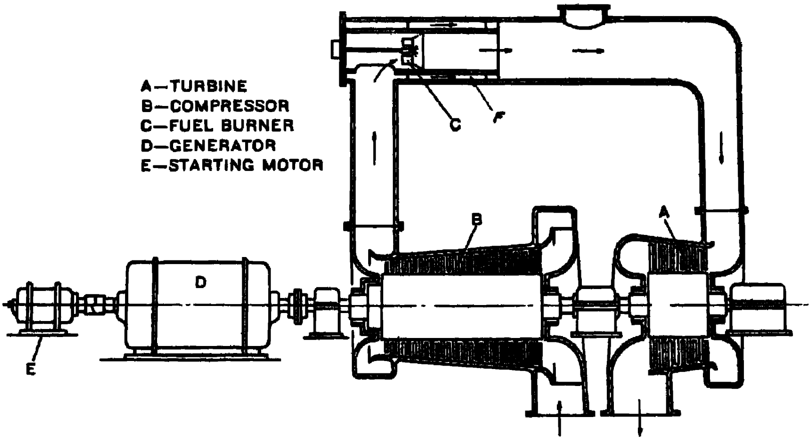


Fig. 8-95. Section through combustion gas-turbine–axial-compressor unit.

The arrangement of a modern combustion gas-turbine unit is depicted in Fig. 8-95. Air from the atmosphere is compressed in a multistage axial-flow compressor *B*, driven by a reaction-type gas turbine *A*. Liquid or gaseous fuel injected at *C* is burned with part of the air discharged from the compressor; the remaining, and greater, part flows through the annular space *F* and, upon emerging, mixes with and cools the products of combustion to a suitable turbine-inlet temperature. Expansion of this gas to atmosphere in the turbine produces more power than that required to effect compression of the air, and the excess power is supplied to a generator *D*. For starting purposes, a motor *E* is provided to bring the unit up to approximately 25 per cent of normal running speed, beyond which the turbine is capable of driving the compressor unassisted.

A speed governor, regulating the fuel supply and thereby the turbine-inlet temperature, affords the necessary control equipment. If a designated overspeed is exceeded, an emergency governor actuates a valve, causing the gas to by-pass the turbine. Balance pistons are not required since the axial thrusts of the turbine and compressor, being equal and opposite in direction, are effectively neutralized through the solid coupling connection. A thrust bearing is provided, however, to absorb any slight axial unbalance that may inadvertently exist.

The axial-flow type of compressor, as shown in Fig. 8-95, is believed to be the most desirable for gas-turbine units, principally because of its higher efficiency. Other types, however, have also been used. Gas turbines have been built with positive-displacement compressors of the Lysholm type. In most instances, centrifugal compressors have been employed in aircraft jet-propulsion gas turbines, although many axial compressors have been used for this service too.

DESIGN FEATURES

Turbine. Since the gas turbine in the prime mover size range has been in successful operation for only a relatively short period, its applications have been necessarily limited and of a special nature.

The principal commercial application in this country has been in oil refineries employing the Houdry process, a catalytic cracking method of manufacturing high-octane gasoline. Air, discharged from the axial compressor after being used in the process to reactivate a catalyst, is returned to the turbine as a high-temperature gas which, upon expansion, produces power to drive the compressor. The excess power is furnished to the generator. In this arrangement no combustion chamber is required since the process itself acts in this capacity after the sequence of operations in the cycle has been established.

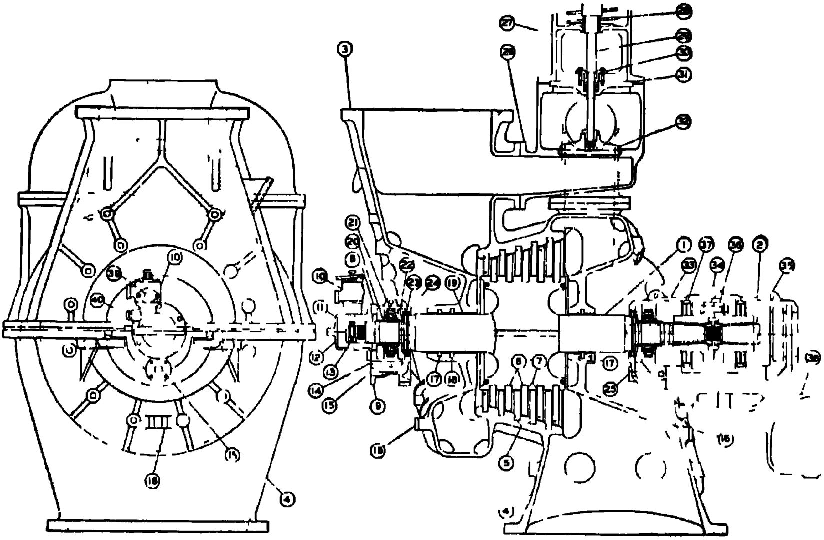


FIG. 8-96. Section through 40,000-cfm gas turbine.

As the pressure drop through the process is appreciable when compared with the maximum-cycle pressure, the net power developed is not so great as that which could be produced if a conventional combustion chamber were employed. The depreciation in useful power can be condoned in this particular instance, however, as the primary function of the gas-turbine unit is the furnishing of compressed air to the process, and any excess power may be considered a by-product. The use of the gas turbine in this application contributes to the economic justification of the Houdry process.

The oil-refinery gas turbines were originally designed and manufactured by Brown-Boveri[9] and Company of Baden, Switzerland, which has done much pioneering development work in the gas-turbine field. Most of the Houdry units, however, have been built by Allis-Chalmers,[10] at first to Brown-Boveri designs under a license arrangement, but later to modified designs after expiration of the license several years ago.

To fulfill the air requirements of various installations embodying the application of the Houdry process, units having compressor capacities of 16,000, 23,000, 40,000, and 60,000 cfm have been constructed. Since the Houdry units are representative, in general, of gas turbines operating on the basic open cycle, one of them will be used for illustrative purposes so that design features can be described. A section through the gas turbine for a 40,000-cfm unit is shown in Fig. 8-96. Neglecting the numerous details, reference will be made only to the major parts.

A solid chromo-nickel-steel forging comprises the turbine spindle 1 on whose periphery are inserted five rows of reaction blading 6, to be described more fully later.

The turbine casing, split on the horizontal center line, is cast of carbon-molybdenum steel. The gas inlet passage 3 and exhaust passage 4, both located in the same vertical plane, are cast integrally with their respective cylinder halves. Five rows of reaction cylinder blading 5 are inserted in grooves machined in the casing. A by-pass connection 26 joins the inlet and exhaust regions through a valve 32 operated by an emergency governor 10 when overspeed conditions exist.

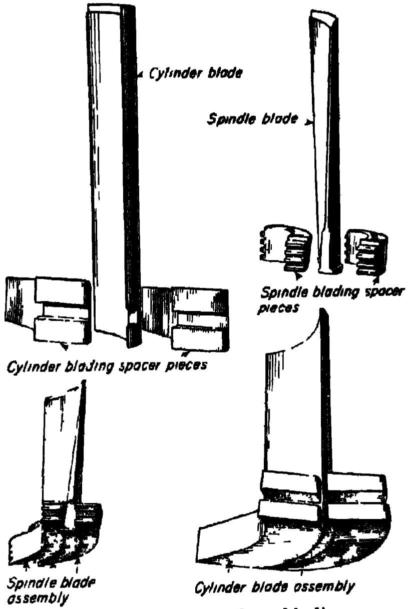


Fig. 8-97. Gas-turbine blading.

Labyrinth glands 19 are provided to prevent leakage of gas to atmosphere where the turbine-spindle ends emerge from the casing. To reduce gas leakage further, sealing air from the compressor discharge is injected at a suitable point 17 in the glands.

Roller bearings 21 are shown, however, these would be of the sleeve type in a machine designed for power purposes only. Some of the later Houdry units also have sleeve bearings. A solid coupling 36 joins the turbine spindle 1 with the rotor 2 of the axial compressor.

The turbine cylinder and spindle blades with spacer pieces are shown in Fig. 8-97. The tapered and twisted spindle blades, milled from 19 per cent chromium–9 per cent nickel steel containing small amounts of tungsten, molybdenum, titanium, and columbium, are securely held in place by the engaging of the upset end of the blade with the serrated spacer pieces. The cylinder blades, made of the same material, are fastened in their grooves by the engaging of a projecting ring in the groove with a slot in the blade and the close-fitting spacer pieces. Later designs have an integral root blade.

Compressor. A sectional view of the axial compressor used on a 40,000-cfm unit is shown in Fig 8-98 The forged-steel rotor 1 consists of a shaft end having a hollow cylinder for the spindle proper on which the rotating blades 6 are mounted and a drum with the other shaft end The hollow cylinder is shrunk onto the drum and locked

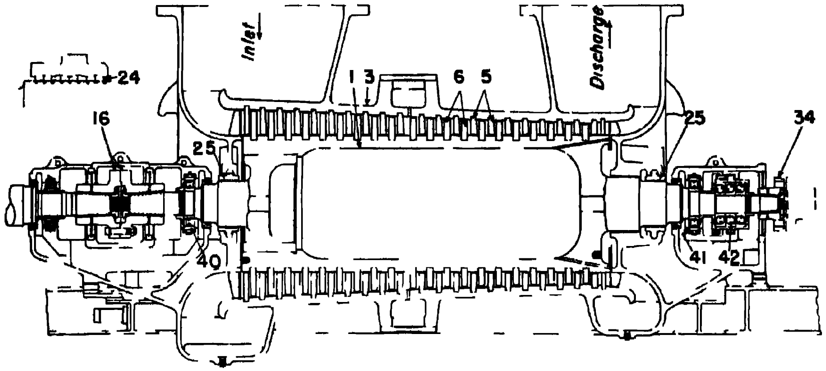


FIG 8-98 Section through 40 000-cfm axial compressor

The cast-iron casing 3 is horizontally split with intake and discharge openings are directed vertically upward and cast integrally with the upper half The stationary blading 5 is inserted in grooves machined in the inner periphery of the cylinder

Where the shaft ends pass through the casing, labyrinth sealing glands 25 are provided These glands consist of radially projecting fins 24 on the shaft which rotate with close radial clearance in grooves in the casing 25 The turbine glands are similarly constructed

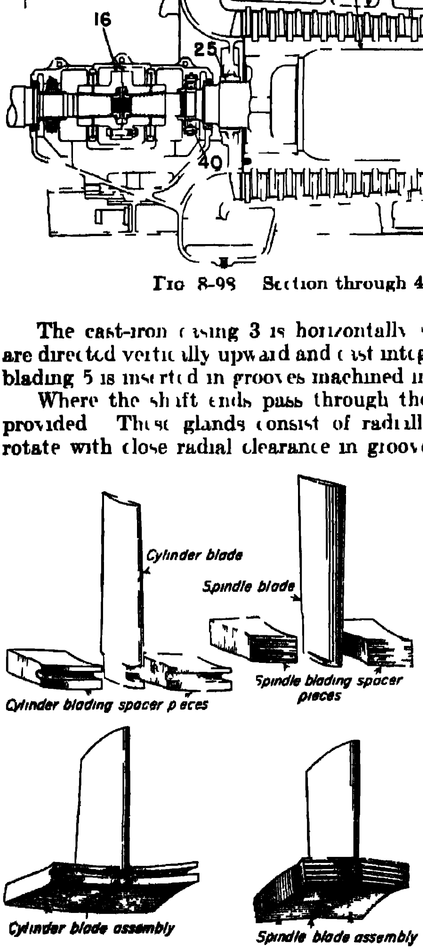


FIG 8-99 Axial-compressor blading

Roller bearings 40 and 41 are shown, but as in the case of the turbine, a power unit would be designed with bearings of the sleeve type A ball thrust bearing 42 absorbs any slight axial thrust that may exist In a power unit this thrust bearing would be of the Kingsbury type

The solid coupling 16 transmits the driving torque of the gas turbine to the compressor The excess power of the unit is supplied to the generator through a gear connected to the coupling end 34

The cylinder and spindle blading used in the axial compressor is shown in Fig 8-99. Since high temperatures are not involved, 5 per cent nickel steel is used as blading material Root fastenings similar to those described for the turbine, are depicted, but later designs have a one-piece blade construction where the root is integral with the flow-passage part of the blade The blades have airfoil cross sections, as aerodynamic requirements dictate such contours in order to produce a high-efficiency axial compressor

Combustion Chamber. When used as a power-producing unit, the gas turbine would have a combustion chamber located between the compressor discharge and the turbine inlet. The chamber is essentially a cylindrical pipe with a fuel-burner element located in the end near the compressor.

After removal from the combustion chamber proper the burner is shown in Fig. 8-100. Louvers, with provision for external adjustment, assure a suitable apportionment of the air discharged from the compressor so that the correct quantity enters the burner for combustion purposes. The remaining air flows through the annular passage defined by the combustion chamber wall (not shown) and the steel liner. Deflectors located in this passage impart a whirling motion to the air, creating a turbulent condition which produces a thorough subsequent mixing with the products of combustion emerging from the central portion of the liner.

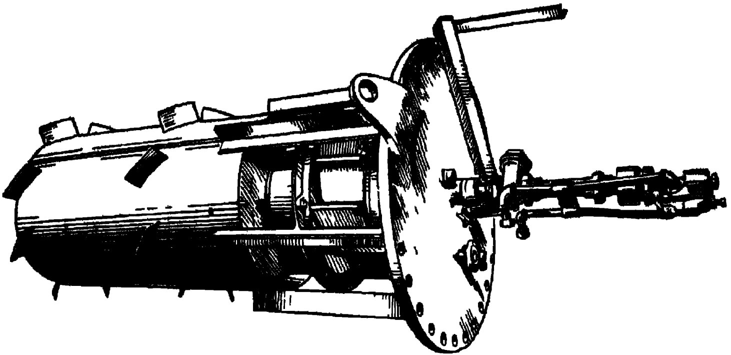


Fig. 8-100. Fuel-burner element.

Assembly. A partially assembled 23,000-cfm gas-turbine axial-compressor unit is shown in Fig. 8-101. Mounted on a common base plate, the six-stage turbine at the left is directly coupled to a 21-stage compressor at the right. This unit, operating at 6,000 rpm, is geared to an 1,800-rpm generator not shown. Lashing wires installed in the first stationary and last two rotating blade rows in the turbine increase the vibrational frequency and reduce the bending stresses in the blades. Roller bearings and labyrinth sealing glands may be clearly seen.

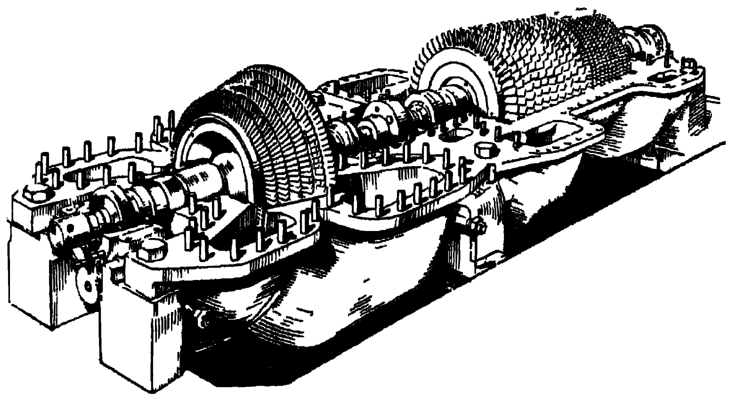


Fig. 8-101. Partially assembled 23,000-cfm gas-turbine–axial-compressor unit.

Except for the installation of the combustion chamber, a completely assembled 40,000-cfm gas-turbine axial compressor unit is shown in Fig. 8-102. This machine

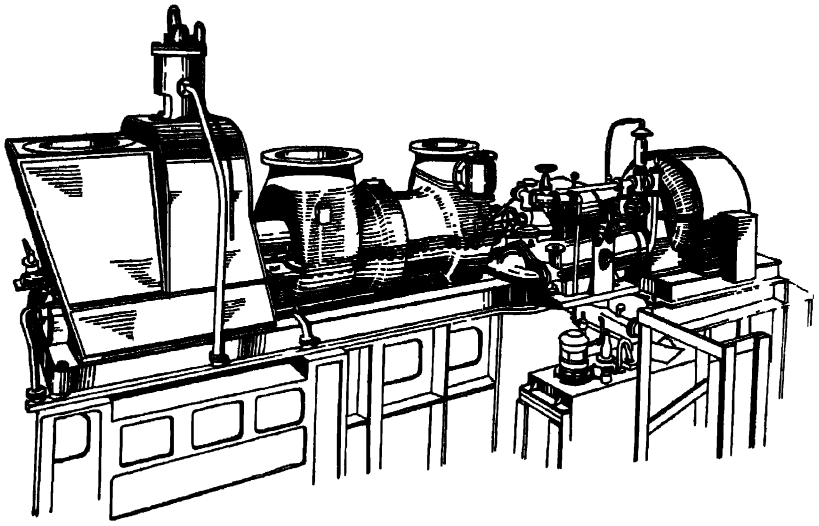


FIG. 8-102. Allis-Chalmers gas-turbine–axial-compressor unit of 40,000 cfm cap

would have a net coupling output of 2,600 hp when operating on 1200°F gas at the turbine inlet.

PERFORMANCE CHARACTERISTICS

Basic Cycle (Cycle A). The performance of a gas-turbine unit depends, among other factors, upon the type of cycle adopted. The basic cycle shown in Fig 8-95 is reproduced schematically in Fig. 8-103 and designated as cycle A. The thermal efficiency of this cycle is given in Fig 8-104 as a function of pressure ratio for various turbine-inlet gas temperatures from 800 to 1600°F. The performance data presented in Fig 8-104 and subsequent Figs. 8-106, 8-108, 8-110, and 8-112 are based on a turbine efficiency of 86 per cent, a compressor efficiency of 84 per cent, a combustion efficiency of 98 per cent, a compressor intake temperature of 60°F, and mechanical losses of 0.5 per cent in the turbine and also in the compressor.

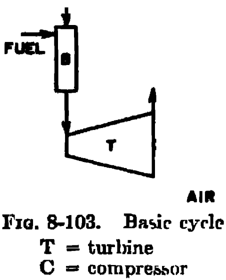


FIG. 8-103. Basic cycle (cycle A).

T = turbine

C = compressor

B = combustion chamber

In those cycles employing regeneration, the heat-exchanger surface is 5 sq ft per coupling hp, and the over-all heat-transfer coefficient from gas to air is 10 Btu/(hr) (sq ft)(deg F). Where intercooling is used, the surface of the intercooler is 1.5 sq ft per coupling hp, and an over-all water-to-air coefficient of heat transfer of 20 Btu/(hr) (sq ft)(deg F) is taken. The performance calculations for all the cycles account for the pressure drops occurring in the various elements.

In all cases the thermal efficiency is defined as the ratio of the useful output at the coupling to the energy required from the fuel, each expressed in appropriate units

Referring to Fig. 8-104 it can be seen that the thermal efficiency improves with increasing gas temperature at the turbine inlet. It will also be noted that the higher temperatures require an increase in cycle pressure ratio in order to achieve maximum thermal efficiency As will be observed later for all cycles, the efficiencies are enhanced by the use of elevated temperatures, and, similarly, such temperatures necessitate higher pressure ratios for optimum conditions to exist.

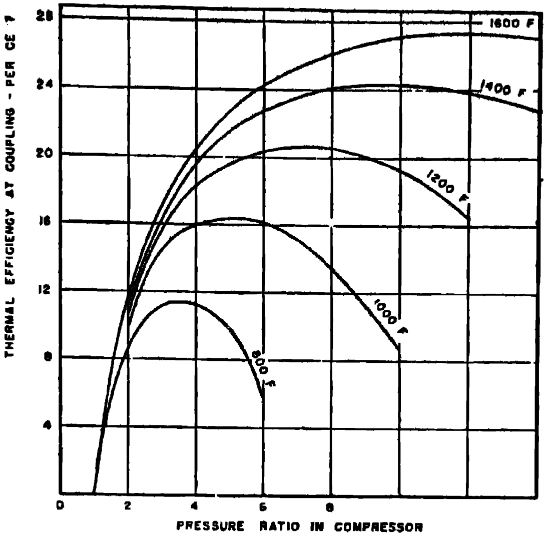


FIG. 8-104. Thermal efficiency of basic cycle.

Regenerative Cycle (Cycle B). In order to improve the efficiency, cycle A can be modified from the basic form into a regenerative type by the addition of a counterflow heat exchanger as shown by cycle B in Fig. 8-105. With such an arrangement the turbine exhaust gas is used to preheat the compressor-discharge air before entry of the latter into the combustion chamber. Figure 8-106 shows the efficiency of this kind of cycle, and the improvement effected by the heat exchanger is readily apparent. At 1200°F, for example, the maximum thermal efficiency is 28.4 per cent, representing an increase of 38 per cent over the corresponding efficiency of cycle A at the same temperature. It will be noted that, in addition to increasing efficiency, the heat exchanger also has the effect of reducing the value of the pressure ratio for the point of maximum efficiency. Regenerative heating by this method influences all cycles similarly.

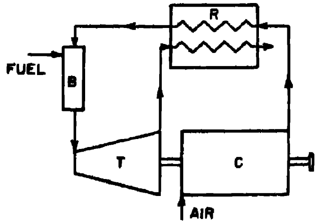


FIG. 8-105. Regenerative cycle (cycle B).

T = turbine
C = compressor
B = combustion chamber
R = regenerator

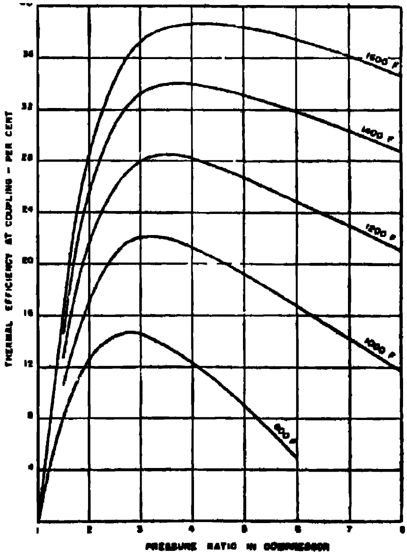


FIG. 8-106. Thermal efficiency of regenerative cycle.

Regenerative Cycle with Reheating (Cycle C). A still higher efficiency cycle can be obtained by incorporating a reheat turbine, as shown by cycle C in Fig. 8-107. In this system, instead of the expansion occurring uninterruptedly in the turbine as in cycle B, the motive fluid is withdrawn at an intermediate point and heated externally until its temperature corresponds to the value at the turbine inlet. It is then reintroduced into the turbine where it expands to the exhaust pressure. For optimum conditions, the reheat point should be selected so that the work done by the portions of the turbine before and after the reheat belt is approximately equal. A comparison of Figs. 8-106 and 8-108 will show the benefit of reheating. It can be seen that reheating has an effect

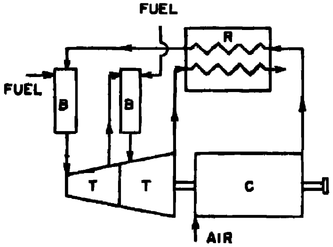


FIG. 8-107. Regenerative cycle with reheating (cycle C).

T = turbine
C = compressor
B = combustion chamber
R = regenerator

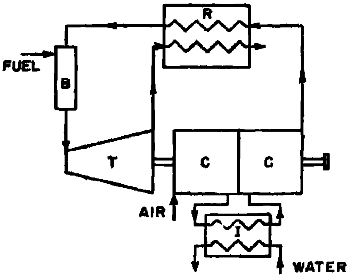


FIG. 8-109. Regenerative cycle with intercooling (cycle D).

T = turbine
C = compressor
B = combustion chamber
R = regenerator
I = intercooler

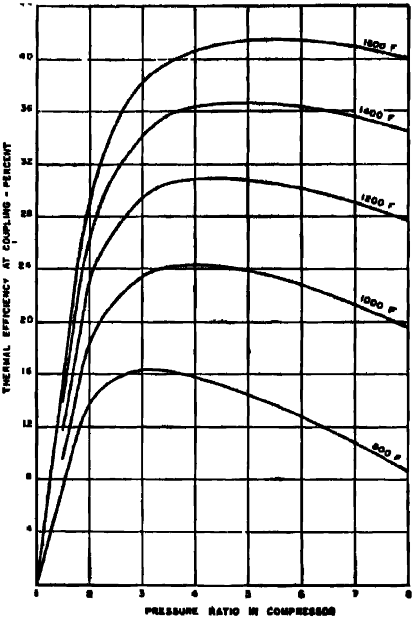


FIG. 8-108. Thermal efficiency of regenerative cycle with reheating.

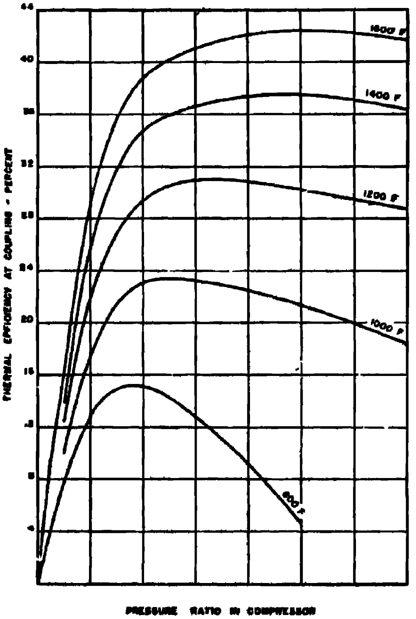


FIG. 8-110. Thermal efficiency of regenerative cycle with intercooling.

counter to that of regeneration in that optimum results occur at higher pressure ratios.

Regenerative Cycle with Intercooling (Cycle D). Another method of augmenting the performance of cycle B is to incorporate an intercooler in the compression part of the cycle and thereby reduce the power required by the compressor. Thus the air from the atmosphere is partially compressed and then cooled indirectly by water in a counterflow heat exchanger, after which it undergoes its final stage of compression. This combination is shown as cycle D in Fig. 8-109. The performance characteristics of cycle D are given in Fig. 8-110. As in the case of reheating, when compared with cycle B, the introduction of intercooling increases the pressure ratio associated with maximum efficiency conditions.

Regenerative Cycle with Reheating and Intercooling (Cycle E). When all the aforementioned elements are used in a gas-turbine cycle, best results in the way of thermal efficiency are obtained. Cycle E, employing such a combination, is depicted in Fig. 8-111 The thermal efficiencies of the cycle at various temperatures and pressure ratios are given in Fig. 8-112. The added refinements of reheating and inter-

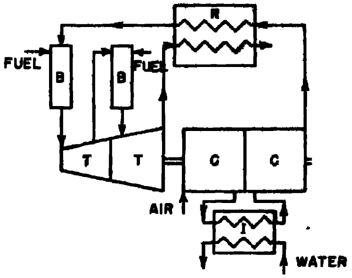


FIG. 8-111. Regenerative cycle with reheating and intercooling (cycle E).

T = turbine
C = compressor
B = combustion chamber
R = regenerator
I = intercooler

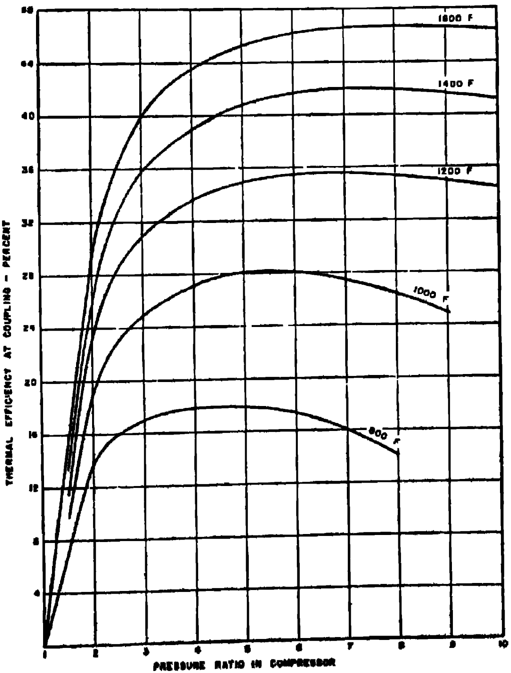


FIG. 8-112. Thermal efficiency of regenerative cycle with reheating and intercooling.

cooling have, as in the case of cycles C and D, moved the point of maximum efficiency to ever higher pressure ratios. Flatter efficiency characteristics have also resulted.

Comparison of Cycles. The maximum efficiencies have been plotted in Fig. 8-113 as a function of temperature for all the aforementioned cycles so that a comparison can be readily made. The increase in thermal efficiency with higher turbine-inlet temperatures is readily apparent. Because of such beneficial results obtained with the use of elevated temperatures, there is a real incentive for metallurgical advances so as to develop superior alloys to operate in the upper regions of temperature.

While all cycles improve with increased turbine-inlet temperature, it will be noted by comparing cycles A and B that the single modification showing the greatest relative gain is the addition of the heat exchanger to the basic cycle. From this observation it can be concluded that the most obvious manner in which to improve fuel consumption is to employ a heat exchanger, as it is a comparatively inexpensive piece of equipment with relatively low maintenance.

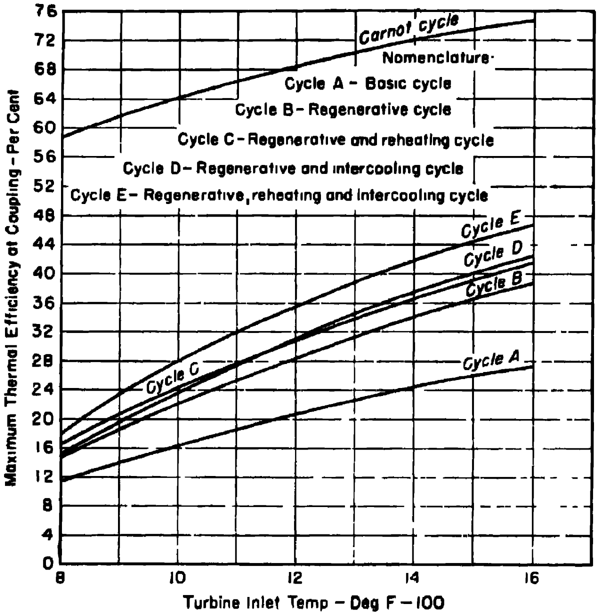


FIG. 8-113. Maximum thermal efficiency of various cycles.

Reference to Fig. 8-113 will reveal that the rate of improvement occasioned by higher temperatures increases as additional elements are embodied in the cycle. Thus a given increment in temperature benefits the more complex cycles to a greater extent than the others. For the conditions selected, reheating and intercooling both show about the same average gain when applied individually to the regenerative cycle.

The Carnot efficiency, representing the maximum possible attainment of any cycle between given temperature limits, is also shown on Fig. 8-113. Notwithstanding all the refinements incorporated in cycle E, it is still lacking to a considerable degree when compared with the ideal reversible Carnot cycle at a given temperature In the limiting case, however, with infinite surface in the regenerator and an infinite number of reheating and intercooling points, cycle E will then be theoretically reversible and will have the same efficiency as the Carnot cycle between any given temperature levels. The magnitude of the difference between the actual cycle E and the

Carnot is a measure of the departure of the performance of the various cycle components from ideal conditions.

In the foregoing cycles, it was intended to show the effect on performance of a change in the cycle by adding a single element at a time, starting with the basic cycle. In each cycle where it is used, the regenerator has the same surface for a given output and the same coefficient of heat transmission. This also applies to the intercooler. Thus the various cycles are directly comparable when determining the effect of the respective elements since corresponding components are based on the same conditions.

As a result of this premise, therefore, the regenerator and intercooler effectivenesses are different for each cycle and temperature. Effectiveness is defined as the ratio of the change in temperature of the heated or cooled air and the difference between the highest and lowest temperatures occurring in the heat-exchange process. As an illustration, the effectiveness of the regenerator in cycle B with a turbine-inlet temperature of 800°F is 77 per cent, while in cycle E at 1600°F, it is 90 per cent. Similarly, the intercooler effectiveness of cycle E at 1600°F is higher than for cycle D at 800°F.

In all cases it is the same size regenerator or intercooler, but in the more complex cycles the effectiveness is higher because of the reduced quantity of motive fluid being handled per unit output. Consequently, a smaller and less expensive heat-exchange element may be used in one of the more elaborate cycles than in the simpler ones for the same efficiency.

This procedure of using the same heat-exchange elements in different cycles is the most efficacious manner in which the effect of the cycle itself can be demonstrated. If, in addition to altering the cycle, the same effectiveness of the heat-exchange apparatus had been retained instead, then the datum points would be incompatible, and any comparisons would be rendered illusory.

While the foregoing discussion utilized five different cycles for illustrative purposes, it should be understood that a variety of other cycle combinations is possible, and each may have certain distinct advantages. Regardless of the type of open cycle adopted, however, it will employ several or all of the elements contained in the cycles just discussed Furthermore, in a given cycle one or more units of the same element may be used in order to accomplish a particularly desired effect. Several other open-cycle arrangements will be discussed under Gas-turbine Applications

Specific Fuel Consumption. Figure 8-114 shows the minimum specific fuel consumption, in pounds per brake horsepower-hour, of cycles A to E, inclusive, and of the Carnot cycle when oil is used as a fuel. It was obtained by converting the thermal-efficiency curves of Fig. 8-103 to a specific fuel consumption basis, assuming the fuel oil to have a lower heating value of 17,800 Btu per lb. Reference to Fig. 8-114 will reveal the obvious improvement in performance resulting from the adoption of elevated temperatures. Above 1200°F the specific fuel consumption of the more elaborate cycles is in the range of the diesel engine, while the fuel cost will be considerably less because of the cheaper grade of fuel which the gas turbine is capable of utilizing.

Since in open-cycle gas turbines the products of combustion pass through the turbine blading, such machines have usually been considered as suitable primarily for employing only liquid fuels so as to reduce the solid particles in the motive fluid to a minimum. However, the Locomotive Development Committee,[11] under Bituminous Coal Research, Inc., is at present conducting tests whose promising results indicate the practicability of using pulverized bituminous coal as a gas-turbine fuel. This application will be discussed in more detail under Gas-turbine Applications.

With coal fuel, Fig. 8-115 gives for cycles A to F, inclusive, and the Carnot cycle the minimum specific fuel consumption as a function of turbine-inlet temperature. A bituminous coal having a lower heating value of 12,500 Btu per lb has been selected. Above 1200°F the fuel rate for the regenerative cycles is comparable with that obtainable in modern central steam stations.

Fuel Costs. Also in Fig. 8-114 is given the fuel cost, in cents per brake horsepower-hour, for a fuel oil having an API gravity of 16, a cost of 4 cents per gal, and a density of 8 lb per gal. In the higher temperature regions the cycles employing regeneration and other means for increasing efficiency are attractive on a fuel-cost basis, especially when compared with other prime movers using liquid fuels.

Figure 8-115 shows, in addition, the specific fuel cost for the various gas-turbine cycles when using coal costing $5 per ton. The desirability of using solid fuels is manifested by the low cost of generating power, as indicated in Fig. 8-115. In the upper temperature ranges the fuel economy is better than that of present-day steam stations using fuel of equivalent price.

Optimum Pressure Ratio. Although the pressure ratio at which maximum efficiency occurs in a given cycle is not precisely a linear function of temperature, it may be approximated as such, as shown in Fig. 8-116. It can be seen that the greatest variation in optimum pressure ratio with temperature occurs in the basic cycle Thus, to build such a cycle for maximum performance at elevated temperatures would necessitate many stages in the turbine and compressor. By adopting regeneration (cycle B), these elements can be simplified, because of the lower pressure ratio required, and higher efficiency can be achieved at a given temperature. Except in some special applications which may justify it, it is unlikely that the basic cycle would be constructed for peak performance at high temperatures.

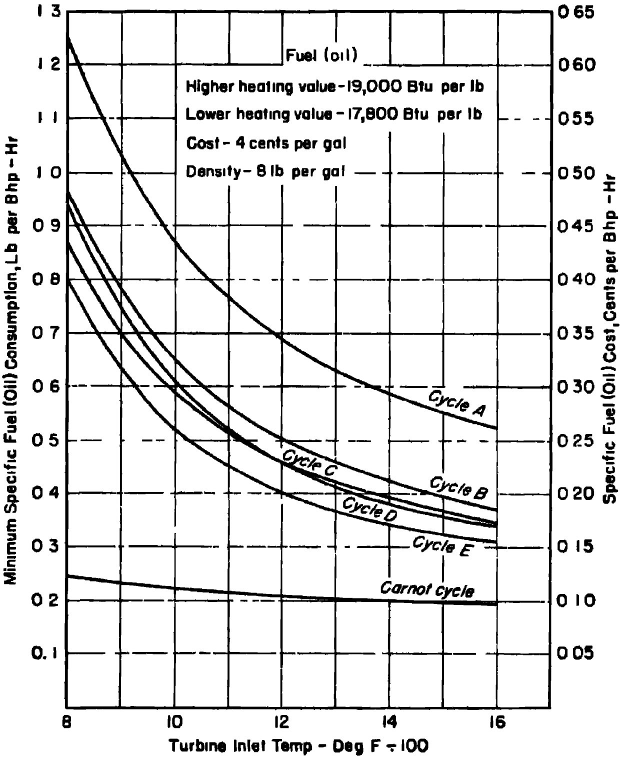


FIG. 8-114. Minimum specific fuel consumption and cost for various cycles using oil fuel

Reference to Fig. 8-116 reveals that the adoption of reheating or intercooling, individually or in combination, necessitates higher pressure ratios for the attainment of optimum conditions. Above a turbine-inlet temperature of 1050°F, however, even with cycle E, these pressure ratios are still less than the basic cycle requires.

Effect of Ambient Temperature. As mentioned previously, the foregoing performance data are based on the air entering the compressor having a temperature of 60°F. Any variation in this temperature will be reflected in a change in performance. In general, the thermal efficiency will decrease about one percentage point in efficiency for each 15° increase in inlet air temperature.

Performance at Partial Load. In Figs. 8-104, 8-106, 8-108, 8-110, and 8-112 the maximum thermal efficiency is depicted as a function of pressure ratio for various turbine-inlet temperatures. Usually the maximum efficiency will occur at or near the full-load point of the unit. Operation at partial loads is accompanied by reduced thermal efficiency, and the degree of reduction depends upon the cycle arrangement and the mode of operation adopted.

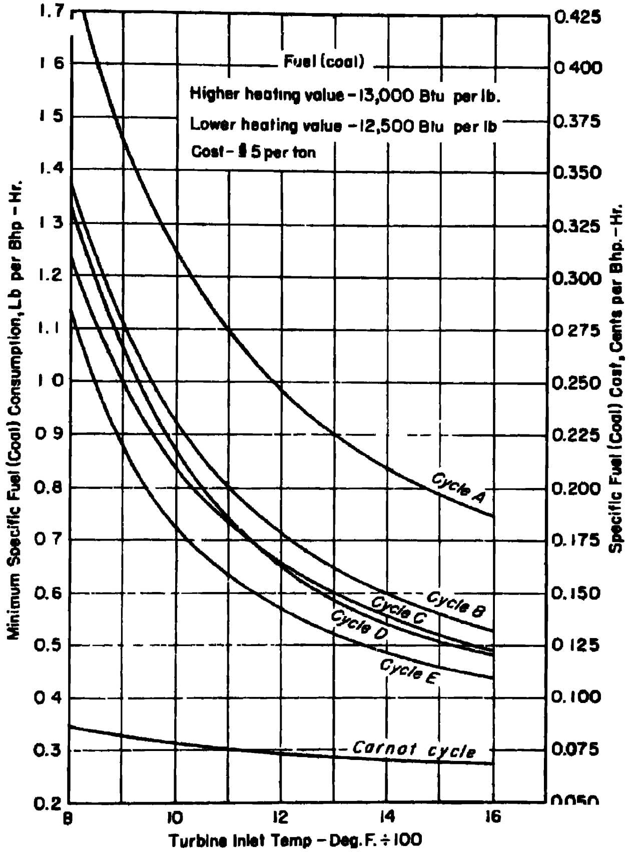


Fig. 8-115. Minimum specific fuel consumption and cost for various cycles using coal fuel.

In general, maximum thermal efficiency at partial load can be achieved by maintaining the turbine-inlet temperature at a high value. Optimum conditions at reduced load ensue when the temperature is kept constant at the design value for all loads. In order to accomplish this in open cycles at reduced output, it is necessary to diminish the flow of the working substance, which requires variable compressor speed. Where the nature of the load is such that constant rotative speed is not necessary, the variable-flow requirement can be fulfilled even in cycles comprising only a single shaft with one turbine and one compressor, thereby making possible the

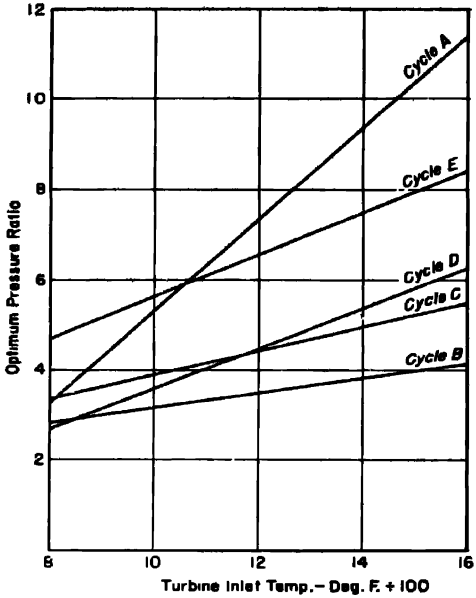


FIG. 8-116. Optimum pressure ratios for various cycles.

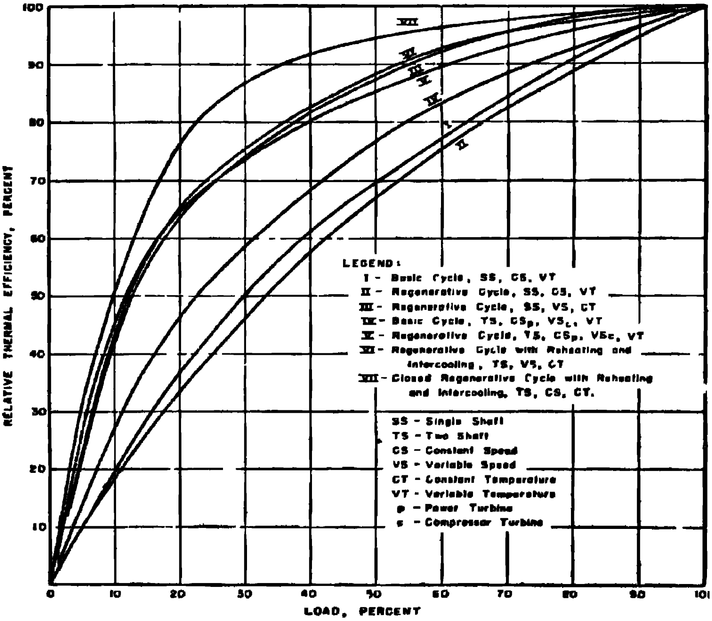


FIG. 8-117. Thermal efficiency at partial load for various cycles.

attainment of good partial-load efficiencies. If the load is of a fixed-speed type, such as an a-c generator operating at constant frequency, in order to achieve variable flow a multiple-turbine arrangement is necessary where one turbine drives the generator and the other the compressor.

The relative thermal efficiencies of various gas-turbine cycles are given in Fig. 8-117 as a function of load. It can be seen that the flattest characteristics are obtained in cycles III, VI,[12] and VII,[14] where the turbine-inlet temperature is maintained essentially constant at the normal design value at all loads. In order to accomplish this, it will be noted from the legend that variable-speed operation is employed in the open cycles. The greatest variation in thermal efficiency is experienced in cycles I and II, where the single-shaft, constant-speed arrangement necessitates the adoption of reduced temperatures at light loads with resulting lower efficiency under these conditions.

The partial-load characteristics of the basic cycle I can be improved by using two shafts, as indicated by cycle IV. In the latter arrangement the variable-flow feature permits higher temperatures to be used in the light-load region. It will be noted that the legend signifies variable-temperature operation for cycle IV. It is so designated since there is a slight decrease in temperature with diminishing output, but the temperature will still be maintained at higher values than in the case of the single-shaft arrangement of cycle I. A similar demonstration of the beneficial effect of multiple shafts can be obtained by comparing the regenerative cycles II and V. The sustained higher temperatures at reduced load, possible with cycle V, yield a flatter efficiency characteristic.

From Fig. 8-117 it can be seen that the closed cycle VII possesses the most desirable light-load performance. This is one of the advantages of closed systems and is produced by a type of control whereby the quantity of the motive fluid circulating through the circuit is varied. Thus, the temperatures, velocities, pressure ratios, and directions of flow remain essentially identical at all loads even though the pressure level of the cycle varies. The component parts of the cycle are, therefore, operating approximately at their normal design conditions under all outputs so that the efficiency is maintained at a high value even at light loads.

CLOSED CYCLES FOR GAS TURBINES

The majority of gas turbines that have been built operate on an open cycle of the type depicted by cycles A to E, inclusive. In contrast to this kind of cycle, however, there is another type referred to as a closed cycle where the motive fluid is confined to a closed system and is not discharged to the atmosphere at any point, as in the open cycle.

Escher-Wyss Closed Cycle. A closed-cycle experimental gas turbine[14] of 2,000-kw generator output which has been proposed and built by the Escher-Wyss Engineering Works, Ltd., of Zurich, Switzerland, is shown in Fig. 8-118. Air is used as the working fluid in the closed circuit of this unit. Starting at the intake, the air traverses the axial-flow compressor and two intercoolers, after which it leaves the compressor and passes through a counterflow heat exchanger where its temperature is raised by the turbine exhaust air. The heated air leaving the high-pressure section of the regenerator enters an indirect-fired combustion chamber where its temperature is increased to the desired turbine-inlet value. After expanding through a high-pressure turbine, which drives the compressor, it then passes through a low-pressure turbine furnishing power to an electric generator. The exhaust air from this turbine has its temperature reduced in the regenerator from which it enters a precooler where its temperature is further diminished indirectly by cooling water. The cooled air then enters the compressor, and the cycle is resumed. The use of the precooler is necessary so as to prevent prohibitive compressor power requirements.

Control of Closed Cycle. In a closed cycle the output may be varied by changing the quantity of fluid circulating through the system. With a decrease in load, the speed-responsive governor actuates valves which permit some of the motive fluid to be withdrawn from the cycle and stored in a low-pressure accumulator. When the load increases, fluid is readmitted to the system from a high-pressure accumulator.

A withdrawal of less than 20 per cent of the working fluid is sufficient to vary the output from full load to no load. A separate compressor set is used to charge the high-pressure accumulator to compensate for fluid withdrawn during governing. Fuel supply to the system is altered so as to maintain essentially constant motive-fluid temperatures at all loads.

Advantages of Closed Cycle. Several points of advantage are claimed for the closed cycle over the open type. The fact that the system is closed means that it is possible to use fluids other than air in the cycle. For instance, if a monatomic gas, with its higher isentropic exponent, were used instead of air, optimum performance could be obtained at a lower pressure ratio. When lighter gases are used, the output for given plant dimensions is increased. With inert gases, the tendency toward oxidation of metallic parts would be reduced.

Since the system is closed, the working fluid may be kept clean as there are no combustion products circulating through the internal circuit. Hence, the possibility of deposits occurring on blading and heat-exchanger surfaces is reduced to a minimum so that high turbine efficiency and heat-exchanger effectiveness may be maintained.

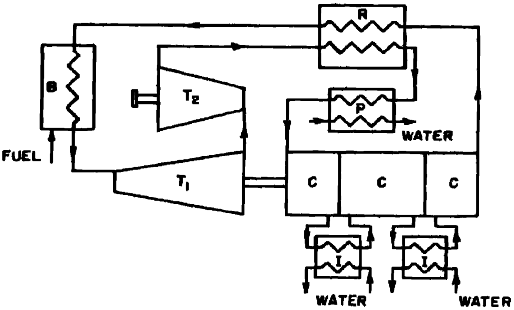


FIG. 8-118. Escher-Wyss closed cycle.

T_1 = compressor turbine
T_2 = power turbine
C = compressor
B = combustion chamber
R = regenerator
P = precooler
I = intercooler

The optimum pressure ratio of a closed-cycle gas turbine will be about the same as that of an essentially equivalent open-cycle unit, but its pressures at corresponding points in the circuit may be much higher. In the Escher-Wyss cycle shown in Fig. 8-118, for instance, the pressure at the compressor discharge at approximately full load is given as 342 psia when the intake pressure is 85 psia. The pressure ratio of 4.04 is of the order of that which would exist in a similar open cycle, but because of the much higher pressures themselves the densities of the motive fluid in the closed system are increased considerably.

Because of the greater density the dimensions of the equipment for a given output are correspondingly reduced from those of an open cycle. This results in small, lightweight, high-speed, lower cost machinery. With the closed cycle it appears feasible to build gas-turbine units having a capacity up to 25,000 kw and with space requirements as low as 3.5 cu ft per kw. The low-pressure levels existing in open cycles result in large specific volumes and concomitantly increased dimensions which limit the maximum capacity of such cycles to approximately 10,000 kw at the present time.

In addition to reducing dimensions, the higher density greatly increases the heat-transmission coefficients in the cycle. Therefore, the combination of these two effects, coupled with the improved cleanliness factor, results in much smaller heat-exchanger equipment.

One of the most important advantages of the closed cycle, where control is by variation of the quantity of motive fluid in the system, is that the efficiency at partial loads is maintained at a high value. For example, tests on the Escher-Wyss unit showed an over-all thermal efficiency of 25.5 per cent at one-quarter load compared to 31.4 per cent at full load. These efficiencies are based on the lower heating value of the fuel, in accordance with standard European practice, and with very cold circulating water, which existed at the time the tests were performed, which would tend to increase efficiency.

The flat efficiency characteristic is obtained because, with control by quantity variation, even though the pressure level in the system is changed, the temperatures, pressure ratios, velocities, and blading angles of attack remain practically the same at all loads. Therefore, the respective cycle elements are operating at or near their design conditions approximately continuously.

With a closed cycle, because of the indirect type of heater employed, it is possible to use either solid, liquid, or gaseous fuel since the products of combustion do not pass through the other elements in the cycle.

The closed cycle would appear to be particularly suitable for marine applications, primarily because of the large quantities of water available for cooling purposes but also because of the reduced size of external ducting since the only outside air requirements are those for burning the fuel and not for supplying the entire cycle motive fluid, as in an open system. Accordingly, the intake and exhaust piping could be much smaller.

Disadvantages of Closed Cycle. Counteracting some of the previously enumerated advantages, the closed cycle also has certain undesirable features not found in gas turbines of the open type. The precooler, a characteristic element of the closed cycle, is not required in an open system. It is a relatively large piece of equipment since it must reduce the temperature of the working fluid entering the compressor to a reasonably low value in order to minimize the negative work of the cycle.

The indirect heater that is required is also of appreciable size. For a pulverized-coal installation, it would be approximately of the same proportions as a steam-generating unit for a steam plant of equivalent output. It is possible to reduce the size, however, by resorting to a pressurized construction. As a compensating feature, the heater does lend itself to outdoor installation which tends to mitigate the objection of excessive size. A further disadvantage of the heater is that it operates indirectly and, therefore, less efficiently than a combustion chamber in an open cycle.

Although some of the principal advantages of the closed cycle are realized because of the high-pressure levels at which it functions, this particular feature increases the difficulty of preventing leakage. More elaborate designs of glands are required than in an open system which operates at lower pressures. In cycles where motive fluids other than air are employed, the importance of minimizing leakage is manifest.

An attractive feature of the open cycle is that practical and efficient designs can be built without any water requirements. Such is not the case for a closed circuit as relatively large quantities of water are needed in the precooler. This limits the range of applications of closed-cycle gas turbines as it would not be feasible to use them in locomotives or aircraft or in localities where water is not readily available. If air instead of water were used as a coolant, the space requirements of the heat exchangers would be appreciable and might be prohibitive for a practical installation.

Westinghouse Modified Closed Cycle. The Westinghouse Electric Corporation in this country has under development a modified closed cycle[14] which eliminates the conventional indirect heater, as shown in Fig. 8-119. In this arrangement a direct-fired combustion chamber of smaller size and higher efficiency, similar to that of an open cycle, is used. Since the system is internally fired, the oxygen required for combustion must be continuously replenished. This is accomplished by an auxiliary turbine compressor which supplies fresh air to the main cycle at the upper pressure level in sufficient quantity to furnish the requisite oxygen. The auxiliary turbine is driven by gas having the same initial conditions as the main turbine, but its pressure ratio and that of the auxiliary compressor are much higher than that which exists in the main circuit.

While the cycle depicted in Fig. 8-119 replaces the indirect heater with a smaller and more efficient type, it still requires the precooler necessitating large quantities of water for its operation. Since it passes the products of combustion through the internal parts of the main system, subjecting them to any contamination that may be present, it is similar in this respect to the open cycle. It also requires an additional turbine and compressor for its proper functioning, and there would be no choice of working fluids, as in the case of the conventional closed cycle, since the use of air is mandatory.

Because of the higher efficiency of the direct-fired combustion chamber, the arrangement shown in Fig. 8-119 should give better optimum performance than a conventional closed system, other things being equal. However, any such advantage may diminish with time if there is deposition of foreign material on the blading and heat-exchanger surfaces.

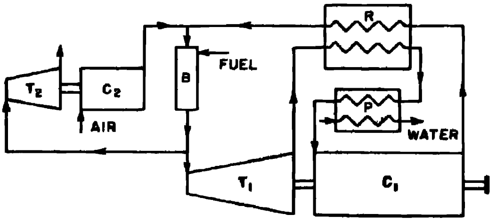


FIG. 8-119. Westinghouse modified closed cycle.

T_1 = main turbine
T_2 = auxiliary turbine
C_1 = main compressor
C_2 = auxiliary compressor

B = combustion chamber
R = regenerator
P = precooler

If the cycle is governed basically by fuel supply, as in open systems, the partial-load efficiency would be less than that for the cycle of Fig. 8-118 where quantity control of the motive fluid is employed.

Sulzer Bros. Modified Closed Cycle. Another modification of the closed cycle, which appears to have certain advantages, is being developed by Sulzer Bros. of Winterhur, Switzerland.[15] This cycle, shown in Fig. 8-120, retains the indirect heater and, consequently, the purity of the motive fluid in the main cycle, but it also utilizes the products of combustion by expanding them in a second turbine which furnishes the useful output. The combustion air is obtained from the primary system and is restored by an auxiliary compressor driven from the main shaft. Unlike the conventional closed system, the motive fluid is restricted to air in the main circuit and combustion products in the power turbine.

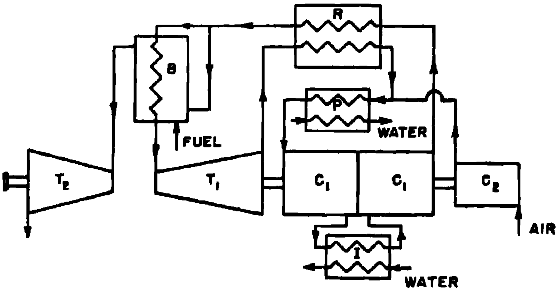


FIG. 8-120. Sulzer Bros. modified closed cycle.

T_1 = compressor turbine
T_2 = power turbine
C_1 = main compressor
C_2 = auxiliary compressor

B = combustion chamber
R = regenerator
P = precooler
I = intercooler

This arrangement combines some of the advantages of both of the cycles shown in Figs. 8-118 and 8-119, although it still requires a precooler with its accompanying water supply. As in the case of the cycle of Fig. 8-119, the optimum efficiency would probably be higher than for a conventional closed system composed of otherwise corresponding elements. Except for the power turbine, however, it would not be subject to decreased performance due to flue-gas contamination.

The comments presented previously regarding the effect of the method of governing on partial-load efficiency of the cycle of Fig. 8-119 also apply, in general, to that of Fig. 8-120. In the latter cycle, however, the cleanliness factor will be higher, but any gain in this direction is counteracted by the lower efficiency of the auxiliary compressor and power turbine because of their operating at variance with design conditions when under partial load. This is a more influencing factor than the auxiliary turbine of Fig. 8-119 because the power turbine involves more energy and, therefore, will affect cycle performance to a greater extent.

OPERATING PROCEDURE

The exact gas-turbine operating procedure in a given installation will depend upon the design and size of the particular equipment involved. For instance, the lightweight, high-speed, and relatively small aircraft jet-propulsion gas turbines can be started from cold, in a push-button manner, in less than 60 sec, primarily because of their simplified and symmetrical design. For more elaborate plants, such as the stationary type, however, where complex piping systems and other influencing factors are involved, the starting and loading interval must be of somewhat longer duration. In general, the following procedure will be applicable for most gas-turbine installations. In particular instances, after a definite starting routine has been established, it may be decided to omit some of the steps suggested below.

Starting.

1. Start auxiliary oil pump If separate relay and lubricating pumps are provided, these may be started individually.
2. Check oil pressures and oil-sight feeds. This also applies to starting device, whether motor, steam turbine, or other type.
3. Arrange all valves in system so that minimum compressor-discharge pressure will exist when maximum flow is passing through turbine that drives compressor. This is particularly necessary where axial-flow compressors are involved so as to prevent the occurrence of unstable conditions.
4. Test gas-turbine overspeed device by tripping by hand, and then reset.
5. If starting device is steam turbine, the following additional procedure should be carried out
   - *a*. Test steam-turbine overspeed device by tripping by hand, and then reset.
   - *b*. Admit cooling water to steam-turbine bearings if oil-ring type.
   - *c*. Open all steam-turbine drain valves.
   - *d*. Open steam-turbine exhaust valve.
6. Engage starting device and bring compressor unit up to about 35 per cent of normal rotative speed. If steam turbine is used, a lower speed than this should first be attained by cracking throttle valve and checking immediately to see if oil rings are revolving. Close drain valves when water is out of steam system.
7. Admit water to oil cooler when temperature of return oil from main bearings reaches 125°F. Regulate flow with discharge valve so as to maintain temperature between 125 to 130°F.
8. Check bearings for sound and temperature. Check glands for sound. Open valves to gas-turbine gland seals. Check sound of rotors for rubbing.
9. Close fuel-oil shutoff valve to burner and start fuel pump.
10. Adjust burner fuel control so that flow will be about 10 per cent of normal.
11. Turn on ignition device.
12. Open shutoff valve to burner. Flame should develop immediately in combustion chamber, resulting in a temperature rise of several hundred degrees Fahrenheit in the turbine-inlet gas temperature.
13. Turn off ignition device.

14. **Increase fuel flow until speed of unit is self-supporting.** The rate of increase in fuel flow will depend somewhat upon the design of the equipment, but for a unit operating on the basic cycle, the temperature may be increased approximately at the rate of 100°F per min up to the self-supporting point. For more complex cycles involving several turbines and more elaborate piping, it may be desirable to establish a reduced heating rate in conformity with the particular installation concerned.

15. Disengage and shut down starting device. If steam-turbine type, close throttle and open drains.

16. If load-absorbing means is a-c generator, increase fuel flow according to above rate of temperature increase until normal speed is attained.

17. Synchronize the generator.

18. Apply load by varying fuel flow in conformity with rate of temperature increase established above. If other type of load-absorbing means is involved, apply previously established appropriate rate of temperature increase until desired conditions obtain.

19. Shut down auxiliary oil pump when main pump is furnishing proper relay pressure.

Compressor Stability. In the operation of gas turbines care must be exercised not to permit conditions to exist which would cause the compressor to operate in an unstable region. This applies particularly where axial-flow compressors are involved, as this type is the most susceptible to instability. This results from the steep pressure-volume characteristic associated with compressors of this kind. Unstable operation is referred to as "surging" or "pumping" because of the pulsations which occur when working beyond the normal operating range. Under such conditions violent surging may occur, resulting in potential damage to the equipment, thereby indicating the obvious necessity of avoiding such incidents.

Pumping is caused by the existence of excessive pressures at the compressor discharge, resulting in a breakdown of stable operating conditions. The pumping limit may be considered the transition point from the stable to the unstable region. It may be established by increasing the discharge pressure of the compressor at various constant rotative speeds until pulsations ensue at each speed. As the pumping limit is approached, usually the pulsations will gradually become audible, but under some conditions no warning pulses will be observed The detection of the approach to the unstable region can be facilitated by connecting a manometer to the compressor discharge and noting the fluctuating columns. The frequency of the oscillations will probably be less than 1 cycle per sec.

The normal design point of axial compressors when operating at rated speed is such that the pressure at the compressor discharge is approximately 20 to 30 per cent less than that corresponding to the pumping limit at that speed. Notwithstanding such a margin, however, in some cycle arrangements certain combinations of conditions may result in unstable operation. Sudden increases in fuel flow, producing rapid rises in temperature, accompanied by corresponding pressure increments, may cause pumping.

As mentioned previously, pumping ensues when the pressure at the discharge of the compressor exceeds the stable value. The determining factor which establishes the magnitude of the discharge pressure is turbine capacity. Thus the turbine nozzles and blading represent a constriction in the compressor-discharge system. The stability margin will be greatest, therefore, in those cycles where the turbine driving the compressor also furnishes the useful output. In such an arrangement, because of the large turbine capacity, the peak pressures in the cycle would not transcend the pumping limit over the usual load range, and stable conditions would prevail. In cases, however, where the function of one turbine is only to drive the compressor and the useful output is delivered by another turbine on a separate shaft, special design precautions must be adopted to avoid pumping.

One method of accomplishing this is to provide a by-pass arrangement on the compressor turbine so that additional capacity may be obtained by shunting the initial stage of the turbine. Such a procedure is shown in Fig. 8-131. If the turbines are arranged in series and the high-pressure turbine drives the compressor, increased turbine output may be obtained by by-passing the low-pressure turbine so that the exhaust of the compressor turbine discharges to atmosphere. In this manner, because of the greater energy drop produced, additional power is obtainable from the high-pressure turbine without its inlet pressure approaching the pumping limit.

Regulation. Governor regulation of a prime mover is defined as the change in speed, usually expressed as a per cent of normal speed, resulting from a load change equal to the full load of the unit. For steam-turbine practice where the driven unit is an electric generator, the speed regulation is of the general order of 3 or 4 per cent of the normal speed. In the case of steam turbines, the governor regulates the valves controlling the flow of the working substance, steam. Because of the large forces required to actuate the valves, a relay system is employed where the governor only operates a pilot valve. The latter, in turn, by means of a hydraulic system transmits the action of the governor to power pistons which furnish the amplified forces for positioning the control valves.

With gas turbines a different system of control can be used, requiring much smaller actuating forces. Instead of regulating relatively large valves which control the flow of the motive fluid, as is done in steam-turbine practice, a comparatively simple governing arrangement can be employed in gas turbines whereby the output is varied by regulating the fuel flow. In this instance, a single valve of small capacity is involved, and the required control mechanism can be of reduced proportions. Thus, because of the lower masses and resulting shorter accelerating periods, the gas-turbine mechanical-governing system adapts itself to the attainment of rapid response characteristics. The over-all control characteristics of a gas-turbine unit also depend, however, on the rate of thermal response encountered during the combustion process resulting in the transferring of energy from the fuel to the motive fluid and the subsequent conversion to mechanical work in the turbine.

Owing to the limited number of gas-turbine-electric-generator types of units that have been constructed, there are not many governor-regulation data available. However, the published tests[16] of a 4,000-kw unit, built by Brown-Boveri and installed at Neuchâtel, Switzerland, include regulation results. During these trials, when the load was changed from full to no load, transient and settled speed increases of 5.8 per cent after 22.5 sec and 2.5 per cent after 1 min, 48 sec, respectively, were observed. Application of the load from no load to full load resulted in transient and settled speed decreases of 5 per cent after 33 sec and 2.65 per cent after 1 min, 45 sec, respectively.

MAINTENANCE

As mentioned previously, the commercial application of the modern combustion type of gas turbine has been rather limited and of short duration, so extensive maintenance data regarding gas turbines in general have not been established as yet. The use of the gas turbine in the Houdry catalytic cracking process has afforded about the only opportunity for a consistent service record to be obtained on units in the prime-mover category. The first Houdry unit was installed in 1936, and in the intervening 12 years a total of 38 units have been built, 32 by Allis-Chalmers and 6 by Brown-Boveri, so a sufficient background of experience now exists from which reliable deductions may be formed. Of this number, one unit has a normal rating of 16,000 cfm, 16 of 23,000 cfm, 20 of 40,000 cfm, and one of 60,000 cfm. If used as conventional power units operating on the basic open cycle at a turbine-inlet temperature of 1200°F, these four sizes would develop respective net coupling outputs of approximately 900 bhp, 1,400 bhp, 2,600 bhp, and 4,400 bhp.

One of the principal users of oil-refinery gas turbines has been the Sun Oil Company, of Philadelphia, Pennsylvania, and a paper[17] describing this company's service experience with gas turbines has been prepared. This paper presents one of the few authoritative published discussions of gas-turbine maintenance based on actual operating records, and continued reference will be made to it in this section. Since the method of operation of the oil-refinery turbines has been given in a previous section, it will not be repeated here.

The seven Houdry gas turbines which the Sun Oil Company has been operating in its oil refineries are given in Table 8-44. All the units are of the 40,000-cfm size, shown in Fig. 8-102, except unit D which is rated 16,000 cfm and is a modified 23,000-cfm unit, the latter being illustrated by Fig. 8-101.

Table 8-44. Houdry Gas-turbine Installations, Sun Oil Company

Plant	Location	Capacity		Rpm		Starting device	Date in service	Manufacturer
		Compressor std cfm	Generator hp	Turbine and compressor	Generator			
A	Marcus Hook, Pa.	40,000	1,500	5,180	1,800	Motor	Dec., 1936	Brown-Boveri
B	Marcus Hook, Pa.	40,000	1,500	5,180	1,800	Motor	July, 1939	Brown-Boveri
C	Toledo, Ohio	40,000	1,500	5,180	1,800	Motor	Aug., 1939	Allis-Chalmers
D	Marcus Hook, Pa.	16,000	500	6,000	1,800	Turbine	Nov., 1940	Allis-Chalmers
E	Marcus Hook, Pa.	40,000	1,500	5,180	1,800	Turbine	Dec., 1940	Allis-Chalmers
F	Marcus Hook, Pa.	40,000	1,500	5,180	1,800	Turbine	Jan., 1941	Allis-Chalmers
G	Marcus Hook, Pa.	40,000	1,500	5,180	1,800	Turbine	Aug., 1943	Allis-Chalmers

Types of Failures. The operating experience obtained has revealed that the principal causes of shutdown are (1) bearing failures, (2) lubrication failures, and (3) blading failures. A list of all the types of failures encountered which resulted in outages is given in Table 8-45 in the order of their prevalence.

An itemization of the type, frequency of occurrence, and duration of the failures causing gas-turbine outages is given in Table 8-46. It will be noted that bearing failures represent 63 per cent of the total number and account for 56 per cent of the total outage duration. Lubrication difficulties are next, with percentages of 13 and 15, respectively, followed by blading troubles with 9 and 11. Reference to Table 8-46 will show that, while bearing troubles were predominant, the ratio of frequency of occurrence to duration in the case of bearings is higher than that of the other types of failures, indicating that individual bearing failures caused less proportionate delay than the others.

Table 8-45. Types of Failures Causing Gas-turbine Outages, Sun Oil Company

1. *Bearings*
   - Thrust bearings
   - Spring rings (roller bearings)
   - 1,500-hp motor bearings
2. *Lubrication*
   - Dirt in oil sprays, causing bearing failures
   - Water in oil (condensate) causing bearing failures
   - Main oil pumps, broken shaft spline as result of bearing failures
   - Auxiliary oil pump, failure to start when main pump failed
3. *Blades*
   - Loose blades (latest design with integral root, no loose blades in 22 months on one unit)
   - Erosion, gas-turbine blades.
4. *Air-seal strips* (failure does not necessarily cause shutdown)
   - Material
   - Rubbing, due to change in alignment
   - Expansion joints, harness
5. *Explosions*
   - By-passing air into oil vapors, experimental case
   - Reformer, air by-passed into oil vapors
   - Cases, gas explosion, gas burner opened by mistake
   - Dirty oil burner, no atomization
6. *Miscellaneous*
   - Foreign objects in compressor or turbine
   - Failure to synchronize before cutting motor in when starting unit

In Table 8-44, units A to F, inclusive, were originally installed with an angular-contact ball-bearing type of thrust bearing, but, because of the difficulties that developed, these have been changed to the Kingsbury type. Unit G has always had a Kingsbury thrust bearing. There have been no thrust-bearing failures with the Kingsbury type, and the belief is advanced that the bearing change has essentially eliminated future trouble from this source. Some bearing outages were caused by failure of the spring support rings of the roller bearings supporting the rotating shafts. In later designs where sleeve bearings have been used for this purpose, an exemplary service record has been achieved.

Table 8-46. Number and Duration of Failures Causing Gas-turbine Outages, Sun Oil Company

Plant	Bearings		Lubrication		Blades		Air seals		Explosions		Miscellaneous		Total	
	No	Days	No.	Days	No	Days	No.	Days	No	Days	No	Days	No.	Days
A	18	43							1	5	1	2	20	50
B	4	12			1	4	1	3	1	2			7	21
D	4	31	3	14									7	45
E	3	15	3	11					1	5			7	34
F					3	15					2	14	5	29
G													0	0
Total	20	101	6	28	1	19	1	3	3	12	3	16	46	179
Per cent	63	56	13	15	9	11	2	2	6.5	7	6.5	9	100	100

The outages caused by lubrication difficulties resulted primarily from foreign matter in the oil and oil-pump failures. The installation of blotter presses has apparently corrected this condition as no lubrication failures have occurred since they were used.

Blading failures, while in third place as far as duration is concerned, are probably the most serious type of individual failure with respect to the delay and expense involved. As has been previously described, the original blading designs comprised separate spacer pieces which held the blades in their grooves. Observations indicate that failures have occurred primarily due to fatigue cracks originating from an indentation produced at the line of contact of the spacer pieces and the blade surfaces. In later designs by Allis-Chalmers, where integral-root blades have been employed, no difficulties have been encountered in an operating period of almost 4 years.

Unavailability of the equipment due to failures of air seals at the ends of the rotating shafts has been of minor consequence. Mishaps of this sort would rarely cause any extended delays as such repairs can be postponed until regularly scheduled shutdown periods.

Table 8-46 also indicates that explosions have been responsible for 6.5 per cent of the total outage period. These explosions occurred because of operating errors with other pieces of equipment in the cycle and have no relation to the design of the gas turbine. No serious damage resulted from any of the three explosions listed in Table 8-46.

Several miscellaneous shutdowns due to maintenance errors and not related to turbine design were experienced. Under this category are included outages caused by leaving foreign objects in the compressor and similar mishaps.

Availability. Reference to Table 8-47 will show that the average availability of the gas-turbine units reported was 98.1 per cent. Figures of this order are very good and indicate a high degree of reliability. It should be pointed out, however, that the Houdry gas-turbine equipment is of conservative design, operating at low stresses and temperature, the latter being in the range of 875 to 900°F which is well below the temperatures advocated for present-day gas-turbine power units. It might be contemplated, therefore, that higher temperature designs may exhibit a tendency toward increased maintenance over that stated above until further operating experience at elevated temperatures is obtained.

The availability of 98.1 per cent noted above corresponds approximately with that of central-station steam-turbine equipment. Such similarity is to be expected, however, as modern steam turbines are operating at approximately the same temperature and stress levels, and the purely rotary motion of both types of turbines is conducive to low maintenance.

Anticipated Availability of Modified Designs. It is interesting to reflect on what the anticipated availability might be of Houdry gas turbines whose designs have been modified in accordance with the dictates of operating experience. For instance, the Sun Oil engineers estimate that thrust-bearing outages would have been reduced 60 per cent if the Kingsbury instead of the ball type of thrust bearing had been used in the original designs. Assuming further that sleeve instead of roller bearings were adopted, a revised value of 70 per cent would probably be reasonable. Using this figure, the outage for bearings of 101 days, given in Table 8-46, would become 30.3 days.

Table 8-47. Summary of Gas-turbine Operation to May 31, 1945, Sun Oil Company

Plant	Total days	Outage		Availability	
		Days	Per cent	Days	Per cent
A	2,657	50	1.88	2,607	98.12
B	1,921	21	1.09	1,900	98.91
D	1,458	45	3.09	1,413	96.91
E	1,438	34	2.36	1,404	97.64
F	1,394	29	2.08	1,365	97.92
G	569	0	0	569	100.00
Total	9,437	179	1.90	9,258	98.10

Referring to lubrication difficulties, since no failures of this type have occurred subsequent to the installation of blotter presses, it is assumed that shutdowns of this nature would be reduced 75 per cent if these had been employed originally. On this basis, the 28 days' outage of Table 8-46 would diminish to 7 days.

As mentioned previously, no blading failures have occurred since the installation of the new integral-root blading in the Houdry gas turbines of the Sun Oil Company. If this modified design had been applied initially, it is reasonable to expect that blade life would at least have been doubled, so that an outage of 9.5 days would result instead of the 19 days of Table 8-46.

In establishing an anticipated availability with modified design, it is assumed that delays due to air-seal difficulties would be of the same order as those actually experienced as these are of minor importance. Also, inasmuch as shutdowns due to explosions and miscellaneous causes had no relation to gas-turbine design, these have been omitted in determining expected availability. Accordingly, if a Houdry gas turbine of modified design functioned in conformity with the aforementioned assumptions and on the same operating schedule as the units reported on, it would have an availability of 99.5 per cent, a value hardly excelled by any other type of equipment.

It is desired to reiterate that the exceptionally good availability figures under discussion refer to equipment designed for long life with relatively low stresses and moderate operating temperatures—under 1000°F. The good service records obtained with existing units do demonstrate, however, that the gas turbine is a reliable piece of equipment and not a radical innovation. Other types of gas turbines now being proposed and built where efficiency, weight, or space is of prime importance will be designed for considerably higher temperatures, probably in the range from 1200 to 1600°F. Designs employing the upper temperature region represent an appreciable advancement in the state of the art and, considering the ultimate gains to be achieved with developments of this nature and scope, any necessary concomitant increased maintenance should be condoned until the experimental phase is consummated.

MATERIALS

As has been previously observed, the performance of gas turbines improves with the use of elevated temperatures. Consequently, the necessity for adopting suitable materials to withstand high-temperature service is manifest. Progress in the gas-turbine field will always be contingent upon metallurgical advances, and real incentives exist for the development of alloys capable of operating at extreme temperature limits. In an attempt to accomplish such attainments, serious consideration is being given to the use of ceramics with the anticipation that the employment of temperatures in excess of 2000°F may be a reality in the not too distant future. Experiments with ceramics, both as a coating for metallic parts as well as a base material, indicate encouraging results. Other than for some limited small-scale laboratory investigations, however, all gas turbines so far have been constructed of metals, and the greatest impetus is being devoted to metallurgical research on metallic materials.

The metallurgical background for the production of adequate high-temperature alloys for gas turbines has been obtained from the many years devoted to the development of materials for use in the construction of steam turbines operating at increased temperatures. After an evolutionary period embracing approximately half a century, the steam-turbine art has reached the stage where it is now considered feasible to build a unit to operate at 1000°F. Reference to the performance data previously presented makes it rather obvious that, for gas-turbine service, temperatures below 1000°F could not be tolerated if reasonable efficiency is desired.

Hence, the seemingly incongruous situation exists where, for similar types of equipment, a temperature of 1000°F represents an upper limit in one case and a lower in the other. The apparent inconsistency is partly explainable when it is considered that steam turbines are normally designed for a minimum service life of approximately twenty years. A comparable figure for gas turbines of the prime-power type is about ten years, which would permit the use of higher temperatures. Also, the maximum pressures encountered in open-cycle gas turbines are less than 10 per cent of those of modern steam plants, so for units of the same output and factor of safety, the stationary parts, such as piping and casings, of an efficient gas-turbine cycle may operate at increased temperatures, notwithstanding the larger specific volumes of the motive fluid involved.

Furthermore, because of the required higher temperatures of gas-turbine plants it has been necessary to accept creep rates approximately ten times as much as those used in the design of corresponding parts of steam turbines, and the stress level is lower than that found in steam turbines. In addition, because of the much shorter bearing span of spindles for gas turbines, somewhat higher temperatures can be sanctioned since the clearance and distortion problems under otherwise similar conditions are normally less acute

In general, at the present time the maximum design temperature for prime-mover type of gas turbines having a service life of about ten years is approximately 1300°F. Higher temperatures than this have been employed, however, in special applications. For instance, temperatures as high as 1600°F are used in gas-turbine superchargers for aircraft. Also, for aircraft jet-propulsion units, temperatures of 1500°F are used at the inlet to the gas turbine. In these cases, however, since space and weight are at a premium, the higher temperatures required to produce these features are condoned, and limited-service life is accepted. In aircraft equipment of this type, such life is considered in terms of hundreds of hours instead of years.

High-temperature Alloys. The materials used in the higher temperature steam turbines were not considered adequate for gas-turbine service, so since 1940 a large-scale metallurgical development program has been conducted under sponsorship of various government agencies. Scores of different alloys were investigated, but at present some of the most promising materials[18] appear to be as shown in Table 8-48.

The alloys listed in the table are of two general kinds: forged and cast. The forged type can be further divided into two classes: those which are normally used without heat-treatment and those which usually obtain their optimum properties when heat-treated. Alloys 1 to 6, inclusive, would be of the first class and 7 to 18, inclusive, of the second. Alloys of the first class are usually cold-worked so as to realize the best results from the materials. The cast alloys, such as 19 to 24, inclusive, when used for smaller parts, such as turbine blading, are made by the lost-wax precision-casting process which will be described in more detail later. Alloys 25 and 26 have been used for turbine diaphragms and casings which are produced by essentially standard foundry methods. While alloys 1 to 18 have been included in the forged classification, some of them, also, show good properties when in precision-cast form.

Table 8-48. High-temperature Materials for Gas Turbines

Alloy	Name	Chemical composition, per cent												
		C	Si	Mn	N	Cr	Ni	Mo	Co	W	Cb	Ti	Al	Fe
	Forged alloys													
1	17W	0.40				14.00	19.00	0.50		2.50				
2	19-9 W Mo	0.11	0.42	0.60		18.87	8.63	0.40		1.36	0.28	0.45		
3	19-9 DL	0.20	0.57	0.52		18.95	9.05	1.22		1.19	0.29	0.21		
4	234 A 5	0.38		4.00		19.0	5.00	1.00		1.00	0.50			
5	16-25-6 (Timken)	0.06	0.85	1.54	0.07	16.55	25.20	6.46						
6	Gamma columbium	0.38	1.21	0.74		14.51	21.04	4.41			4.34			
7	Nimonic 80	0.04	0.47	0.56		21.18	74.23					2.44	0.63	
8	Inconel (mod.)	0.03	0.52	0.53		14.0	75.0					2.77	0.63	
9	Refractaloy 26	0.03	0.65	0.70		17.9	37.0	3.03	20.00			2.99	0.25	
10	N 153 (low C)	0.08			0.15	15.0	15.0	3.00	13.00	2.00	1.00			
11	N 153 (high C)	0.38	0.52	1.78	0.07	16.20	14.98	3.01	12.82	2.19	1.06			
12	N 155 (low C)	0.14	0.52	1.48	0.14	21.30	20.00	3.06	20.00	2.20	1.10			
13	N 155 (high C)	0.32	0.59	1.54	0.11	21.08	20.80	3.00	20.54	2.18	0.98			
14	S 495	0.50	0.88	0.73		14.44	20.52	4.38		3.92	2.76			
15	S 590	0.49	0.21	0.60		19.50	19.78	3.95	19.35	4.15	4.04			
16	S 816	0.38	0.25	0.82		18.87	19.70	4.04	45.64	4.71	3.43			2.94
17	Hastelloy B	0.05	0.19	0.59			65.10	28.63						4.71
18	K 42 B	0.06	0.34	0.70		18.00	42.00		22.00			2.56	0.59	13.00
	Cast alloys													
19	Vitallium	0.25				28.00	2.00	6.00	Bal					
20	61	0.45				28.00	2.00		Bal	6.00				
21	6059	0.45				28.00	35.00	6.00	Bal					
22	422-19	0.45				28.00	16.00	6.00	Bal					0.65
23	N 155	0.45				20.00	20.00	3.00	20.00	2.00	1.00			
24	X 40	0.48	0.72	0.64		28.00	12.00		Bal	7.50				0.55
25	25-12	0.30	1.00	0.80	0.11	25.00	12.00							
26	25-20		2.00			25.00	20.00							

The materials which have probably been used in greatest quantities are the 16-25-6, alloy 5 and vitallium, alloy 19. These two materials were used for disks and blades, respectively, in the gas-turbine wheels of the aircraft turbosuperchargers which were manufactured in such large volume during the Second World War. One of these superchargers is shown in Fig. 8-140, and the exposed turbine wheel may be readily observed. In this application, the cast-vitallium blades were welded to the forged 16-25-6 wheel by means of the union-melt process. More than 6,000 tons of the Timken alloy have been produced, so much experience has been gained with it. Before adopting a high-temperature alloy, it is desirable to determine its properties when produced in large heats, as the results obtained from laboratory samples may not be entirely representative.

Elements. The metallurgy of alloys suitable for high-temperature service is a complex phenomenon, and the attainment of a superior alloy is usually the result of painstaking experimentation. It is generally known what elements are desired to produce beneficial high-temperature properties, but even after the kind and magnitude of the ingredients of an alloy have been established, it is usually the subsequent treatment and processing that determine the final properties of the material.

In general, the elements given in Table 8-48 contribute some desirable characteristic to high-temperature alloys. Practically all high-temperature materials will be of the chromium-nickel type, and most of them will contain molybdenum. The forged and cast alloys are usually of the iron- and cobalt-base type, respectively, but some work is being done on vacuum-cast chromium-base alloys which show much promise for still higher temperatures.

Molybdenum is a desirable constituent as it produces strength at high temperatures. Chromium offers resistance to oxidation and corrosion. Nickel, in proper proportions, improves ductility and produces an austenitic structure. It also increases the strengthening effect of carbon and the corrosion resistance of chromium. Carbon and nitrogen have strengthening and stabilizing characteristics, but higher carbon contents tend toward lower ductility, which is undesirable. Nitrogen also improves grain refinement and ductility in steels high in chromium. Silicon in small amounts in combination with other alloying elements, such as chromium, enhances resistance to corrosion and heat. Manganese acts as a deoxidizing agent and, in proper amounts, increases strength. Cobalt in many respects is similar to nickel in the properties which it produces. Tungsten as an alloying element has an effect similar to molybdenum, but several times as much is usually required. Titanium and columbium are stabilizing elements used to prevent intercrystalline carbide precipitation. Aluminum is primarily used as a deoxidizer and for grain refinement. Resistance to oxidation is also improved by the use of aluminum.

Physical Characteristics. In the design of high-temperature gas turbines, the following information is required regarding the materials involved in the construction of the equipment:

1. Ultimate strength
2. Yield strength
3. Stress-rupture
4. Creep
5. Fatigue
6. Elongation
7. Reduction in area
8. Hardness
9. Impact
10. Oxidation
11. Corrosion
12. Damping capacity
13. Relaxation
14. Modulus of elasticity
15. Thermal expansion
16. Thermal conductivity

Most of the above tests are fairly well standardized, so no further elaboration of their nature will be presented here. The stress-rupture test, however, may not be so well known as it has become particularly significant recently because of the advent of the gas turbine. It probably furnishes the most useful information to be used in the design of gas turbines. It can be considered as a modified or long-time type of ultimate-strength test.

The test for ultimate strength is a short-time tensile test at constant temperature where failure occurs in a matter of minutes. If a sustained lighter load had been applied, a longer time would have been required for failure of the specimen. Therefore, if a succession of diminishing static-breaking loads is applied to a group of specimens at a constant temperature, respective failures will take place at correspondingly increasing time intervals. A test of this nature is referred to as the stress-rupture type. The stresses at which rupture occurs are usually plotted as ordinates against the times for rupture as abscissas on log-log coordinates, and a linear relationship is generally found to exist for most alloys. In the majority of cases, tests of this nature are usually conducted for a period of about 1,000 hr, and the linear-relationship feature affords a ready means for extrapolation to longer time intervals. This latter operation should be pursued with caution, however, as erroneous predictions can result, especially if the material is susceptible to oxidation.

A conception of some representative stress-rupture values at 1200 to 1500°F may be obtained from Fig. 8-121 where alloys 16-25-6,[19] S-590,[20] and 422-19[18] have been selected at random for illustrative purposes. Alloys 16-25-6 and S-590 are of the forged type, and 422-19 is precision-cast. It can be seen that the cast alloy shows superior properties at high temperatures. The forged 16-25-6 alloy, as mentioned previously, has been used in very large quantities for supercharger and jet-propulsion gas-turbine disks where temperatures are in the 1200°F range. It has very good stress-rupture properties at such temperatures. For gas-turbine disks operating in the vicinity of 1500°F, some of the more recently developed forged alloys, such as S-590 or S-816, show good characteristics. It will be noted that the relative difference between S-590 and 16-25-6 at 1200°F is less than at 1500°F. For smaller parts, such as blading, which adapt themselves to the size limitations of the precision-casting process, the cast alloys lend themselves to ready adoption and show good strength at high temperatures, as can be seen from Fig. 8-121.

It will also be noted from Fig. 8-121 that the slope is steeper at the higher temperatures, indicating that materials lose strength at a more rapid rate at elevated temperatures. Because of the importance of this time factor involved, the stress-rupture test is possibly the most informative for furnishing high-temperature gas-turbine design data.

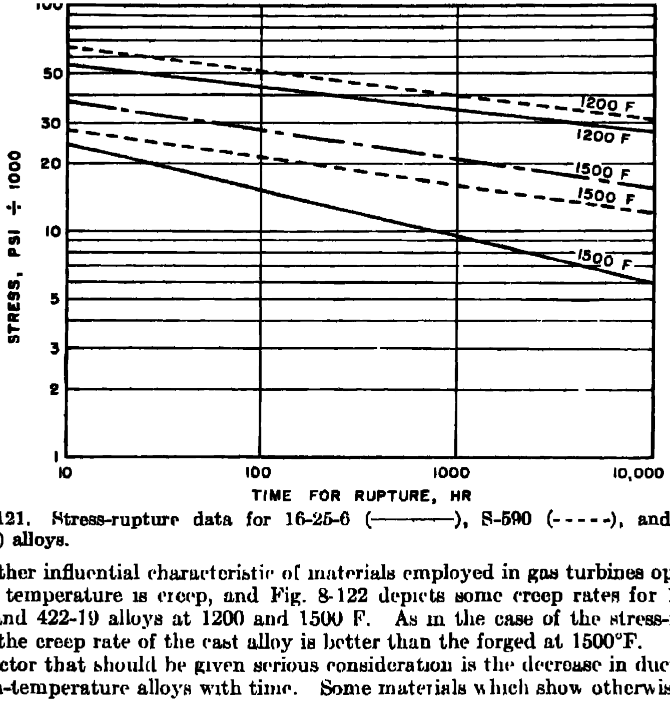


Fig. 8-121. Stress-rupture data for 16-25-6 (————), S-590 (- - - - -), and 422-19 (— — —) alloys.

Another influential characteristic of materials employed in gas turbines operating at high temperature is creep, and Fig. 8-122 depicts some creep rates for 16-25-6, S-590, and 422-19 alloys at 1200 and 1500 F. As in the case of the stress-rupture values, the creep rate of the cast alloy is better than the forged at 1500°F.

A factor that should be given serious consideration is the decrease in ductility of the high-temperature alloys with time. Some materials which show otherwise excellent properties exhibit elongations as low as 5 per cent after testing for several hundred hours at temperatures in the range of 1500°F. In general, the cast alloys show the lowest ductility under sustained stress at high temperatures.

While a material may exhibit good short-time properties, if it has poor oxidation or corrosion resistance, failure may occur after a certain period of operation because of increased stresses caused by reduced cross section resulting from loss of metal. Therefore, these characteristics should be given adequate consideration because of the possibility of the diminishing factor of safety which can occur after a design has been established. Oxidation tests on some high-temperature alloys, by the gain-in-weight method, show an increase in weight of about 1 mg per sq cm at 1500°F in 1,000 hr.

Comparison of Forged and Cast Alloys. No one type of alloy is superior in all respects. Forged materials have advantages in certain characteristics and cast in others. In stress-rupture tests at 1500°F, the cast alloys of the Co-Cr-Ni- and Co-Cr-base type show better strengths than the forged alloys. The higher the temperature, the more predominant the difference becomes. In general, the cast alloys show better creep resistance at 1500°F than forged alloys, but this situation is reversed at 1350°F. Also at 1200°F, the forged alloys have the best fatigue properties, while at 1500°F the cast alloys are better. The ductility of the forged alloys is better than that of the cast. In general, it may be stated that in the vicinity of 1500°F and above, the cast alloys, except for ductility, exhibit properties superior to those of forged materials.

Because of the forging operation, the wrought materials have a fine-grain structure in contrast to the coarse grains of the cast alloys. The larger grains of the cast materials result in beneficial high-temperature stress-rupture and creep properties as their presence reduces the magnitude of the weaker grain boundary. On the other hand, the coarser grains decrease the endurance limit.

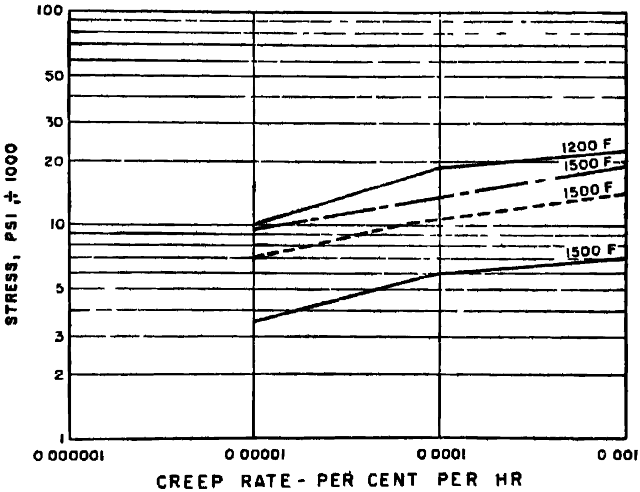


FIG. 8-122 Creep rates for 16-25-6 (— — —), S-590 (- - - - -), and 422-19 (— — —) alloys

The test results obtained with forged alloys appear to be more consistent and amenable to duplication from one heat to another than do those of cast materials. This is probably due to the more difficult control aspects of the precision-casting process. Another important consideration is that for forged alloys the physical properties of the material as determined in the laboratory are essentially representative of those of the machined part as finally installed in the turbine. For cast alloys, however, there is the possibility of divergence between the properties as obtained in the laboratory and those actually existing in a precision-cast part of the same material. In a precision-cast test specimen of uniform circular section, the grain orientation will be different from that of a turbine blade of unsymmetrical cross section. The results obtained on a test specimen with essentially uniformly arranged coarse grains will not necessarily be applicable to a blade where the thin trailing edge will exhibit a fine-grained structure, and the heavier inlet and center portions will have larger grains

Under such circumstances, because of the unrelated grain structures, the physical properties obtained on the test specimen are not directly applicable to the blade, and to use them as a basis for design may result in fallacious factors of safety. The situation is particularly aggravated in the blade-trailing portion where greatest divergence from test conditions occurs The blade would tend to have lower stress-rupture and creep-resistance properties because of its finer grain structure in this region.

Precision-casting Process. The precision-casting or "lost-wax" process, previously used primarily by the dental and jewelry trades, was adopted during the Second World War as a means of mass-producing gas-turbine supercharger blades The vitallium alloy, listed in Table 8-48, has been employed to the greatest extent, but other shapes and alloys are being used. The process actually had its origin in China about 2000 B C. and was perfected by Benvenuto Cellini for use in his medieval statuary. It was then dormant for many centuries until revived by the dental trade. The term "lost wax" is applied since the wax pattern is melted out and, therefore, "lost" during the process The method is a valuable contribution to gas-turbine construction as some of the best high-temperature alloys are produced by it.

The precision-casting process requires that first a master pattern be made of metal or wood, slightly oversize to allow for shrinkage or draft. From the master pattern, an injection die of soft metal is prepared A special wax is injected into the die under pressure and assumes a shape identical to the master pattern After solidifying, the wax pattern is removed, and several of them are mounted in a cluster, connected to each other by wax runners, and these, in turn, are joined to a main gate The runners and gate provide channels for the subsequent flow of metal into the mold.

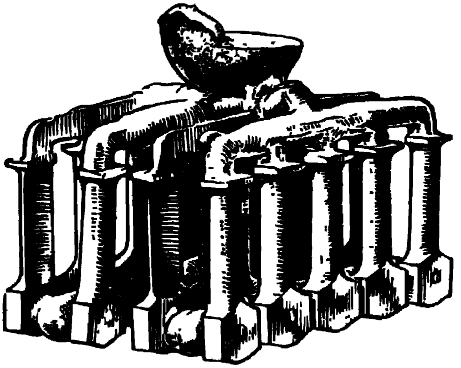


FIG. 8-123. Precision-cast, gas-turbine, supercharger blades

The wax-pattern assembly is coated with a fine silica solution which will become the mold facing and give a smooth velvet finish to the surface of the final metal casting. The coated wax assembly is placed in a container in which an ethyl silicate mixture, or investment, is poured around it. The container or flask is then vibrated until the investment solidifies into a firm, self-venting mold surrounding the wax assembly. The completed molds, in inverted position, are placed in an oven where the melted wax drains out, leaving cavities in the mold which are replicas of the master pattern.

The incandescent mold is clamped with its opening opposite that of a furnace containing the molten metal to be cast. The assembly is inverted, causing the metal to run into the mold, and air pressure is applied to ensure filling all cavities After cooling, the investment is knocked from the casting assembly. Figure 8-123 shows such an assembly of gas-turbine supercharger blades with the gates and risers still in place. An abrasive wheel then cuts off the gates and risers, and the individual cast pieces are sand- or shot-blasted. Any local surface irregularities are corrected by grinding, but no machining is required. The castings are finally inspected radiographically by automatic X-ray equipment.

Practically all kinds of castable metals can be used in the lost-wax process. Tolerances ranging from 0 003 to 0.008 in per in are possible of attainment, depending upon the shape and weight of the objects being cast The process is particularly suitable for intricate shapes that are too complex for conventional casting or machining. In its present state of development, the maximum size of part for which the method is suitable is approximately 20 lb, with no dimension exceeding 15 in.

In the precision-casting process, it is important that accurate control be maintained if it is desired to produce anticipated properties in the product. Grain size is directly affected by metal pouring temperature and mold temperature. The nature of the investment material and the shape of the casting are also influencing factors. Therefore, in order to obtain uniform results, coordination of all these variables is necessary.

Design Considerations. In steam-turbine practice, various properties of the materials employed have been used as a basis for design. In particular, the ultimate, yield, and creep strength values have been found useful in determining the proportions of structures having suitable factors of safety based on these properties. In gas-turbine design, however, where higher temperatures are usually encountered, the most informative property of the materials, from a design standpoint, is the stress-rupture value, as has been mentioned previously. It is more significant than ultimate strength as it involves the element of time to a much greater extent.

Reference to Fig. 8-121 will show that the higher the temperature, the more rapidly the material loses strength. Thus, the designer must take cognizance of the time factor involved when selecting or specifying materials. In steam-turbine design where substantially lower temperatures are involved, the time element is not of so much significance as far as rate of loss of strength is concerned. In gas turbines, however, where elevated temperatures are necessary, the length of operating period must be considered.

Therefore, in order to obtain a comprehensive representation of all the factors involved in the design of a high-temperature gas turbine, it is necessary to consider simultaneously (1) the desired service life of the equipment, (2) the temperature at which it will operate, (3) the allowable deformation of the respective parts, and (4) the appropriate stress.

Knowing the time and temperature factors involved, a design could be based on the total allowable deformation of the respective elements of the equipment occurring in the time interval. The permissible stress would then be the one that resulted in the prescribed deformation in the time allotted and at the specified temperature for a given material. If the deformation is put on a relative basis, it is referred to as strain and is stated in terms of per cent. It can be determined from the initial deformation under load and the secondary creep rate. In the design of a unit intended for long service life, it would be well to limit the maximum total strain to 1 per cent. This is particularly applicable to the high-temperature materials which would be used in gas-turbine construction. It is known that the ductility decreases with time, and in general, tests have not been performed much beyond 1,000 hr on these materials, so for conservative design, the total deformation should be restricted to a low value so as to preclude the possibility of failure due to poor ductility.

Another basis of design is to use stress-rupture values with a suitable factor of safety incorporated. A satisfactory design of a long-life unit, *e.g.*, a service life of 10 years, could employ average stresses of one-half the rupture value at 10,000 hr for the particular temperature involved. This would be a more conservative basis than limiting the deformation to 1 per cent total strain. In aircraft applications, where weight and space are at a premium, limited service life is accepted, and higher stresses are used. On this basis, some aircraft gas turbines have been designed with a factor of safety of 1 based on the stress-rupture value at several hundred hours.

Actually the above two bases of design could be considered equivalent. The *allowable total strain* could be determined as that produced by a rupture stress at a given time corrected for an appropriate factor of safety. Conversely, an *allowable stress*, based on a rupture value incorporating a safety factor, could be selected so as to result in an acceptable total strain. In actual design practice, however, using the latter or rupture basis with a safety factor appears to be the most direct and readily applied procedure. Nevertheless, it should be remembered that both methods will produce the same result if they are mutually appropriately related.

From a design standpoint, other properties of materials are also important. The fatigue or endurance limit should be considered because of the vibratory nature of the forces encountered in turbine practice. It is difficult, however, to establish a design criterion based on fatigue, for at high temperatures there does not appear to be a definite fatigue limit even after 500 million cycles of testing in some cases. It is usually assumed that a design based on rupture or total strain considerations, as previously described, will result in stresses well removed from the endurance limit, so the former bases are believed to reflect adequate conservatism.

The amplitude of vibration of a structure is influenced by the damping capacity or logarithmic decrement of the material. The damping capacity is a measure of the tendency of a material to come to rest, because of internal molecular friction, after it has been set freely vibrating. Because of the pulsating exciting forces prevalent in turbine designs, harmful vibrational stresses can occur, particularly in partial-admission turbines, and a high damping capacity is desirable so as to reduce such stresses to a minimum. Unfortunately, the high-temperature alloys do not exhibit good damping properties, particularly with increasing time and temperature, so compensating features must be incorporated into the design.

The natural frequency of the various parts of construction and the critical speeds of the rotating elements are dependent upon the modulus of elasticity of the material employed. At elevated temperatures the modulus decreases appreciably from its value at room temperature. For instance, the latter figure[20] for S-590 is 29 million psi, while at 1400°F the modulus of elasticity is 16 million psi, a reduction of 45 per cent. Consequently, in order to avoid dangerous resonant conditions, high-temperature gas turbines require designs with relatively higher room-temperature vibrational-frequency characteristics than steam turbines if similar multiples of rotational frequencies are desired under operating conditions.

In designing the bolting on gas turbines, it is important to know the relaxation properties of the material employed. When a nut and bolt combination is tightened initially, a strain results which produces a stress sufficient to maintain a tight joint With the passage of time, the bolt will lengthen or creep, because of plastic flow, and a relaxation or residual stress less than the original will ensue. The magnitude of the residual stress may be calculated by special formulas[21] after the relaxation-creep properties of the alloy have been determined It is necessary that the residual stress be sufficiently high to maintain joints without leakage. Relaxation-creep tests on the forged S-816 alloy show that, with an initial tightening stress of 50,000 psi at room temperature and a resulting deformation of about 0.173 per cent, the residual stress after 1,000 hr at 1350°F would be 11,725 psi for the same strain. After 10,000 hr, other conditions remaining the same, the residual stress would be 8,860 psi. For the same conditions, the corresponding residual stresses for 19-9 DL are 6,925 psi and 4,115 psi, respectively.

Thermal-conductivity data are also required. The use of the high temperatures encountered in gas-turbine practice necessitates in many instances the adoption of various means of cooling some of the parts. The turbine shown in Fig 8-131, for instance, has provision for cooling some of the disks and nozzles. The quantities of cooling air required and the heat flow in the metallic parts can be determined with a knowledge of the thermal conductivity of the materials involved.

The thermal expansions of the various elements of a gas-turbine installation can be of appreciable magnitude because of the large temperature differences involved. From cold to hot conditions, the axial expansion of a turbine may approach ½ in. Throughout such motion the proper axial clearances between the stationary and rotating elements must be maintained. Hence, it is necessary to know the coefficients of thermal expansion of the materials employed so as to make correct allowance for relative displacement. The problem is accentuated with the types of alloys used in gas-turbine construction as their expansion coefficients are appreciably higher than those of ordinary steel. For instance, at 1500°F the S-590[20] alloy has a coefficient of 8.4×10^{-6} in. (in.)(deg F). The piping is also influenced by thermal motion, and in order to prevent excessive strains, it is necessary to incorporate expansion and swing joints in appropriate locations.

The machinability and weldability characteristics of an alloy must be considered as they directly affect the manufacturing of the equipment. Most of the high-temperature materials do not machine as readily as ordinary steels, and some of the cast alloys necessarily have no machining operations performed on them, but grinding is used instead For turning operations cemented-carbide tools are generally used, and high-speed steel tools are employed for milling, broaching, reaming, and drilling

In many cases the design is facilitated if a welded construction is employed. This applies particularly to the fastening of blading to disks, for in such instances, for the same stress, welding will permit a narrower rim width and, consequently, shorter spindle with higher critical speed than if a mechanical blade fastening is used Welding is also used in joining adjacent disks to each other With the high-temperature alloys, in particular, there is a size limitation on the obtaining of sound forgings Therefore, in some designs the disks are machined from individual forgings and subsequently welded together to form the complete spindle In the turbine shown in Fig 8-131 the two impulse disks and the reaction drum were each machined from separate forgings and then joined by welding The resulting structure was then welded to the adjacent shaft at the low-pressure end of the reaction drum

GAS-TURBINE APPLICATIONS

Various present-day applications of the gas turbine, indicating the practicability of this type of equipment, have furthered its emergence from a realm of academic interest to one of reality The many advantages of the gas turbine make it a natural aspirant for numerous applications in land, marine, rail, and aircraft fields, and these will be dealt with individually below

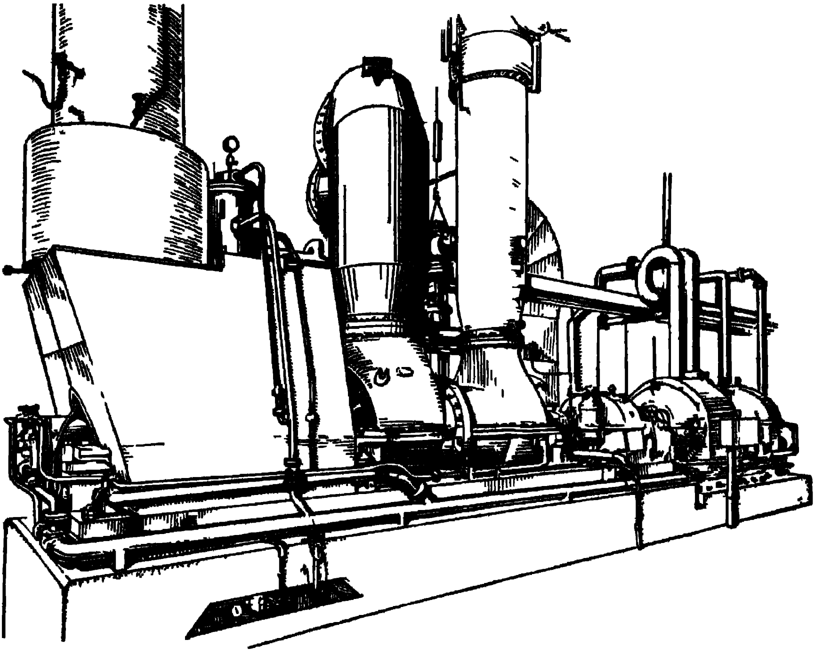


Fig 8-124 Allis-Chalmers Houdry gas-turbine unit of 60,000 cfm capacity.

Land. As mentioned previously, the combustion gas turbine in the United States has had its principal commercial application in the oil-refinery industry where it operates in conjunction with the Houdry process A typical installation view of such an application employing a 60,000-cfm unit is shown in Fig 8-124, wherein, from left to right, may be seen the gas turbine, axial compressor, reduction gear, generator, and a steam turbine for starting purposes In this application, because of the pressure drop encountered in the process, the energy supplied to the generator is less than if the unit were used primarily for purposes of power production.

The first unit of this type has been in service since December, 1936. At present 28 of the Houdry gas turbines are in operation in this country. Of these, 6 were built by Brown-Boveri and 22 by Allis-Chalmers. The latter company has also built 10 additional units for installation in Europe.

A 4,000-kw electric-generating unit,[16] built by Brown-Boveri, has been installed in an underground emergency stand-by power station in the city of Neuchâtel, Switzerland. Since early in 1940 this unit has been operating for certain periods every week. Its simplicity and independence of water facilities ideally adapt the gas turbine to this class of service. Recently published tests indicate a coupling thermal efficiency of 18.04 per cent for this unit when operating on a nonregenerative cycle with a turbine-inlet temperature of 1000°F. Regeneration was not adopted since it could not be economically justified because of the anticipated intermittent operation associated with this particular installation. This is the first unit designed for the primary purpose of producing power. The Neuchâtel unit on exhibition at the Swiss National Exposition at Zurich is shown in Fig. 8-125.

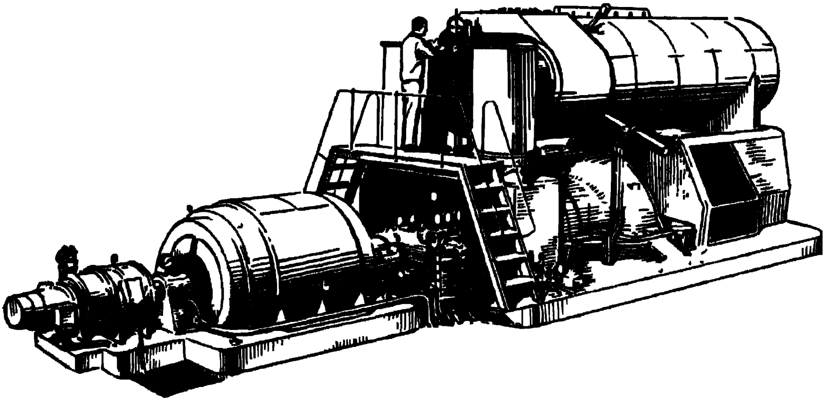


FIG. 8-125. Brown-Boveri gas-turbine–electric-generating unit of 4,000 kw capacity.

Brown-Boveri has also built a 10,000-kw gas-turbine electric-generating unit intended as a stand-by set for the Filaret plant of Bucharest, Rumania. The machine consists of a two-shaft arrangement employing reheating and intercooling, but because of the stand-by nature of the service, no regenerator will be used. The equipment is shown in Fig. 8-126, the low-pressure turbine, compressor, and generator being at the left, the combustion chambers, the high-pressure turbine, and compressor in the center, and the intercoolers on the right. A 4,000-kw unit for Peru is shown in the upper right center of the figure. Tests on the unit showed a thermal efficiency of 23 per cent at full load, based on the lower heating value of the fuel. With a turbine-inlet temperature of 1112°F and inlet air of 68°F, a maximum load of 12,000 kw was maintained.

About 80 gas-turbine–axial-compressor sets have been built by Brown-Boveri to operate in conjunction with the Velox boiler, a pressure-combustion type where the axial compressor furnishes air at a pressure of 30 to 35 psi gauge to the combustion chamber, and the resulting products of combustion actuate the gas turbine; and hundreds are used for supercharging diesel engines.

While no gas-turbine–electric-generating units have been installed in any central stations in this country as yet, much thought is being devoted to such applications, and a model of a 10,000-kw unit proposed by Allis-Chalmers for utility service is shown in Fig. 8-127. A schematic diagram of its regenerative cycle with reheating and intercooling is also depicted. A two-shaft arrangement is employed whereby the high-pressure turbine drives the compressors, and a separate, double-flow, low-pressure turbine is connected to the 10,000-kw generator The cycle is designed for an inlet temperature of 1300°F to both turbines, and regenerator and intercooler

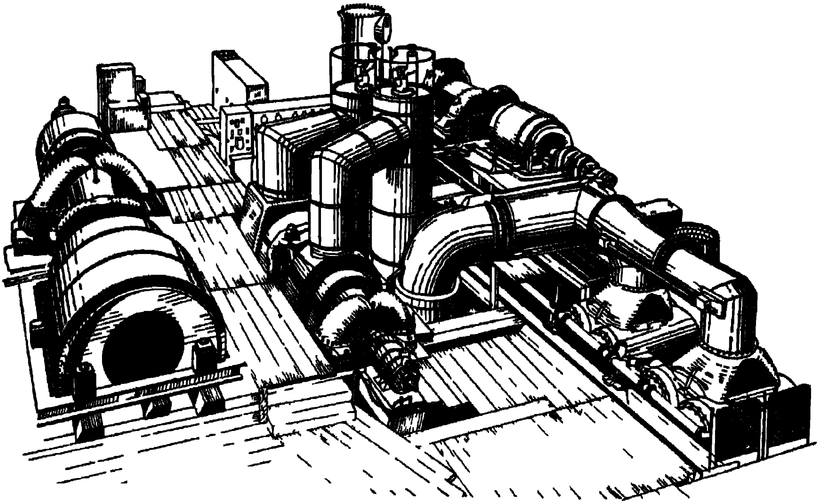


Fig 8-126 Brown-Boveri gas-turbine electric-generating unit of 10,000 kw capacity

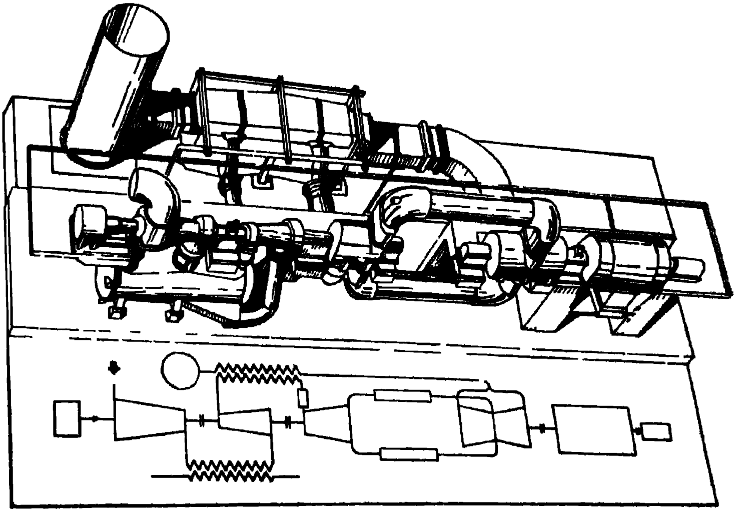


Fig 8-127 Model of 10 000-kw Allis-Chalmers gas-turbine unit including cycle diagram.

effectivenesses of 75 and 85 per cent respectively, under which conditions a coupling thermal efficiency of 30 per cent is obtainable

A promising application for the gas turbine is its use as a blast-furnace blower In such an arrangement no excess power need be developed, but, instead, the equivalent required quantity of air for the blast-furnace operation is tapped off the compressor at some appropriate intermediate pressure In applications of this nature the blast-furnace gas could be used as a fuel, but in order to introduce it into the combustion chamber, its pressure would have to be increased. This could be accomplished by a separate compressor driven from the main shaft. Because of the low heating value of the gas, relatively large quantities would be required, resulting in an appreciable power consumption for the fuel compressor. The majority of this energy is recovered, however in the subsequent expansion through the turbine. It would be desirable to use a regenerator in the cycle so that the discharges from the main and fuel compressors could be preheated by the turbine exhaust gas prior to entry into the combustion chamber and thereby increase the thermal efficiency of the system.

The use of the conventional open-cycle gas turbine as a drive for automobiles has not appeared as yet. Such an application is technically possible, however, but the small sizes involved would not make it so attractive from a design standpoint as some of the larger units which have been built for other purposes. This may be readily understood when it is remembered that the net power available for useful purposes is the difference between two relatively larger powers. In the size range applicable to automobiles, it is difficult to maintain high efficiency in the turbine and compressor elements, so any variations would seriously affect the net output and thermal efficiency. In smaller sizes the free-piston compressor may be found to have desirable features, particularly with regard to efficiency.

Marine. The use of the gas turbine for marine service is one of the natural applications whereby its many advantages can be employed with benefit. The small space and weight characteristics of the gas turbine are particularly desirable for naval and maritime duty. The reversing operation when employing this type of unit aboard ship could be facilitated by the use of electric drive or a variable-pitch propeller, thus eliminating the necessity of providing a separate astern turbine. Astern operation with gear drive is a more acute procedure in the case of the gas turbine since, unlike the steam turbine, it normally exhausts at atmospheric pressure, thereby requiring a separate vacuum system to provide a low-density atmosphere in which to rotate the astern and ahead turbines during forward and reverse operation, respectively, so as to prevent prohibitive windage losses.

Prior to the Second World War, the Bureau of Ships of the United States Navy, recognizing the potentialities of the gas turbine, contracted with Allis-Chalmers to construct a plant designed for a temperature of 1500°F and an output of 3,500 hp This unit,[24] the largest stationary gas turbine for this temperature in the world, is shown in Fig. 8-128 on the test floor of the U.S. Naval Engineering Experiment Station at Annapolis, Maryland. As of this date, it has successfully completed its tests at temperatures up to and including 1500°F, the highest operating temperature ever attained in a prime mover of this size. The testing program will also include an investigation of the effects of wet compression.

The cycle on which the unit operates, depicted schematically in Fig. 8-129, consists of an axial-flow compressor C driven by turbine T_1, while a separate turbine T_2, arranged in parallel, furnishes the useful output. The air from the compressor enters the regenerator R where it is preheated by the combined exhaust gas from both turbines. It next passes to two combustion chambers B_1 and B_2 which furnish the high-temperature motive fluid to the turbines.

Referring to Fig. 8-128, the axial compressor may be seen in the lower left corner connected to its driving turbine. The power turbine is on the upper right in line with the other rotating elements. The regenerator in the upper left has three large pipes (white insulation) entering it which from left to right are the compressor discharge, compressor-turbine exhaust, and power-turbine exhaust. Directly under the regenerator are located the two combustion chambers from which may be seen emerging the turbine-inlet connections. The compressor intake air is supplied through the lower pipe in the figure.

The arrangement of the equipment may be seen in Fig. 8-130 where the nomenclature is identical with that of Fig. 8-129. A conception of the relative size of the various components may be obtained by noting the operator sketched in the figure. Since this unit was intended primarily for experimental laboratory purposes, no attempt was made to achieve minimum space and weight in the design. On the contrary, it was purposely extended laterally so as to facilitate the dismantling of the turbines and compressor as scheduled in the testing program The fulfillment of such a premise necessitated an elaborate piping system, requiring appropriate expansion and swing joints to accommodate the relative motions of the various components.

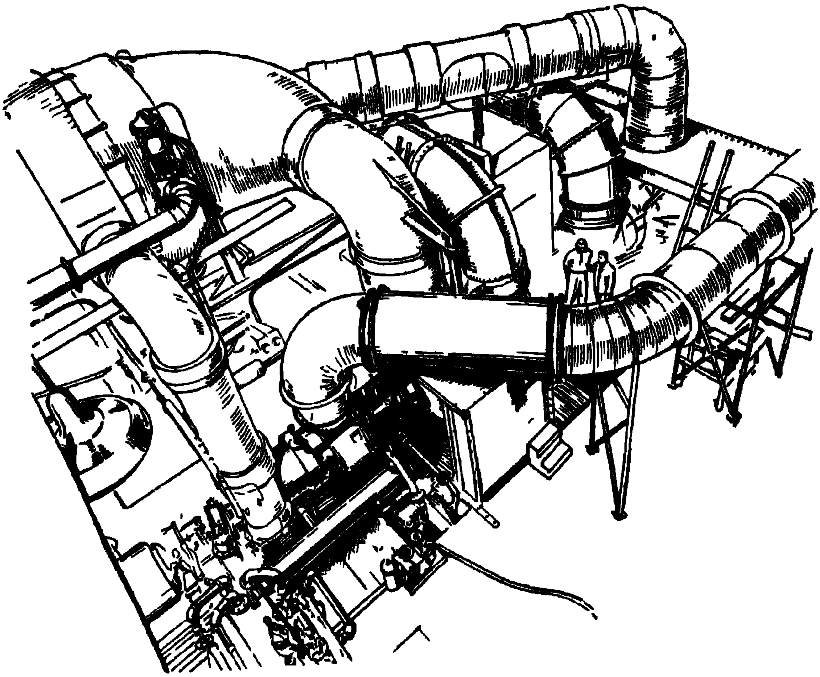


Fig 8-128 Allis-Chalmers 3 500-hp, 1500°F, experimental gas-turbine unit

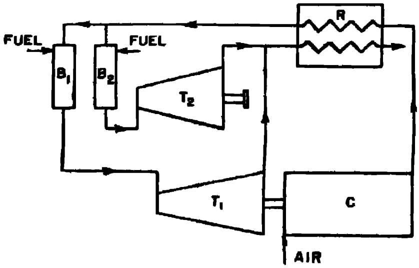


Fig 8-129 Cycle for Allis-Chalmers, 3 500-hp, experimental gas-turbine unit

T_1 = compressor turbine
T_2 = power turbine
C = compressor
B_1 = combustion chamber for compressor turbine
B_2 = combustion chamber for power turbine
R = regenerator

Dictated by the high temperatures involved, a unique design of turbine was adopted as indicated in Fig 8-131 where is shown a section through the compressor turbine In order to dispense with a bearing in the high-temperature end of the machine and thereby eliminate a potential source of trouble and also to provide optimum entrance conditions for the motive fluid, the turbine spindle was overhung

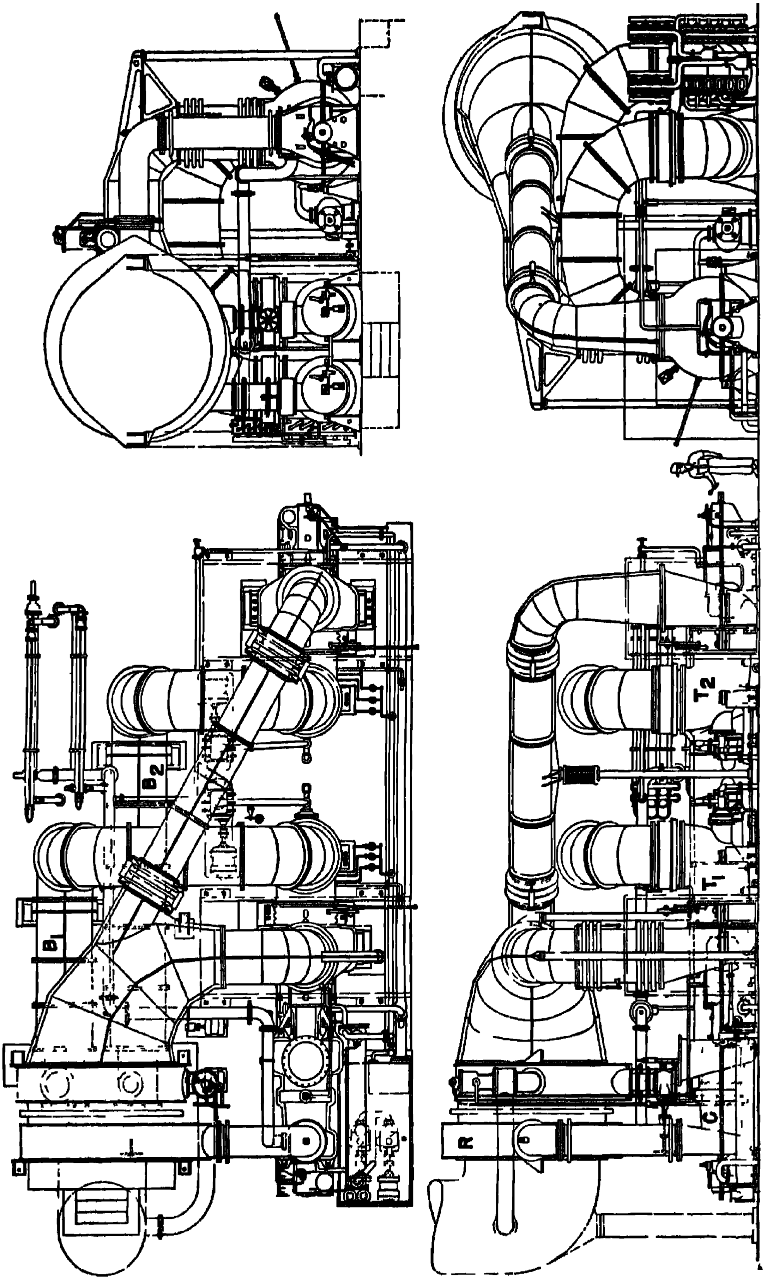


Fig 8-130 Arrangement views of Allis-Chalmers, 3 500-hp, experimental gas-turbine unit.

from a single sleeve bearing at its low-pressure end. The rotating element consists of two Rateau impulse wheels followed by three reaction stages. The individual impulse disks are joined to each other and to the single forging carrying the reaction blading by welding.

Several novel cooling features were employed to mitigate the effects of the exceptionally high temperatures involved. A separate recirculating cooling system directs cooled air on the periphery of the first disk so as to reduce its operating temperature. Air at reduced temperature is introduced radially at several locations in the casing so as to cool the first-stage nozzles and also the second-stage diaphragm, from which it impinges on the downstream side of the first disk. At still another point further down the casing air is admitted to cool the internal-reaction blade ring.

A by-pass is provided on the compressor turbine so that, during periods of starting and rapid acceleration, more capacity may be obtained by the motive fluid circumventing the first impulse stage. The by-pass valve is controlled externally by an air-operated valve. Except for the by-pass valve feature, reduced blade heights, and a modified balance piston, the power turbine is essentially a duplicate of the compressor turbine.

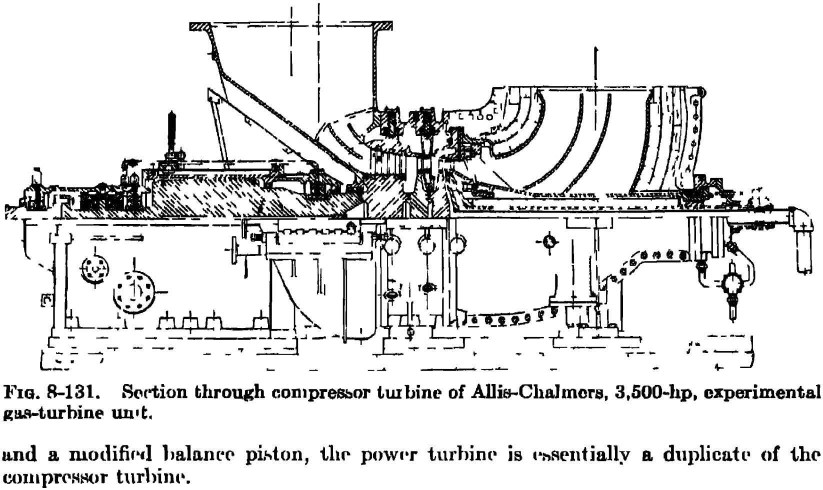


FIG. 8-131. Section through compressor turbine of Allis-Chalmers, 3,500-hp, experimental gas-turbine unit.

The Elliott Company, under the sponsorship of the Bureau of Ships, manufactured a 2,500-hp marine gas-turbine plant[12] which is shown in Fig. 8-132. The unit is designed for a turbine temperature of 1200 F. A novel feature is the use of a Lysholm positive-displacement type of compressor which has the advantage of a flat efficiency characteristic, although its peak efficiency is not so high as that of the axial compressor.

A schematic diagram of the regenerative cycle with reheating and intercooling is given in Fig 8-133 Atmospheric air enters the low-pressure compressor C_1 from which it is discharged to the intercooler I where its temperature is reduced. After next traversing the high-pressure compressor C_2, it passes through the regenerator R where it absorbs heat from the turbine exhaust gas. Fuel is burned with the heated air in combustion chamber B_1, and the resulting high-temperature gas drives the high-pressure turbine T_1 which furnishes power to the low-pressure compressor. The exhaust gas from the high-pressure turbine, still containing adequate oxygen to support combustion, has its temperature increased by the burning of additional fuel in combustion chamber B_2. The ensuing products of combustion, at a temperature approximately the same as that at the inlet to the high-pressure turbine, propel the low-pressure turbine T_2, which has the dual function of driving high-pressure compressor C_2 and also furnishing the useful shaft output. The exhaust gas from the low-pressure turbine, after disposing of heat in the regenerator, is discharged to the atmosphere.

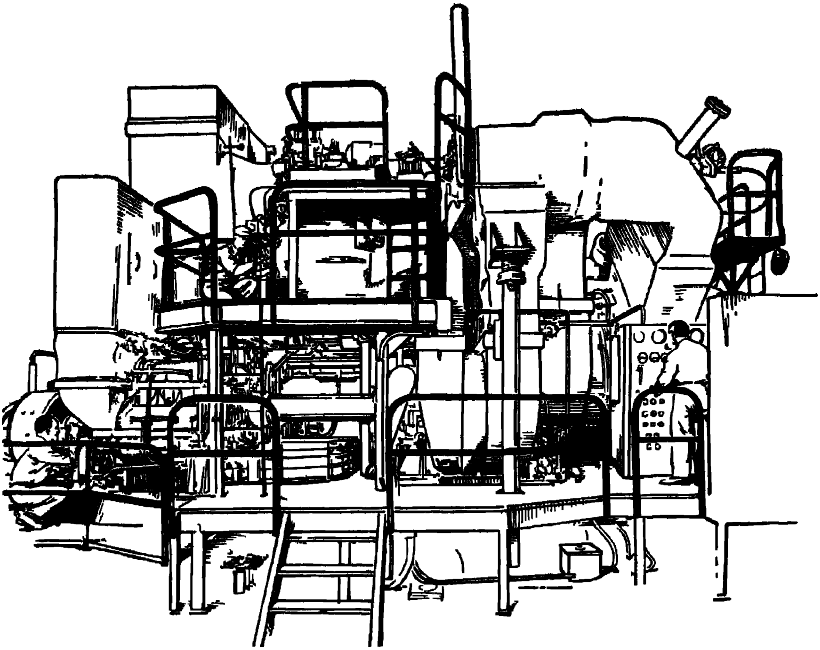


FIG 8-132 Elliott, 2,500-hp, marine gas-turbine plant.

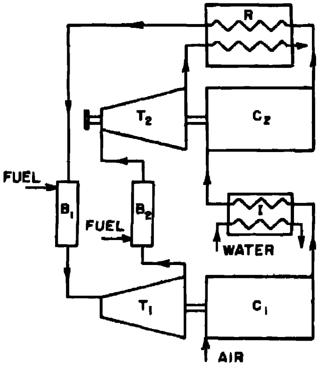


FIG 8-133 Cycle for Elliott, 2,500-hp, marine gas-turbine plant

T_1 = high-pressure turbine
T_2 = low-pressure (power) turbine
C_1 = low pressure compressor
C_2 = high-pressure compressor
B_1 = high-pressure combustion chamber
B_2 = low-pressure combustion chamber
R = regenerator
I = intercooler

Based on the lower heating value of the fuel, tests on the plant showed coupling thermal efficiencies of approximately 29, 28, 26, and 20 per cent at full, three-fourths, one-half, and one-fourth load, respectively

Rail. In all probability the most desirable application of the gas turbine is in the heavy-traction field When used as a locomotive drive, its many advantageous characteristics place it in an exceptionally favorable position when compared with conventional types. When operating on a regenerative cycle at 1300°F, a rail thermal efficiency of 17 per cent is readily attainable with the gas turbine. Even with liquid fuel, because of the low-grade type it can use, fuel costs for the gas turbine would be lower than for the majority of conventional steam locomotives burning coal and at least equivalent to the best where an efficiency of 9 per cent is rarely exceeded. It would also have lower fuel costs than the diesel locomotive whose better thermal efficiency is offset by the higher grade of fuel it uses. When coal fuel is used in the gas-turbine cycle, the comparison becomes even more favorable.

Since no water is required in its operation, the gas-turbine cycle should be particularly attractive for locomotive installations. Elimination of the now-necessary water-treating provisions and frequent inspections, cleanings, and repairs of boilers used on present steam locomotives should certainly be desirable. Of additional importance would be the improved operating schedules possible on runs that formerly made stops for water.

The large amount of excess air used in the gas-turbine combustion process permits a clear stack completely free from smoke at all loads on the unit. The desirability of such a feature in view of some present city ordinances is manifest.

The purely rotary motion of the gas turbine will result in minimum maintenance and vibration. Absence of any reciprocation, with its concomitant unbalanced forces, is obviously beneficial. Since the elements of the gas turbine are similar to those of steam turbines, maintenance after development should be essentially of a moderate order except for such additional incremental amount as might result from the higher temperatures employed. Operating experience with the Houdry turbines of similar design indicates very low maintenance for this equipment which operates at temperatures in the range of those employed in modern high-temperature central-station practice. Several of the oil-refinery gas turbines have demonstrated an exceptionally high order of reliability by operating continuously for as long as 2 years without shutdown.

Lubrication costs of the gas turbine, there being no sliding surfaces except for journal bearings, should not exceed 1 per cent of the fuel costs. Such costs on present railway equipment are substantially in excess of this figure.

When dynamic braking is considered, the application of the gas turbine to locomotives is exemplary. By operating the traction motors as generators, the motorized main generators may be loaded by driving the gas turbine and its associated compressor. The energy required during compression can be dissipated by discharging the compressed air to the atmosphere. With such an arrangement, the electrical-resistor grids normally required for dynamic braking are eliminated.

In 1941, Brown-Boveri completed a 2,200-hp gas-turbine electric locomotive[23] for the Swiss Federal Railways. A shop view of the gas-turbine power plant is shown in Fig. 8-134. From left to right may be seen the 200-kw auxiliary d-c generator, main d-c generator, reduction gear, regenerator, gas turbine, and vertical combustion chamber. The locomotive has a maximum specific service weight, including fuel, of 92 lb per bhp. Its maximum speed is 70 mph. The tractive effort at starting is 29,000 lb. Acceptance tests of the power plant showed a maximum coupling thermal efficiency at three-fourths load of 18 per cent based on the lower heating value of the fuel.

Based on the experience gained with their first locomotive, Brown-Boveri has recently completed a 4,000-hp gas-turbine power plant for another locomotive.[24] This new unit, shown in Fig. 8-135, is reported to have substantially higher efficiency than the first power plant, and notwithstanding its greater capacity, its physical size is not much larger than its predecessor. The improvement in efficiency and capacity is accomplished by incorporating a device referred to as a "comprex," whereby the over-all pressure ratio of the cycle is increased.

The Locomotive Development Committee[11] is now engaged in a research program which embraces the greatest potentialities regarding the application of the gas turbine as a railway power plant. This committee was formed as an agency of Bituminous Coal Research, Inc., in the latter part of 1944. The committee is composed of the chief executives of six leading coal-hauling railroads and three major coal companies. It was established for the purpose of developing a bituminous-coal-burning locomotive

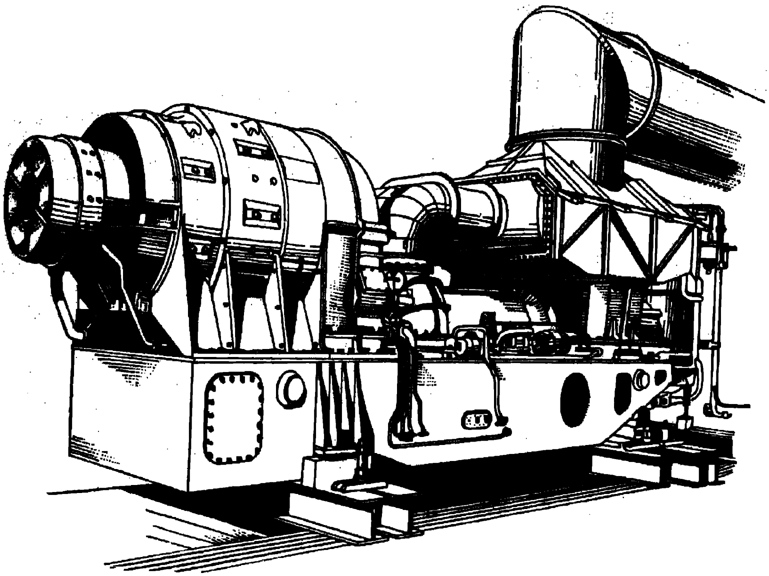


FIG. 8-134. Brown-Boveri 2,200-hp locomotive gas-turbine unit.

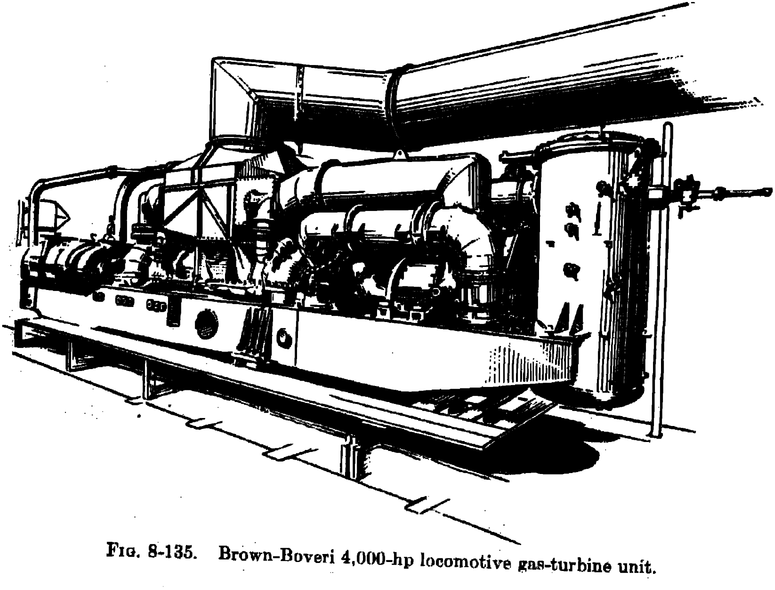


FIG. 8-135. Brown-Boveri 4,000-hp locomotive gas-turbine unit.

which could compare favorably with the diesel locomotive. In the fulfillment of this aim, the gas turbine has been selected as the prime mover which best adapts itself to the requirements involved.

The proposed arrangement of the equipment to be used on the coal-burning gas-turbine locomotive is depicted schematically in Fig. 8-136. The coal is fed from bunker *A*, by means of a motor-operated screw, to crusher *D* where its size will be reduced to −16 mesh. Turbine exhaust gas will be used for drying the coal. By means of fan *E* the coal is conveyed to pressure tanks *F* and *H* after it is crushed. The twin-tank system of coal storage is necessary because of the different pressure levels in the cycle. While one tank is being replenished, it is isolated by means of valves from the high-pressure circuit, and coal is used from the other tank under full pressure.

For instance, when tank *F* was being filled, valves 1 and 2 would be open, and 3 would be closed. The tank would then be under the low pressure existing in the conveying system. The crushed coal mixed with its conveying air enters the tank where an internal cyclone separates the coal from the air, the latter being vented. After tank *F* is filled and tank *H* exhausted, valves 1, 2, and 6 are closed and valves 3, 4, and 5 opened, whereupon the full pressure required for pulverizing exists in tank *F*.

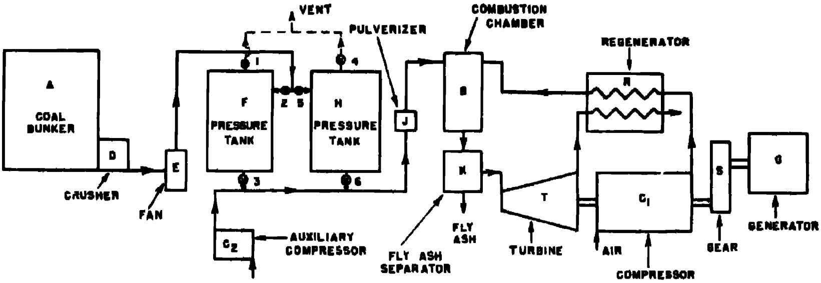


FIG. 8-136. Cycle for coal-burning locomotive gas turbine proposed by Locomotive Development Committee.

The air compressor C_2 will produce a pressure of approximately 140 psig maximum required for pulverization. The intake will be connected with the discharge of the main compressor C_1, and a variable-speed drive of C_2 will permit independent pressure relationships. After the crushed coal from tanks *F* or *H* is picked up by the high-pressure air, it is led to the inlet of the pulverizer *J*. This latter device was developed at the Institute of Gas Technology of the Illinois Institute of Technology in Chicago and was used to produce finely pulverized coal for gasification. It consists of a nozzle through which the compressed air, laden with crushed coal, is expanded to a pressure essentially equivalent to that at the turbine inlet, which will be approximately 60 psig at full load. Expansion through the nozzle imparts a high velocity to the air-coal mixture, which subsequently impinges on a target and is finely pulverized by impact and attrition. With proper design, a product as fine as 80 to 90 per cent of −325 mesh and 5 per cent or less of +100 mesh is obtainable with an air consumption of approximately one pound per pound of coal. Because of its extremely small size and effective pulverization, the adoption of this device appears to be the logical procedure for applications where space is at a premium, as in the case of locomotives.

The pulverized coal and primary air next enter combustion chamber *B* where they mix with the preheated air from the regenerator. The combustion chamber will be basically similar to that shown in Figs. 8-95 and 8-100, comprising an inner shell, where burning will occur with about 40 per cent excess air, surrounded by an annular space through which flows the diluent air. After mixture, the temperature of the gas entering the turbine will be 1300°F. Since the products leaving the combustion chamber contain fly ash, which would seriously erode the turbine blading, it is necessary to introduce an ash separator *K*. This device will consist of a series of is arranged at each end of the locomotive cab, permitting the weight to be more uniformly distributed, with most of it directly over the trucks. For each power unit, the common frame carrying the turbine, compressor, reduction gear, and generators is supported at three points for the purpose of maintaining correct alignment. Two of the support points are located laterally directly over the center axle of each truck, while the third, in vertical line with the No. 1 turbine bearing, is on the center line and near the center of the cab.

The filters for the intake air to the gas-turbine–compressor units are located on both sides of the cab near the top at each end. They are placed as high as feasible so as to reduce the possibility of disturbed foreign objects along the roadbed being inducted. Furthermore, the high intakes are desirable when the locomotive is standing in a station with engines running where passengers may be in the vicinity.

The exhaust gases leaving the heat exchangers are discharged through the roof of the locomotive.

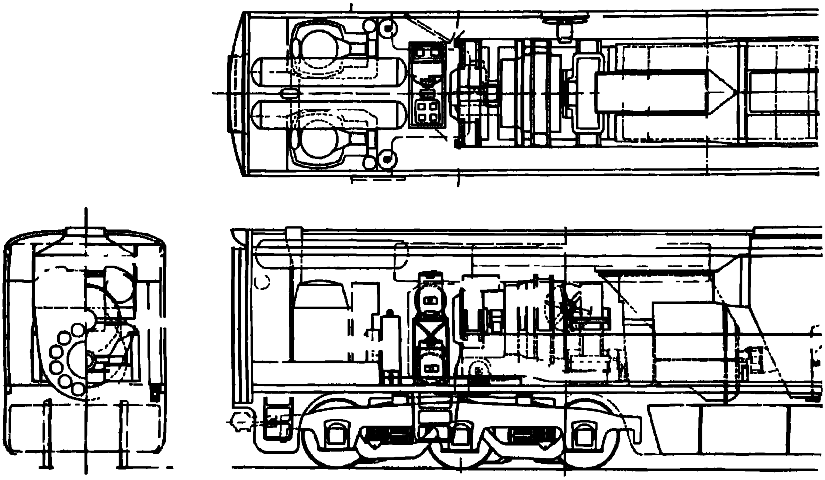


Fig. 8-138. Arrangement of Allis-Chalmers proposed

Two 2,250-lb-per-hr steam generators are provided for train heating. It is possible, with the gas-turbine type of drive, to install a waste-heat boiler in the exhaust passage beyond the heat exchanger and generate steam for heating purposes. However, in the interest of simplicity, it was decided to forego such a refinement on an initial gas-turbine installation, and therefore only the conventional type of boiler is indicated. After a sufficient accumulation of operating experience has proved the merit of the gas turbine itself, it would in all probability be expedient to incorporate in future locomotive designs a waste-heat means of generating heating steam also.

An auxiliary-engine generator set of 150 hp is provided for starting the two gas-turbine power plants by motorizing the main generators. One of the numerous advantages of having two smaller power units instead of a single larger one of the same aggregate output is apparent in this instance, since it permits smaller capacity starting equipment to be employed. By connecting the auxiliary set directly to the traction motors, it may also be used for propelling the locomotive up to speeds of approximately 15 mph, thus enabling the main units to remain shut down when it is desired to move the locomotive about the yards.

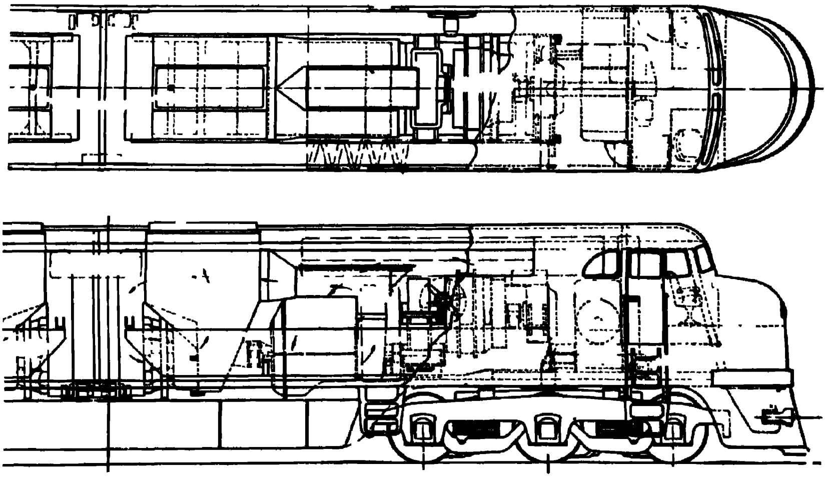
Since the use of a relatively heavy low-grade fuel is contemplated, provision is made for fuel heating. There are various methods of accomplishing this, either with steam or electrical heating, but probably the most logical would be to use the exhaust gas from the turbines. For starting purposes, the light auxiliary-engine fuel, requiring no preheating, could be used temporarily in the combustion chambers of a main power unit until the flow of exhaust gas is established.

In the space under the center of the cab between the two trucks, tank capacity is provided for 4,000 gal turbine fuel oil, 1,500 gal lubricating oil, and 2,600 gal water for train heating. The fuel-oil capacity is sufficient for approximately a 10-hr run. Two 300-gal tanks in the rear of the cab contain fuel for the auxiliary engine and the steam generators.

No water is required for the gas-turbine power plants themselves, as all cooling is done with air, two fan coolers being used for the lubricating oil and blowers for cooling the generators and traction motors.

Train controls and associated electrical equipment are in the forward end of the locomotive.

A section through one of the duplicate gas-turbine power units proposed for the locomotive is shown in Fig. 8-139. A five-stage reaction-type gas turbine *A* drives a 22-stage axial compressor *B*. Filtered air from the atmosphere traverses the compressor where its pressure is increased. It next passes through the cross-flow heat exchanger *C*, where its temperature is raised by the turbine exhaust gas. The heated air then flows in parallel through a group of 12 separate combustion chambers *D* ahead of the turbine. Part of the air is used for combustion purposes in an inner shell, while that remaining flows through the annular space between the inner and outer shells and cools the products of combustion to a satisfactory turbine-inlet temperature. A series of smaller combustion chambers is used instead of a single large one, as it has been found that in certain instances more satisfactory combustion at higher heat release can be obtained in this manner. The gas, a mixture of air and combustion products, then expands through the turbine from which it is exhausted to the atmosphere through the heat exchanger where it preheats the air from the compressor. The power developed by the turbine is greater than that required by the compressor, and the excess power is supplied through the reduction gear *E* to the main generator *F* and the auxiliary generator *G*.



4,800-hp gas-turbine locomotive with electric transmission.

In order to start the unit from a standstill, the generator is switched from shunt to series field and motorized by an auxiliary-engine generator set, bringing the unit up to about 30 per cent of maximum operating speed, at which point the turbine, at full gas temperature, is capable of driving the unit by itself. The full-load speed of the turbine and compressor is 5,000 rpm, and that of the generators is 900 rpm.

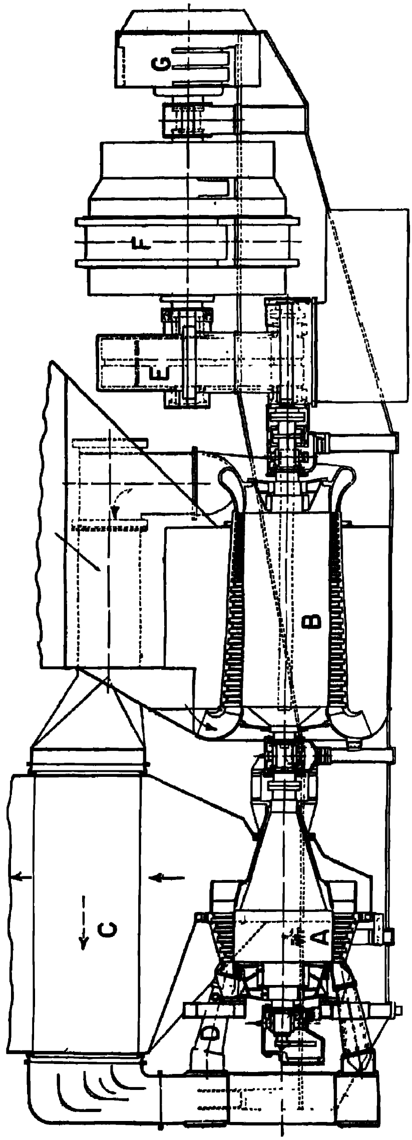


Fig. 8-139. Section through Allis-Chalmers proposed 2,400-hp locomotive gas-turbine power plant.

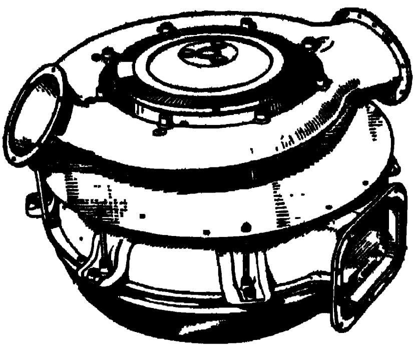


FIG. 8-140. Gas-turbine supercharger.

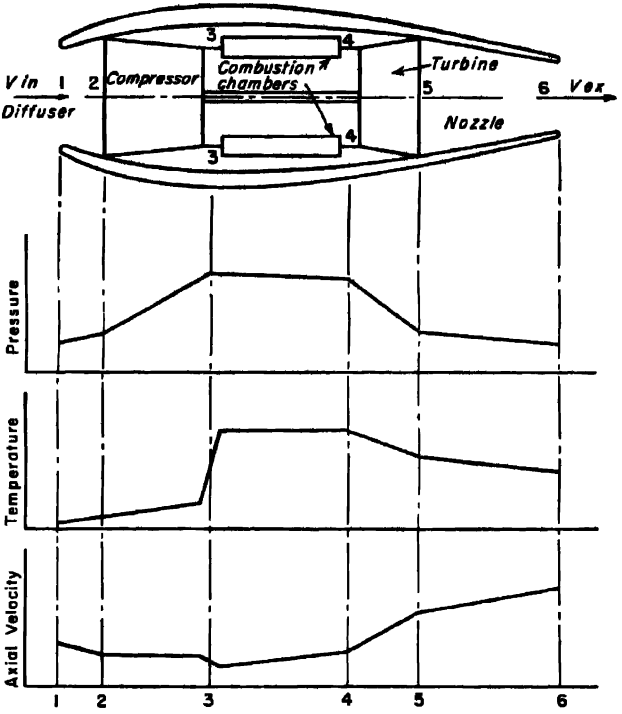


FIG. 8-141. Schematic arrangement of jet-propulsion unit showing variation of pressure, temperature, and axial velocity.

The turbine and compressor, supported on a common frame with the gear and generators, are arranged as a three-bearing unit.

The cross-flow, tubular-type heat exchanger has a surface of 1.4 sq ft per useful horsepower output of the gas-turbine power plant. The effectiveness of the heat exchanger is 50 per cent.

Aircraft. Vast numbers of high-speed, lightweight gas turbines were manufactured during the war and used in aircraft turbosuperchargers. These units operating at 24,000 rpm with a turbine-inlet gas temperature of 1600°F were designed and built by General Electric. Under license arrangement they were also manufactured by Allis-Chalmers and the Ford Motor Company. The combined production of the three companies totaled many millions of horsepower. Figure 8-140 shows one of these turbosuperchargers with the gas turbine on the left and the centrifugal compressor on the right. The exposed turbine wheel may be clearly seen. These units were used on such aircraft as the B-17 (Flying Fortress), the B-24 (Liberator), the B-29 (Superfortress), and the P-38 (Lightning).

Current press releases have publicized the adaptation of the gas turbine to the jet propulsion of airplanes. Figure 8-141 depicts schematically the elements comprising a jet-propulsion unit, as well as the variation in pressure, temperature, and axial velocity occurring throughout the device Referring to the figure, the compressor is driven by the gas turbine, the latter receiving high-temperature motive fluid from combustion chambers interposed between the compressor discharge and the turbine inlet. The residual kinetic energy of the gas leaving the turbine enhanced by a subsequent expansion provides the propulsive jet for propelling the aircraft. The relatively high fuel consumption of the pure-jet type of unit in its present form depreciated its attractiveness for other than military aircraft, although the British have built several transports with turbo-jet engines. Modifications of the current type having improved thermal efficiency and also propeller-jet and reciprocating-engine-gas-turbine combinations, will probably enjoy more diversified future applications.

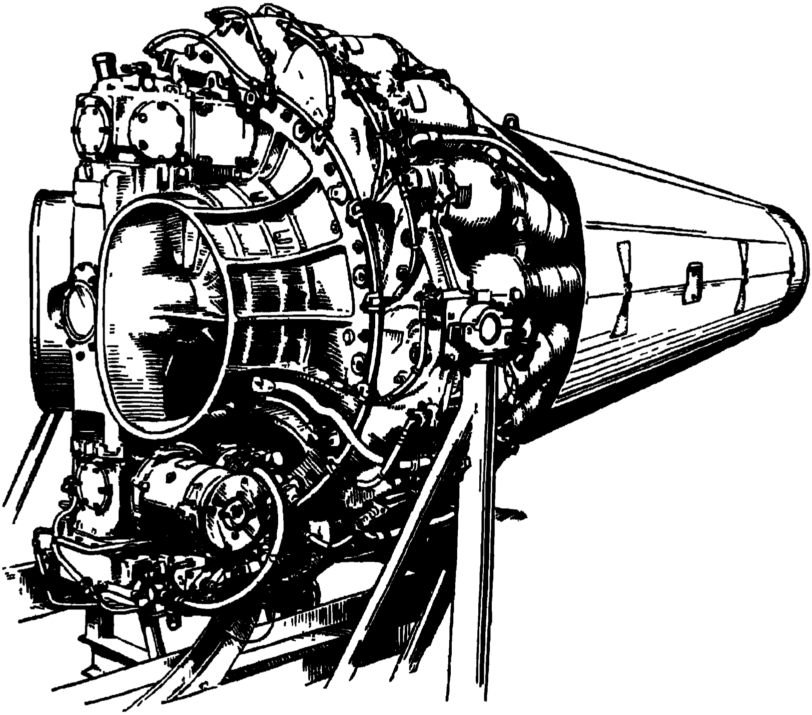


FIG. 8-142. Allis-Chalmers de Havilland 3,000-lb thrust jet-propulsion unit.

All three of the large turbine builders in this country have been engaged in the design and manufacture of aircraft jet-propulsion units Allis-Chalmers, while having several designs of its own which have not yet reached the manufacturing stage, built jet-propulsion units during the war according to the designs of the de Havilland Engine Co., Ltd., of England. Figure 8-142 shows an external view of one of these engines. This design employs a single-stage centrifugal compressor, with single-sided entry of air driven by a single-stage turbine. The unit develops a static sea-level thrust of 3,000 lb when rotating at 10,500 rpm with a turbine-inlet temperature of 1500°F. The complete weight with all auxiliaries is 1,500 lb, resulting in a specific weight of 0.5 lb per lb static thrust. The fuel consumption is approximately 1.2 lb per hr per lb static thrust. The maximum over-all diameter is 50 in.

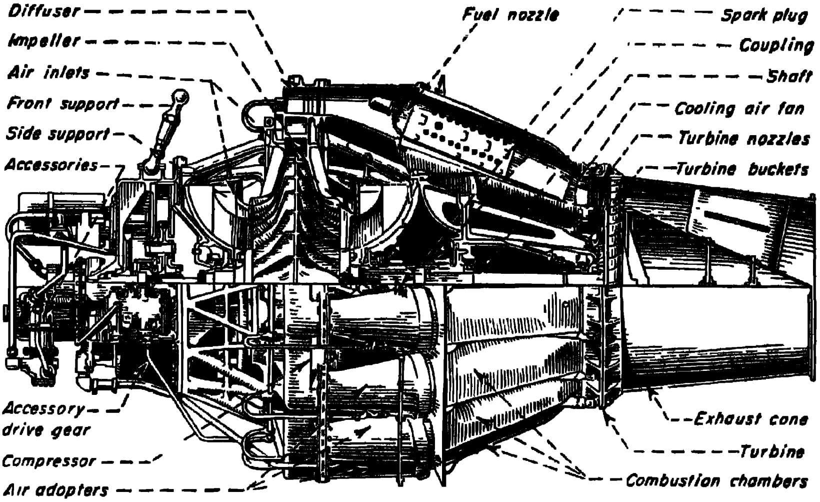


Fig. 8-143. General Electric 4,000-lb thrust jet-propulsion unit.

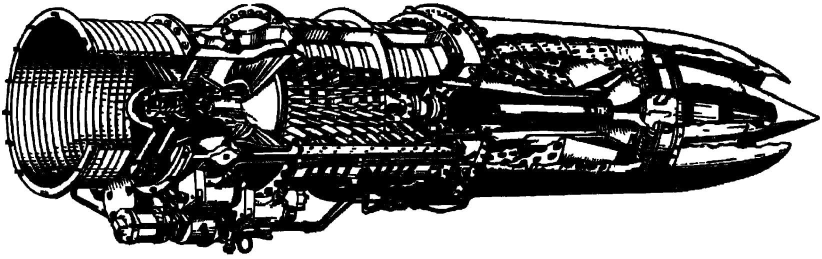


Fig. 8-144. Westinghouse 1,600-lb thrust jet-propulsion unit.

The General Electric Company originally built jet-propulsion engines according to British designs but later developed a unit of higher rating known as the I-40. This unit, shown in Fig. 8-143, while retaining some of the features of the prototype designs, has a sea-level static-thrust rating of 4,000 lb at a rotative speed of 11,500 rpm and a turbine-inlet gas temperature of approximately 1500°F. The specific fuel consumption under these conditions is about 1.2 lb per hr per lb static thrust. The average specific weight of the unit is about 0.45 lb per lb static thrust. The engine has a single-stage centrifugal compressor, with air entry on both sides of the impeller, driven by a single-stage gas turbine. The maximum over-all diameter is 48 in. The double-entry feature has the advantage of reducing the over-all diameter of the unit, although it complicates inlet-flow conditions on the rear side of the impeller.

General Electric also has a pure jet-engine design, designated as TG-180, employing an axial-flow compressor. Another unit, the TG-100, is a propeller-jet combination again with a compressor element of the axial-flow type. The axial-flow compressor has the advantage of higher efficiency and smaller diameter, the latter resulting in reduced drag of the airplane.

The Westinghouse Electric Corporation has designed and built its X19XB-2B jet-propulsion engine employing a 10-stage axial compressor driven by a single-stage turbine. The unit, shown in Fig. 8-144, produces a static sea-level thrust of 1,600 lb when rotating at 17,000 rpm with a maximum turbine-inlet temperature of 1500°F. The specific weight is 0.45 lb per lb static thrust. The diameter is 19 in. The fuel consumption is 1.15 lb per hr per lb static thrust.

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COMPRESSORS AND PUMPS

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COMPRESSORS

Uses of Compressed Air

Compressed air has a wide variety of applications Following are some of its uses:

1 To operate reciprocating tools, such as riveting hammers, chipping hammers, scalers, paving breakers, diggers, tampers, etc.

2. To operate rotating tools, such as grinders, wrenches, screw drivers, air motors, drills, reamers, air-operated pumps, etc.

3. To spray paint, oils, insecticides, etc.

4. To operate air pistons for presses, opening doors, lifting, clamping, etc.

5. Air jets for cleaning, moving packages, etc.

6. To inflate tires and casings of all types, to charge shock absorbers, etc.

7. To start diesel engines

8. To operate control instruments, valves, etc.

9 To aerate and agitate liquids.

The above list is far from complete, but it indicates that compressed air may be utilized to produce almost any mechanical motion. Air-operated tools and devices of many types are available in standard sizes

Types of Air Compressors

Compressed air is air from the atmosphere compressed or squeezed into a small space The extent of the squeezing determines the pressure attained Machines for compressing air are of several types:

1 Reciprocating compressors have a piston moving back and forth in a cylinder. This is the most widely used type of compressor.

2 Turbo- or centrifugal compressors have a bladed impeller which rotates within a casing Such units are normally used for low and moderate pressures up to, say, 40 psi.

3. Rotary compressors are of several types: (*a*) two-impeller, with intermeshing gear-type elements, (*b*) sliding vanes, with a vaned rotor eccentrically mounted in a casing, (*c*) liquid-piston, with a multibladed rotor revolving in a casing partly filled with liquid which acts as a piston.

4. Axial-flow compressors have a bladed rotor or rotors and bladed stationary vanes similar to a reaction steam turbine.

Of the above types the reciprocating compressor has the most general application for the operation of laborsaving tools. The other types, while overlapping the field of the reciprocating compressor to a certain extent, are mainly used where air is required at moderate pressures.

In most plants the general air supply is maintained at 100 psi pressure. Machines for higher or lower pressures are also available. Standard units are supplied for pressures as high as 3,000 psi, and special units for higher pressures have been built.

While we have been speaking in terms of air compressors, the same type of machine is also used for compressing gases of various types. Where a machine is needed for such purposes, a reliable manufacturer should be consulted as special materials and design features may be required depending on the nature of the gas.

Mention should also be made of vacuum pumps. These machines, except air ejectors, are essentially the same as an air compressor except that they operate in a pressure range below atmospheric pressure instead of above.

Selection of a Compressor

A number of factors are involved in the selection of a suitable air compressor or compressors for a plant.

Pressure. First of all, the pressure needed must be determined. Most air-operated tools are designed to operate at pressures from 90 to 100 psi. If this is the case, a 100-lb compressor would normally be selected, and the air-distribution lines laid out to assure a pressure of 90 psi or greater at the tools. Where long distribution lines are required, it may be desirable to install a machine discharging at 110 or 125 psi.

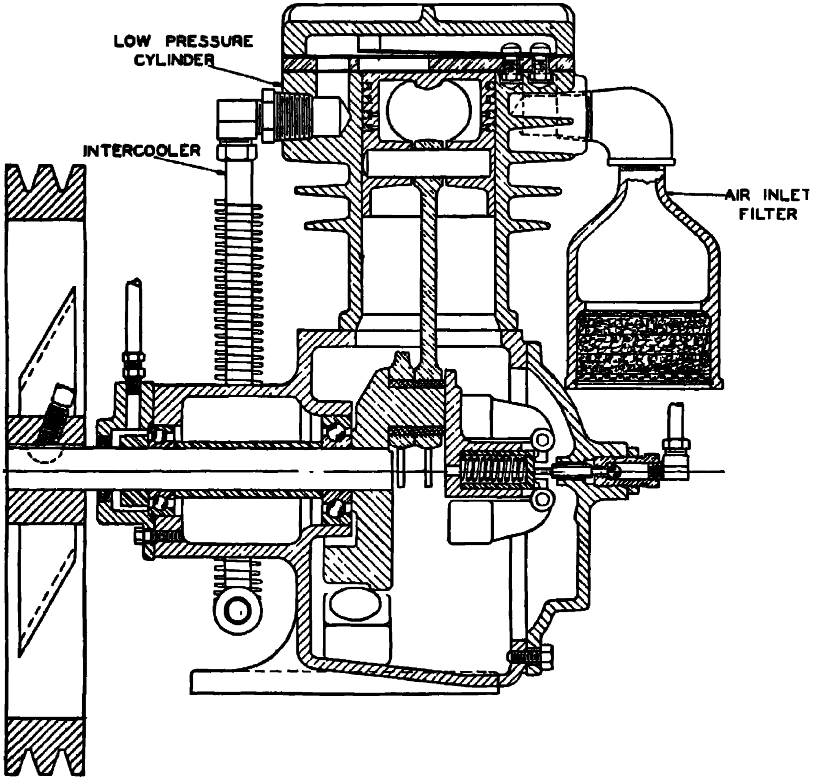


FIG. 8-145. Section through one cylinder of a two-stage air-cooled compressor

Where one or two operations require air at a higher pressure, it is usually more economical and more convenient to install a separate small compressor to furnish air for these operations.

Where small amounts of air are required at pressures lower than that carried in the main distribution lines, they may be obtained by installing a reducing valve in the branch line leading to the operation requiring the low-pressure air. If large amounts of low-pressure air are required, it is more economical to install a separate compressor for the purpose.

Where the air pressure is less than 30 psi and the volume required is comparatively large, a turboblower or rotary-type compressor should be considered.

Capacity. Another important factor in compressor selection is the capacity or volume of air required. This is a factor which is sometimes extremely difficult to evaluate. Obviously the unit selected should be large enough to supply all the air devices which will be operating at any one time. If all the air operations are continuous, the capacity required is simply the sum of the air consumption of each individual tool. In most plants, however, air tools such as chipping hammers, grinders, hoists, etc., are operated intermittently. The compressor capacity in this case is that required to operate as many air-consuming devices as would ever be in use at one time. This may be anywhere from 10 to 100 per cent of the total required by all the tools, depending almost entirely on the nature of the work in the plant. The experience of another plant doing similar work is very helpful. Air-compressor and air-tool manufacturers can often aid in determining the air capacity required. Tables 8-49 and 8-50 will be useful in determining horsepower requirements once capacity has been determined.

Capacity Control. In most installations the demand for air is irregular, and therefore some means of controlling the output of the compressor is necessary. The choice of control depends upon the character of the air load.

Practically all compressor capacity or unloading devices (Fig. 8-151), as they are usually termed, are actuated by the pressure on the discharge side of the compressor. A falling pressure indicates that air is being used faster than it is produced, and more air is required. A rising pressure indicates that more air is being produced than is being used, and therefore less air is required.

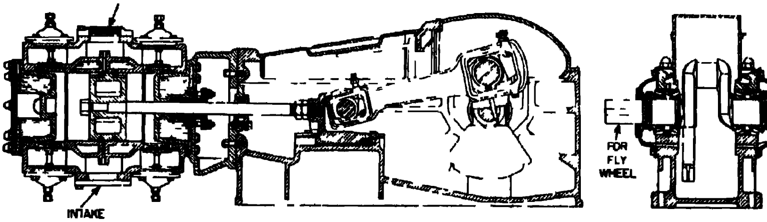


Fig. 8-146 A typical single-stage water-cooled compressor.

An obvious method of controlling a compressor's capacity is to vary the speed. This method is applicable to steam-driven reciprocating and to rotary compressors and, within certain limits, to centrifugal compressors and blowers. It is also applicable to some units driven by internal-combustion engines. In these cases the pressure regulator actuates the steam-admission or fuel-admission valve on the compressor driver and thus controls the speed.

By far the greatest number of compressors, however, are motor-driven, and since a motor is essentially a constant-speed machine, other methods of controlling the capacity are necessary. On reciprocating compressors up to about 100 hp, two types of control are usually available. These are automatic start-and-stop control and constant-speed control.

Automatic start-and-stop control, as its name implies, stops or starts the compressor by means of a pressure-actuated switch as the air demand varies. It should be used when the demand for air will be intermittent.

Constant-speed control should be used when air demand is fairly constant. With this type of control, the compressor runs continuously until shut down but compresses air only when it is needed. Two methods of unloading the compressor with this type of control are in common use: (1) closed-suction unloaders and (2) open-inlet-valve unloaders. The closed-suction unloader consists of a pressure-actuated valve in the compressor intake which prevents compression by shutting off the air intake. Open-inlet-valve unloaders operate to hold the compressor inlet valves open and thereby prevent compression.

It is often desirable to have a compressor equipped with both automatic start-and-stop and constant speed control. When this is done, a switch allows immediate manual selection of either type.

Motor-driven reciprocating compressors above about 100 hp in size are usually equipped with a step control. This is in reality a variation of constant-speed control in which unloading is accomplished in a series of steps, varying from full load down to no load. Three-step control (full load, one-half load, and no load) is usually accomplished with free-air unloaders. Five-step control (full load, three-fourths load, one-half load, one-fourth load, and no load) is accomplished by means of clearance pockets. These pockets or small air reservoirs are opened when unloading is desired. The air is compressed into them on the compression stroke of the compressor piston and refills the cylinder on the return stroke, thus preventing the compression of additional air. On some makes of machines open-inlet-valve and clearance-control unloading are used in combination.

Air-cooled and Water-cooled Reciprocating Compressors. When air is compressed, considerable heat is generated. This heat must be dissipated at least in units where the pressures exceed 25 or 30 psi. On air-cooled machines (Fig. 8-145) this is accomplished by means of fins on the cylinders and a fan which blows air across the fins and also the intercooler surfaces if the machine is of the multistage type. On water-cooled machines (Fig. 8-146), water is circulated in jackets (Fig. 8-150) surrounding the cylinders to pick up and carry away the excess heat. On multistage water-cooled machines, water is also circulated in the intercooler tubes.

In general, air-cooled machines are available for pressures up to 125 psi and horsepowers up to about 100. Some air-cooled units of small size, 5 or 10 hp, are available for considerably higher pressures. Water-cooled units are available from 5 hp on up. Large units are invariably water-cooled.

In the range where both air-cooled and water-cooled machines are available, either or both types may be satisfactory, depending on conditions Air-cooled units are convenient where there is danger of freezing, in isolated locations, and where the expense of attendance or automatic water shutoff valves is not warranted. Water-cooled machines should be used where air temperatures are high or where the heat from the compressor is objectionable.

Single-stage and Multistage Reciprocating Compressors. Single-stage compressors are those in which compression from intake to discharge pressure takes place in a single step; in a reciprocating compressor, it takes place in a single stroke of the piston.

Multistage compressors (Figs. 8-147, 8-148, and 8-149) are those in which compression takes place in two or more distinct steps or stages; in a reciprocating compressor, the succeeding steps usually take place in separate cylinders.

Multistage reciprocating compressors and many multistage machines of other types have intercoolers between stages. These are heat exchangers usually designed to reduce the temperature of the compressed air to near the temperature at which it entered the first stage. This reduction in temperature results in a distinct power saving.

Single-stage air-cooled compressors for intermittent service are available for pressures up to 150 psi in sizes up to about 3 hp.

Two-stage air-cooled compressors (Fig. 8-145) are available for heavy-duty service for 125 psi pressure in sizes up to 100 hp. Two- and three-stage air-cooled units for pressures as high as 3,500 psi are available in sizes below 10 hp.

Single-stage water-cooled compressors (Fig. 8-146) are available for 100 psi pressure in sizes up to 100 hp and for slightly higher pressures at lower horsepowers. Large units (Figs. 8-147, 8-148, and 8-149) are almost invariably of the multistage water-cooled type for pressures above 60 to 80 psi.

Compressor Efficiency. When selecting a compressor, the first requisite is, of course, to choose one that fits the job in hand as regards capacity, pressure, type of control, etc. After that the question of operating efficiency is likely to be considered. So many different efficiency terms are used in connection with air compressors that they become confusing unless one is thoroughly familiar with them. A few of these terms are defined in Table 8-52. When comparisons are made between machines, it

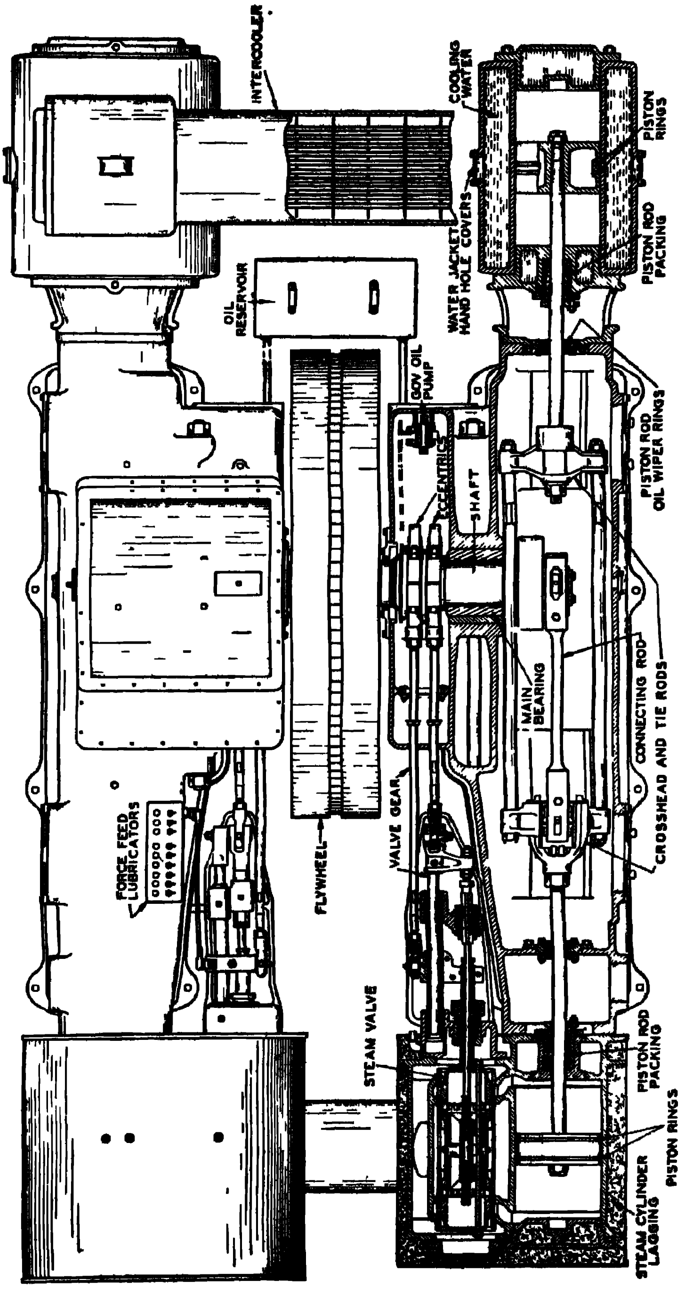


Fig. 8-147. A two-stage steam-driven compressor.

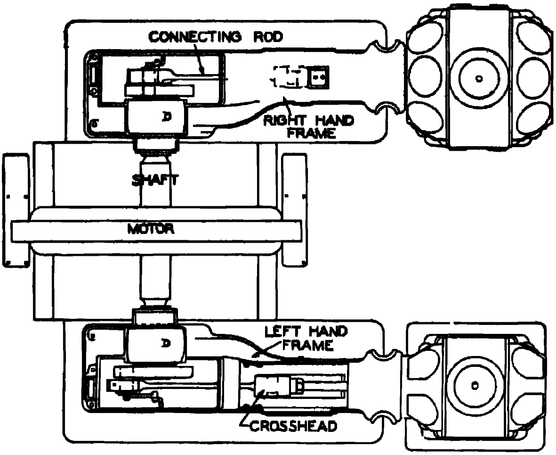


FIG 8-148. View of a duplex compressor looking down from the top

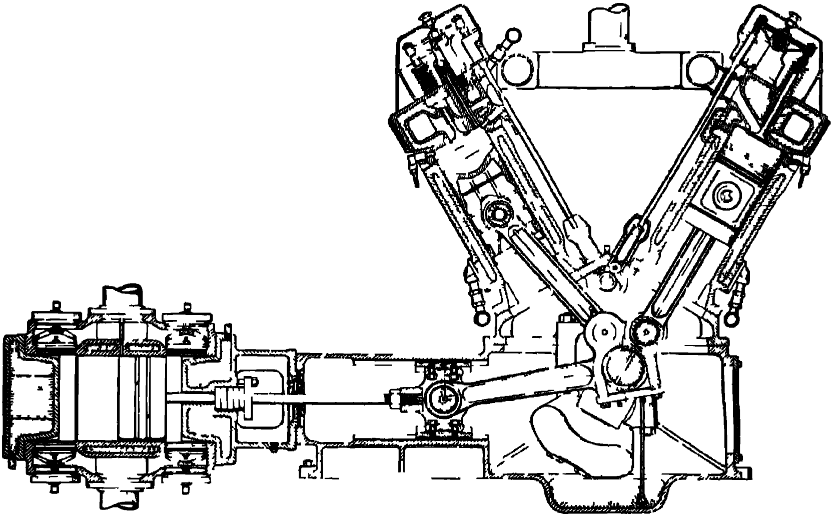


FIG 8-149 A direct-connected diesel-engine-driven compressor

is simpler and less confusing if a basic air delivery and power consumption are used Such measurements for various types of compressors might be

Actual air delivery of, say, 100 cu ft } per {
kwhr (electric-motor drive)
gal of gasoline (gasoline-engine drive)
gal of fuel oil (diesel-engine drive)
lb of steam (steam-engine drive)

If machines with different types of drive are being compared, it is necessary to reduce them all to a common factor by assigning cost values to the different power mediums

Factors other than power cost are involved in operating a compressor. Chief among these are labor, lubricating oil, repairs, depreciation, and interest on the investment. However, power costs are such a large proportion of the total (75 to 85 per cent) that comparisons are sometimes made on this basis alone.

Table 8-50 should be useful in determining approximate horsepower requirements. Table 8-49 gives theoretical horsepower for use in the formulas in Table 8-52.

Water in Compressed Air

All atmospheric air contains moisture. The ability of air to hold this moisture increases with temperature and decreases with compression. Air temperature rises during compression, but in most cases it has dropped to room temperature by the time it reaches the point of use. The net result is that the air has lost its ability to hold the moisture that it originally contained, and the excess condenses in various parts of the distribution system. Since this moisture is detrimental to the operation of air tools and other air-operated equipment, it is essential to remove it.

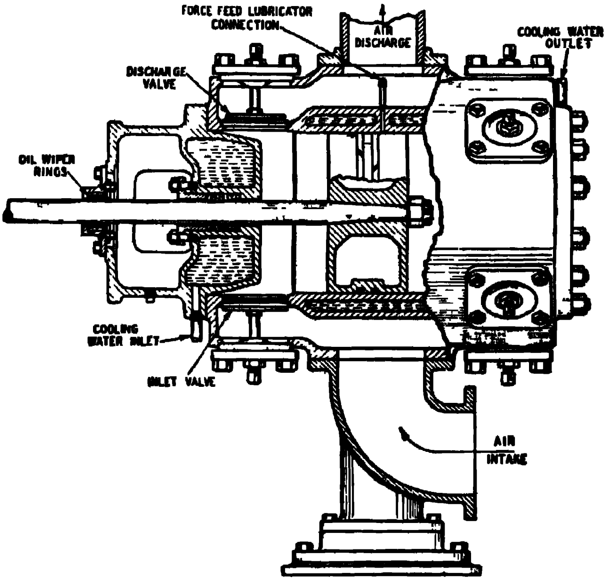


Fig. 8-150. A typical cylinder for a water-cooled compressor.

Intercoolers on multistage machines remove some of the moisture if they are equipped with automatic drains or manual drains which are regularly attended.

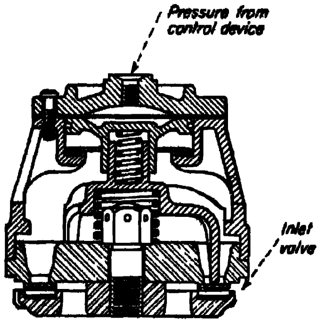
Aftercoolers are a solution to the problem in many instances. These are simple heat exchangers through which water is circulated to cool the air and thus force it to condense excess moisture. In most instances a single aftercooler of suitable size is installed between the air compressor and the receiver.

A number of different types of moisture separators are available These are usually installed at strategic places in the distribution lines near the point of use. Unless regular attendance can be assured, such separators should be equipped with automatic valves for ejecting the moisture they collect.

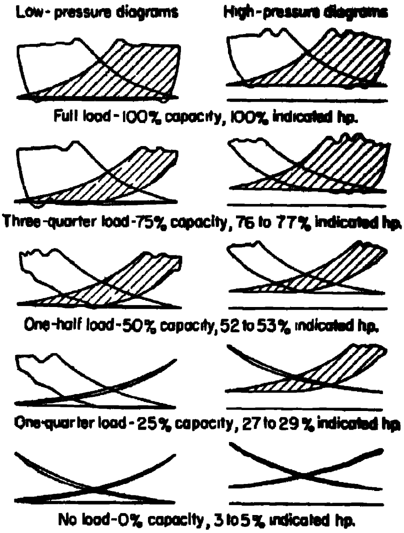
It is good practice to have branch distribution lines going directly to tools come from the top of the main air-supply line. This prevents moisture which may collect in the main from running into the supply lines.

Air Receiver

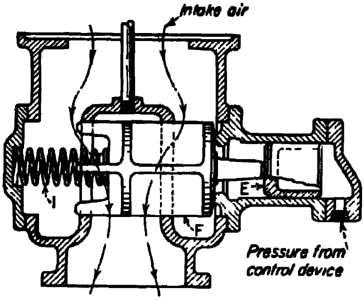
An air receiver is a necessity with reciprocating compressors and desirable in many instances with other types. It smooths out the pulsations in flow from a reciprocating machine, provides reserve storage capacity, and helps to cool the air and



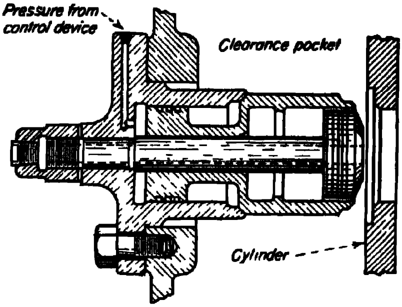
Open-inlet-valve unloader



Actual indicator diagram showing operation of clearance control at five load points of a two-stage compressor



Closed-suction unloader



Clearance valve which opens cylinder into clearance pocket

FIG. 8-151. Compressor unloading devices: (*Upper left*) Open-inlet-valve unloader. (*Lower left*) Closed-suction unloader. (*Upper right*) Actual indicator diagram showing operation of clearance control at five load points of a two-stage compressor. (*Lower right*) Clearance valve which opens cylinder into clearance pocket.

thus condense part of its moisture. The air receiver should be at least large enough to hold all the air delivered by the compressor in 1 min. This may be calculated as follows:

Receiver size, cu ft

$$= \frac{\text{piston displacement or capacity, per min} \times \text{atmospheric pressure, psi}}{\text{discharge pressure, psi,} + \text{atmospheric pressure}}$$

It does no harm to have a receiver larger than this, and in some cases a large one is necessary. This is true where occasional momentary air requirements are greater than the capacity of the compressor. In such instances, the air receiver should have enough capacity to provide all the air required for the momentary operation using air above compressor capacity (also see comments regarding receivers under Safety in the Compressor Plant).

Safety in the Compressor Plant

Compressed air is inherently a safe medium. However, because of its apparent harmlessness, certain fundamental safety precautions are sometimes overlooked.

Air released through a nozzle travels at high velocity. Air at 100 psi expands to nearly eight times its compressed volume on release to atmosphere. It follows, therefore, that workmen should be careful with air jets and should not play pranks with compressed air. Severe cases of internal rupture and even death have resulted when air jets were applied to one of the natural body openings.

Table 8-49. Theoretical Adiabatic Horsepower to Compress 100 Cu Ft per Minute of Air

[Based on sea-level intake (14.7 psi abs) and $n = 1.3947$]

Discharge pressure, psig gauge	Theoretical adiabatic horsepower		Discharge pressure, psig gauge	Theoretical adiabatic horsepower	
	One-stage	Two-stage		One-stage	Two-stage
55	12.6		110	18.9	16.1
60	13.3		120	19.8	16.8
65	13.9		130	20.7	17.3
70	14.6	12.8	140	21.5	17.9
75	15.2	13.3	150	22.3	18.5
80	15.7	13.7	160		19.0
85	16.3	14.1	170		19.5
90	16.9	14.5	180		20.0
95	17.4	14.9	190		20.4
100	17.9	15.3	200		20.9

Theoretical adiabatic horsepower (based on $n = 1.3947$)

Single-stage

$$\text{Hp} = 0.01512VP_1[(P_2/P_1)^{0.283} - 1]$$

Two-stage:

$$\text{Hp} = 0.03084VP_1[(P_2/P_1)^{0.1415} - 1]$$

where V = actual air delivery, cfm, in terms of free air (air at pressure and temperature conditions existing at compressor intake, *i.e.*, at P_1)

P_1 = intake pressure, psia

P_2 = discharge pressure, psia

NOTE 1 Absolute pressure equals gauge pressure plus atmospheric pressure.

NOTE 2 n is ratio of specific heats at constant pressure and at constant volume. It enters into all thermodynamic computations. For air containing 0.57 per cent moisture by weight, the value of $n = 1.3947$, an average for a temperate climate.

Theoretical isothermal horsepower

Single- or multistage

$$\text{Hp} = 0.00436VP_1 \log_e \frac{P_2}{P_1}$$

Air receivers and piping should be heavy enough to withstand the pressure they are to handle. Air receivers should meet code requirements of the state or locality where they will be used (in most cases the ASME code for unfired pressure vessels).

All reciprocating compressors and other positive-displacement compressors will build up higher and higher pressures if air is not used and the compressor is allowed to continue to operate fully loaded. Obviously, under such conditions, if no relief is provided, something must sooner or later give way, and air, because it expands on release, acts with almost explosive force. All compressor auxiliaries such as intercoolers, aftercoolers, and air receivers are equipped with safety valves. These must be kept in service and should be tested at regular intervals. Also, and this is important, a safety valve *must* be placed in the compressor discharge line between the compressor and any stop valve that is placed in this line ahead of the air receiver.

Oil is required to lubricate compressor cylinders. A certain amount of this oil is carried along in the air stream and is condensed out along with moisture in the inter-

cooler, compressor clearance pockets, aftercooler, air receiver, etc. Since oil is inflammable, it is highly important that it be removed. Furthermore, the air receiver, aftercooler, etc., cease to function in the manner for which they were designed if they become filled with water and oil. Drains are provided on all such units. They should be opened at regular intervals or, in the case of automatic drains, should be regularly inspected.

Table 8-50. Approximate Brake Horsepower to Compress 100 Cfm of Air

Altitude, ft	One-stage			Two-stage			
	Discharge pressure, psig			Discharge pressure, psig			
	60	80	100	60	80	100	125
Sea level	16.3	19.5	22.1	14.7	17.1	19.1	21.3
1,000	16.1	19.2	21.7	14.5	16.8	18.7	20.9
2,000	15.9	18.9	21.3	14.3	16.5	18.4	20.5
3,000	15.7	18.6	20.9	14.0	16.1	18.0	20.0
5,000	15.2	17.9	20.3	13.5	15.5	17.3	19.2
10,000	14.1	16.5	18.6	12.3	14.1	15.6	17.2
14,000	13.1	15.2	17.2	11.3	12.9	14.3	15.7

Note: Actual brake horsepower will vary considerably with size and type of compressor.

Air compressors normally operate at moderate temperatures (Table 8-51), but occasionally failure of some part or of cooling-water supplies will allow temperature to rise high enough to ignite oil or carbon deposits.

Excess oil should be eliminated by draining as mentioned in the preceding paragraph. Carbon deposits on valves should not be allowed to accumulate. Excessive lubrication of the air cylinder should be avoided. Valves or other parts should never be cleaned with gasoline or other highly volatile and inflammable liquids.

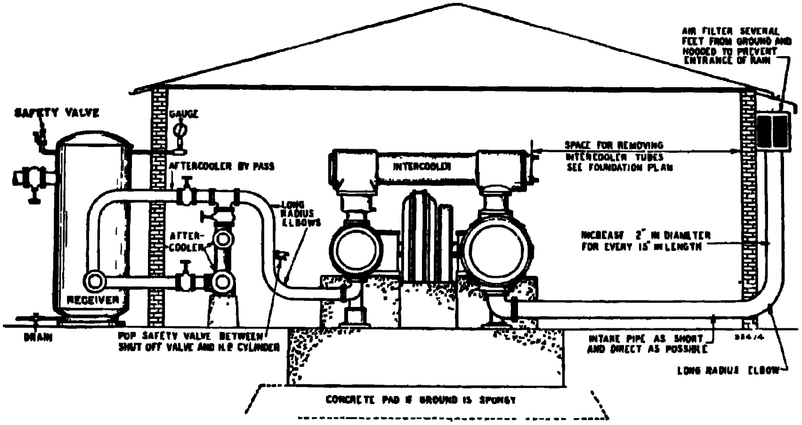

AIR FILTER SEVERAL FEET FROM GROUND AND HOODED TO PREVENT ENTRANCE OF RAIN
SAFETY VALVE
GAUGE
AFTERCOOLER BY PASS
INTERCOOLER
SPACE FOR REMOVING INTERCOOLER TUBES SEE FOUNDATION PLAN
LONG RADIUS ELBOWS
AFTER-COOLER
INCREASE 2" IN DIAMETER FOR EVERY 15' IN LENGTH
RECEIVER
DRAIN
POP SAFETY VALVE BETWEEN SHUT OFF VALVE AND H.P CYLINDER
INTAKE PIPE AS SHORT AND DIRECT AS POSSIBLE
LONG RADIUS ELBOW
CONCRETE PAD IF GROUND IS SPONGY

Fig. 8-152. Typical arrangement of equipment in a reciprocating compressor installation.

Indications of high temperature should be investigated immediately. Unusually hot valve covers are one indication. These point to broken or leaky valves. An abnormal intercooler pressure is another indication. A high intercooler pressure shows trouble in the high-pressure cylinder, and a low intercooler pressure indicates trouble in the low-pressure cylinder. On 100-lb two-stage sea-level compressors, the correct intercooler pressure is usually 27 to 29 lb gauge

On water-cooled compressors an ample supply of cooling water should be circulated at all times. Open funnels are usually provided in cooling-water discharge lines both to prevent siphoning and to permit visual inspection of the water flow.

If conditions warrant, automatic alarm systems or automatic shutdown devices can be installed to operate when the cooling-water supply fails and when temperature in the air discharge line becomes excessive.

A typical arrangement of equipment in a reciprocating-compressor installation to meet safety and convenience requirements is shown by Fig. 8-152.

Table 8-51. Theoretical Adiabatic Discharge Temperature

[Based on 60°F intake temperature, sea-level intake (14.7 psia), $n = 1.3947$, and intercooling to intake temperature on two-stage machines]

Discharge pressure, psig	Final temperature, deg F	
	One-stage	Two-stage
50	331	
60	364	
70	394	206
80	421	217
90	446	227
100	470	235
110	492	244
120	513	251
130	531	259
140	552	266
150	570	272
160		278
170		284
180		290
190		295
200		300

Single-stage

$$T = T_1(P/P_1)^{0.283}$$

Two-stage

$$T = T_1\sqrt{(P/P_1)^{0.283}}$$

where P and P_1 = same as in Table 8-49

T = absolute discharge temperature

T_1 = absolute intake temperature

NOTE. Absolute temperature = deg F plus 460°

Table 8-52. Compressor Efficiencies

$$\text{Volumetric efficiency} = \frac{\text{actual air delivery}}{\text{theoretical air delivery or piston displacement}}$$

$$\text{Compression efficiency, isothermal*} = \frac{\text{theoretical isothermal horsepower}}{\text{indicated horsepower}}$$

$$\text{Compression efficiency, adiabatic*} = \frac{\text{theoretical adiabatic horsepower}}{\text{indicated horsepower}}$$

$$\text{Mechanical efficiency} = \frac{\text{indicated horsepower}}{\text{actual horsepower input (brake horsepower)}}$$

$$\text{Over-all efficiency*} \begin{cases} = \text{compression efficiency} \times \text{mechanical efficiency} \\ = \dfrac{\text{theoretical horsepower}}{\text{actual horsepower input (brake horsepower)}} \end{cases}$$

* When using the terms "compression efficiency" and "over-all efficiency," it is necessary to specify whether the calculation is based on the adiabatic or the isothermal horsepower.

Turboblowers and Compressors

Turboblowers or centrifugal compressors are widely used to handle air or gas at pressures from $\frac{1}{2}$ up to 40 psi. Machines are available for higher pressures up to about 100 psi, but these find their main application in sizes above 15,000 cfm. For pressures below $\frac{1}{2}$ psi, one of the several types of fans is ordinarily selected.

Turboblowers are used for a wide variety of services including cooling; drying; supplying combustion air to furnaces and ovens; for blowing blast furnaces, cupolas, and converters; for transporting solid materials; for flotation processes, for agitation and aeration, for ventilation; as exhausters; etc.

The principle of a turboblower is the same as that of a centrifugal pump, the main difference being that the air or gas handled in a blower is highly compressible, while the liquids handled in a pump are practically incompressible.

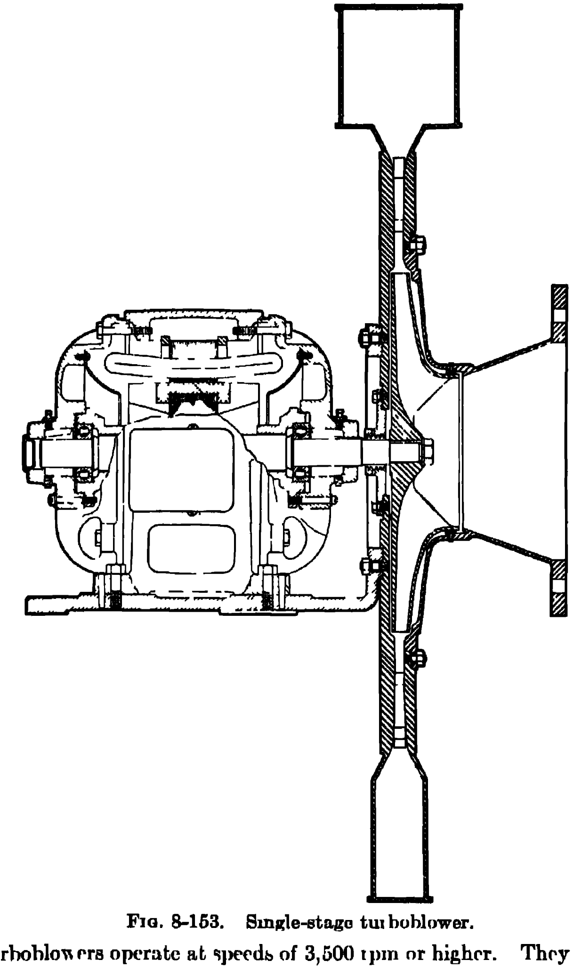


FIG. 8-153. Single-stage turboblower.

Most turboblowers operate at speeds of 3,500 rpm or higher. They are, therefore, almost exclusively driven by electric motors or steam turbines. Single-stage blowers are available to produce a pressure rise of up to 5 or 6 psi (see Fig. 8-153). From two- to seven-stage units produce pressures from 3 to 40 psi (see Fig. 8-154). For higher pressures, even more stages are required.

A turboblower is essentially a constant-pressure machine, and power consumption is almost directly proportional to the volume delivered. Blowers can also be built with impellers designed to produce decreased pressure as the volume increases.

Turboblowers can be equipped with controls to deliver gas at constant-discharge pressure, constant-suction pressure, or constant volume or constant weight

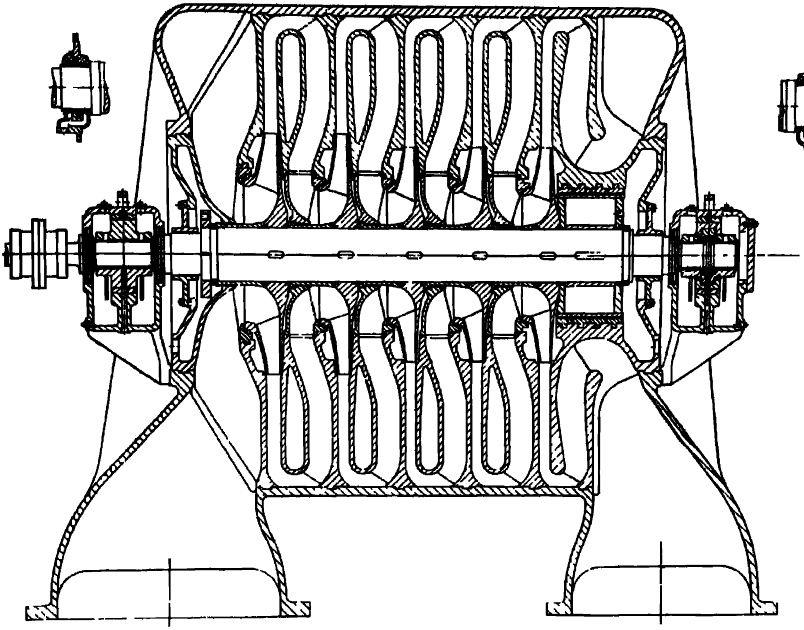


Fig 8-154. A five-stage turboblower.

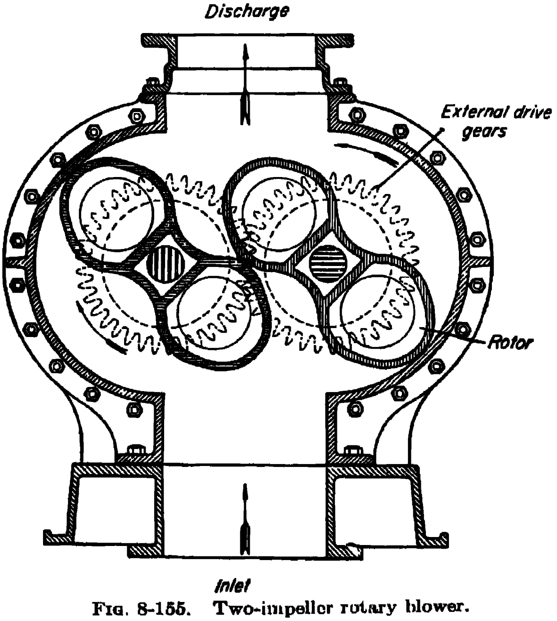


Fig. 8-155. Two-impeller rotary blower.

The usual practice on small- and moderate-capacity units is to insert a blast gate in the inlet or discharge line. This can be set manually to control the volume or pressure.

Where required, the blast gates can be automatically controlled through suitable pressure- or volume-measuring devices. The constant-weight control is an automatic control for the blast gate which controls the volume by correcting for both temperature and pressure changes in the gas handled.

On large turbine-driven units, control is usually effected by varying the speed. On large constant-speed units, control is often effected by the use of inlet guide vanes which can be set at various angles. These vanes on single-stage units show a considerable power saving over the use of blast gate to effect control. On multistage units, additional power savings can be effected by the use of a "power wheel" with the adjustable guide vanes. The power wheel is a reaction-type radial-turbine impeller which converts some of the velocity energy of gas passing at high velocity through the guide vanes into power that assists in driving the blower.

Turboblowers at about one-third to one-half of their rated capacity become unstable in their operation. This point, known as the "pumping or surge point," is accompanied by fluctuations of the inlet and discharge pressures and a whining noise. No damage is done when this point is reached. However, blowers should be selected to operate above the pumping point. Blowers are seldom required to operate at volumes below the pumping point. Where this is necessary, measures should be taken to prevent pumping. The inlet guide vanes previously described make the pumping point come at a lower volume. On small units or on machines where guide vanes will not lower the pumping point sufficiently, it is necessary to have a waste valve in the discharge line.

Rotary Blowers and Compressors

Rotary compressors, blowers, and vacuum pumps are machines of the positive-displacement type Such units are essentially constant-volume machines with variable discharge pressure. Volume can be varied only by changing the speed or by by-passing or wasting some of the capacity of the machine. The discharge pressure will vary with the resistance on the discharge side of the system.

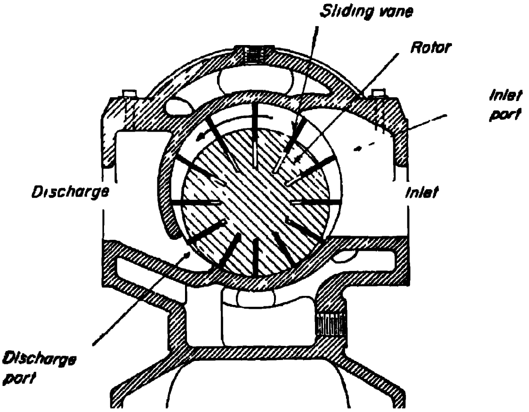


FIG. 8-156. Sliding-vane-type rotary blower.

Rotary compressors are generally classified as two-impeller, sliding-vane, and rotary-piston type

The two-impeller type is illustrated in Fig. 8-155. Such units are available for pressures up to about 15 psi and capacities up to 50,000 cfm. Sometimes multiple units are operated in series to produce higher pressures.

The sliding-vane type is illustrated in Fig. 8-156. These units are offered for pressures up to 125 psi and capacities up to 2,000 cfm. Generally, machines for pressures above 50 psi are built as two-stage units.

The liquid-piston type is illustrated in Fig. 8-157. They are offered as single-stage units for pressures up to about 75 psi and capacities up to about 4,000 cfm. Two-stage units are available for higher pressures. These units have found wide application as vacuum pumps on wet-vacuum service.

The two-impeller and sliding-vane compressors are not satisfactory for handling gas containing dust or other abrasive material. Many such materials also tend to be retained by the liquid in the liquid-piston type. The liquid in the latter type of pump can be selected to prevent contamination of the gas being handled.

Installation and Operation of Compressors

The construction details of various types and makes of compressors differ so widely that only general suggestions can be given here.

The instructions furnished by the compressor manufacturer should be followed. If you do not have these, the manufacturer will furnish them. When requesting such information, the type, size, rating, and serial number of the machine should be obtained from the name plate affixed to it.

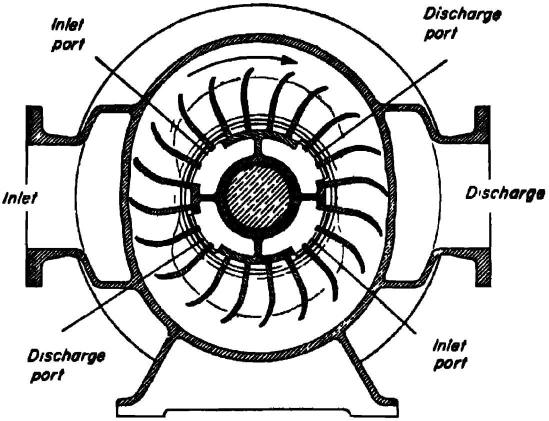


Fig. 8-157 Liquid-piston-type rotary blower.

Machines that are shipped completely assembled can usually be installed by competent mechanics or erectors from your own plant or by experienced local engineers or machinery distributors. It is usually wise on large machines to employ the services of the manufacturer's erector.

The compressor should be located in a clean, light room with ample space around it for cleaning, inspection, etc. On large units, room must be allowed for removing pistons, intercooler tube nests, piston rods, etc.

Some small machines do not require a foundation—just a good solid floor on which they can be set or to which they can be bolted. Most machines, however, will require a concrete foundation.

Compressor intakes should be equipped with intake filters. The compressor-intake piping should be as short and direct as possible. The intake itself should be located away from steam and other exhaust pipes well above the ground and in as cool a place as is available. The intake pipe should be at least as large as the intake opening on the compressor and larger if long. A good rule of thumb is to increase its diameter 2 in. for each 15 ft of length.

Safety valves should be installed on the intercooler, aftercooler, and receiver, and definitely on the discharge line between the compressor and any shutoff valve in the line.

The compressor should be lubricated in accordance with the manufacturer's directions. Small, single-acting machines may use only one type of oil—that placed in the crankcase. Larger machines usually have separate lubrication systems for the cylinders and for the bearings, etc. Air cylinders should not be overlubricated. A good check on oil quantity is to remove a valve and examine it and the cylinder. Parts should have a slight film of oil, but no excess oil should collect in the cylinder or on valve parts.

Valves should be kept clean and free from carbon. Gasoline should never be used in cleaning.

Bearings, crossheads, valves, and other working parts should be regularly checked for wear. When wear becomes excessive, parts should be adjusted or renewed.

The manufacturer's instructions should be carefully studied. His recommendations are based on experience with hundreds or thousands of machines.

PUMPS

Necessity for the movement of liquids by pumping occurs often throughout a manufacturing plant. Raw materials may be delivered to the plant as liquids. Many materials are liquids during all or part of the manufacturing processes. Water and sanitary services may require pumps as does steam-generating equipment, cooling and washing equipment, etc.

Centrifugal Pumps

The centrifugal pump is the type most widely used for transferring liquids of all types—raw materials, materials in manufacture, and finished products; as well as for general services of water supply, boiler feed, condenser circulation, condensate return, etc. They are available through a vast range of sizes; in capacities from 2 or 3 gpm up to 100,000 gpm; and for discharge heads (pressures) from a few feet up to 3,000 psi. The size and type best suited to a particular application can be determined only by an engineering study of the problem.

The principal advantages of a centrifugal pump are simplicity, low first cost, uniform (nonpulsating) flow, small floor space, low maintenance expense, quiet operation, and adaptability to motor or turbine drive.

A centrifugal pump, in its simplest form, consists of an impeller rotating within a casing. The impeller consists of a number of blades, either open or shrouded, mounted on a shaft which projects outside the casing. Impellers may have their axis of rotation either horizontal or vertical to suit the work to be done. Closed-type or shrouded impellers are used for most types of work as they are generally most efficient. Open or semiopen impellers are used for viscous liquids or liquids containing solid materials and on many small pumps for general service. Impellers may be of the single-suction type or double-suction type: single, if the liquid enters from one side; double, if it enters from both sides.

Casings are of three general types but in any case consist of a chamber in which the impeller rotates, provided with inlet and exit for the liquid being pumped. The simplest form of casing is the circular casing, consisting of an annular chamber around the impeller, no attempt being made to overcome the losses that will arise from eddies and shock when the liquid leaving the impeller at relatively high velocities enters this chamber. Such casings are seldom used. Volute casings take the form of a volute, increasing in cross-sectional area as the outlet is approached. The volute converts the velocity energy imparted to the liquid by the impeller into pressure energy, with comparatively low losses.

A third type of casing is used in diffusor-type or turbine pumps. In this type guide vanes or diffusors are interposed between the impeller and the casing chamber. Losses are kept to a minimum in a well-designed pump of this type. This construction is often used in multistage high-head pumps.

Centrifugal-pump Characteristics. Figure 8-108 shows a typical characteristic curve of a centrifugal pump. It is important to note that pump will operate along this curve and at no other points. For instance, on the curve shown, at 200 gpm the pump will generate 87-ft head. If the head is increased to 100 ft, 120 gpm will

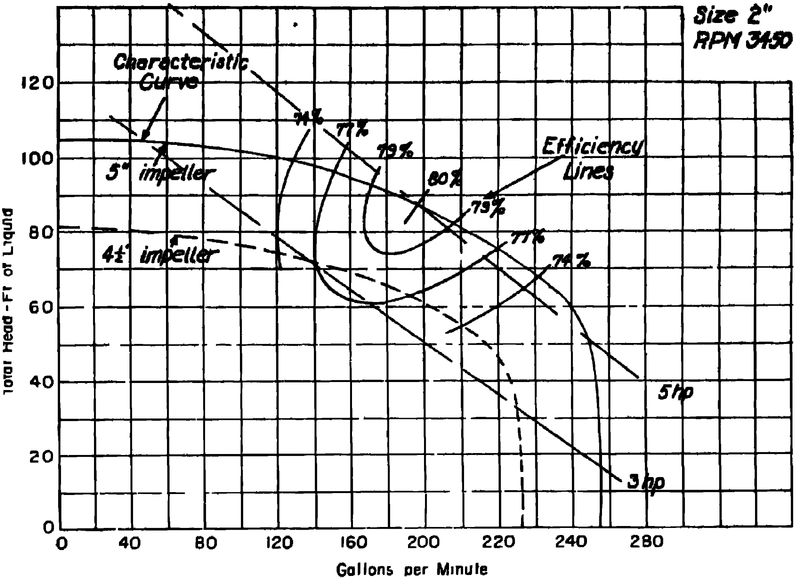


Fig. 8-158 Centrifugal-pump characteristic curve for different impeller sizes

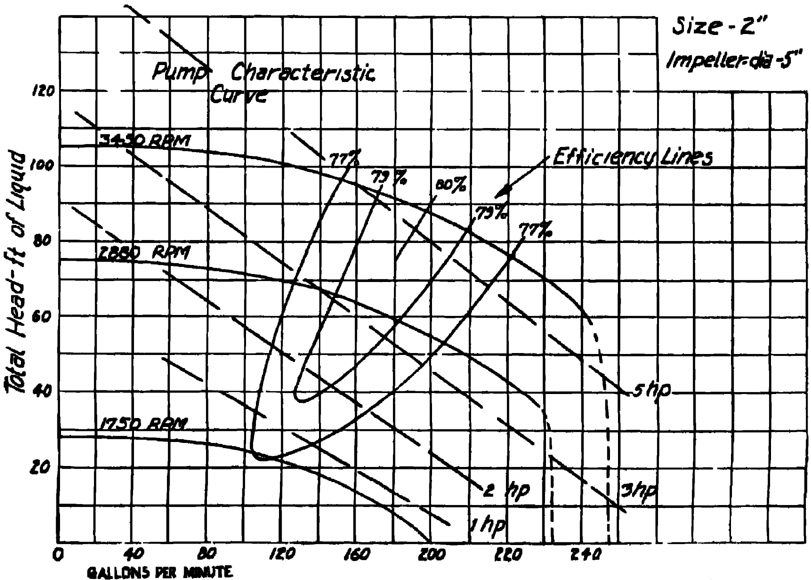


Fig. 8-159. Centrifugal-pump characteristic curve for different speeds.

be delivered. It is not possible to reduce the capacity to 120 gpm at 83-ft head unless the discharge is throttled so that 100-ft head is actually generated within the pump. On pumps with variable-speed drivers such as steam turbines, it is possible to change the characteristic curve as shown by Fig. 8-159.

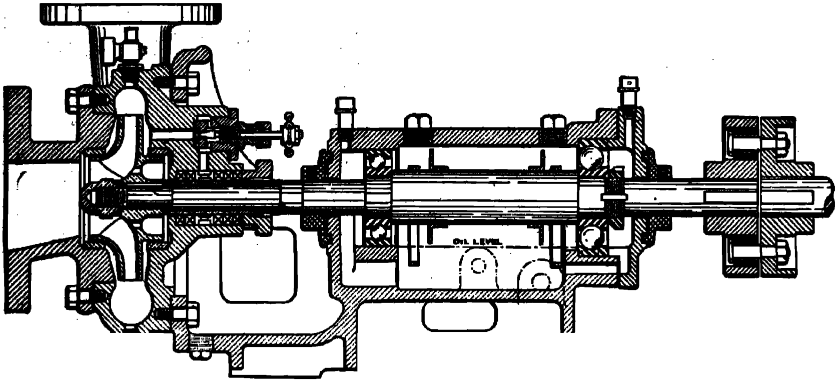


Fig. 8-160. Typical cradle-mounted pump.

Single-stage Centrifugal Pumps. Single-stage centrifugal pumps are available in capacities up to and over 50,000 gpm for heads (pressures) up to 200 ft or sometimes as high as 350 ft. They are available in a variety of designs for particular services.

Cradle-mounted Pumps. Cradle-mounted pump is a term usually applied to single-stage units of simple design with impeller mounted on an overhung shaft for capacities up to about 3,000 gpm and heads up to about 200 ft (see Fig. 8-160). Such units are designed for ease in dismantling and accessibility. They practically always have single-suction impellers.

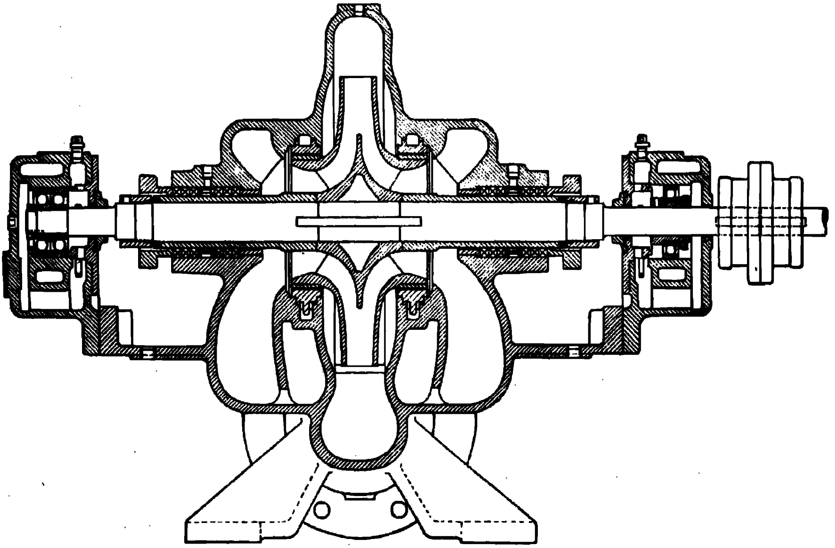


Fig. 8-161. Single-stage double-suction pump.

Double-suction Single-stage Pumps. Double-suction single-stage pumps (see Fig. 8-161) are used for general water-supply and circulating service and for many transfer services. They are available for capacities from about 25 gpm up to and over 50,000 gpm and heads up to 200 ft or so. Such units are available in iron or bronze or iron with bronze fittings. Other materials increase the cost, and where such materials are required, a standard-type chemical pump or other special service pump is usually more economical.

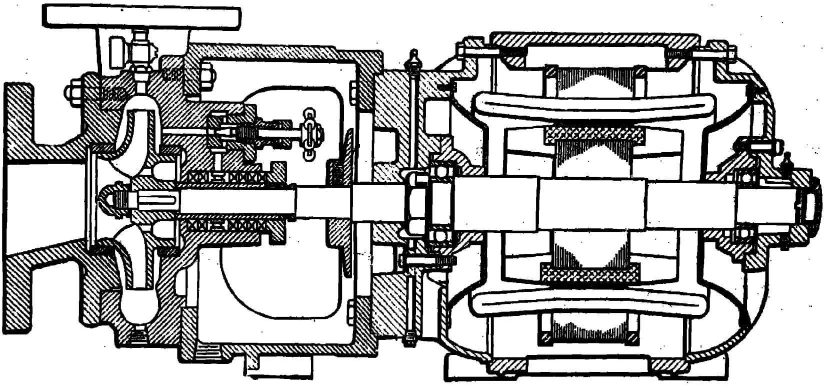


FIG. 8-162. A typical close-coupled pump with built-in motor driver.

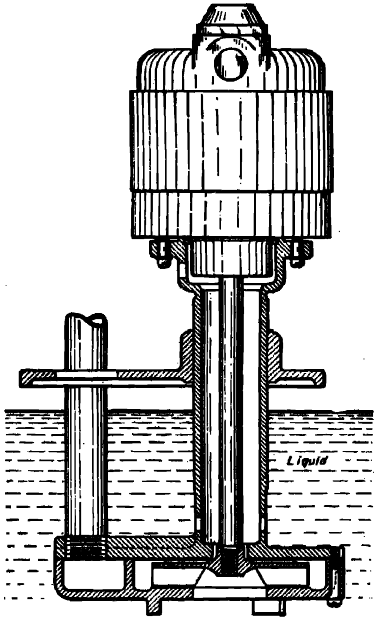


FIG. 8-163. Immersion-type coolant pump.

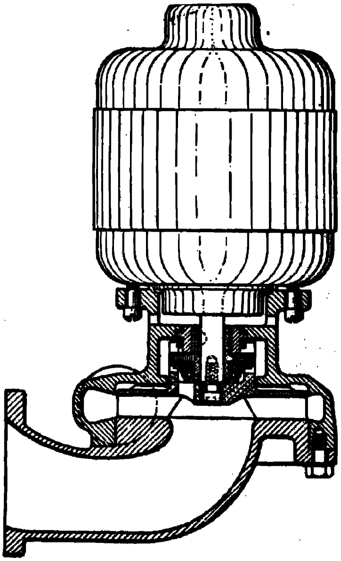


FIG. 8-164. Side-wall-type coolant and circulating pump.

Close-coupled Pumps (see Fig. 8-162). Pumps with built-in electric motor or sometimes steam-turbine driver (*i.e.*, with pump impeller and driver on the same shaft) are known as close-coupled pumps. Such units are extremely compact and are suitable for a variety of services where standard iron and bronze materials are satisfactory. They are available in capacities up to about 2,000 gpm for heads up to about 240 ft. Two-stage units in the smaller sizes are available for heads to around 500 ft, and some manufacturers furnish such pumps with a considerable number of stages.

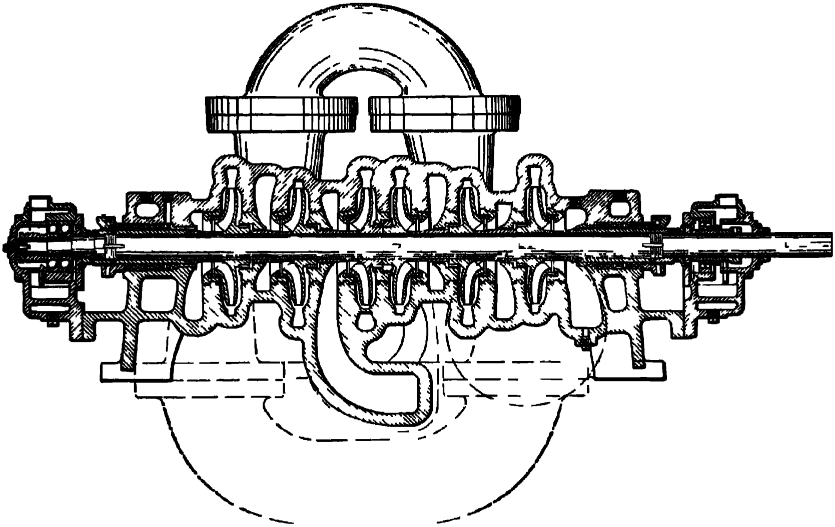


FIG 8-165. Six-stage volute-type pump.

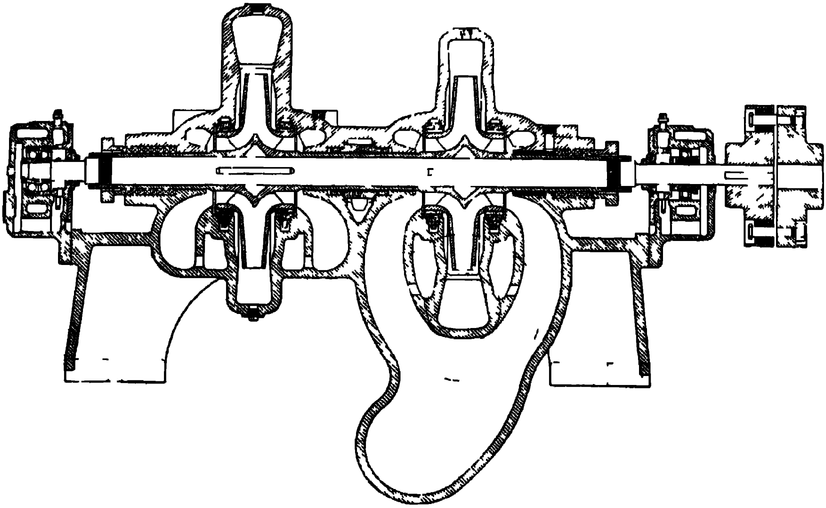


FIG 8-166. Two-stage pump with double-suction impellers.

Special-application Pumps A number of pump designs are available for special applications such as handling chemicals, paper stock, hot liquids, volatile hydrocarbons, etc. Such pumps are basically of the types described but with features designed to meet special needs of the service for which they are to be used. Pump manufac-

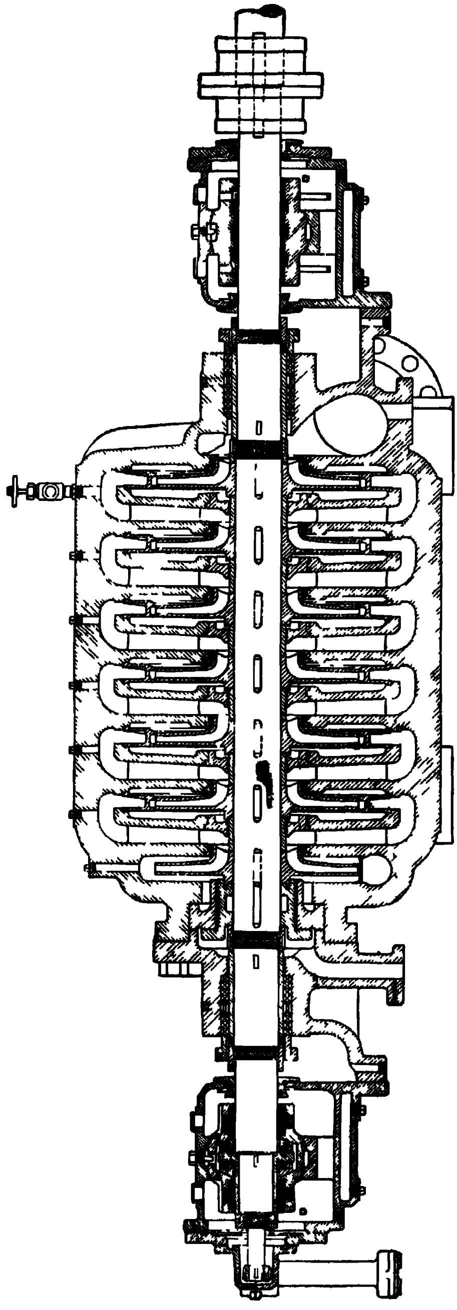


FIG 8-167 Seven-stage diffuser-type pump

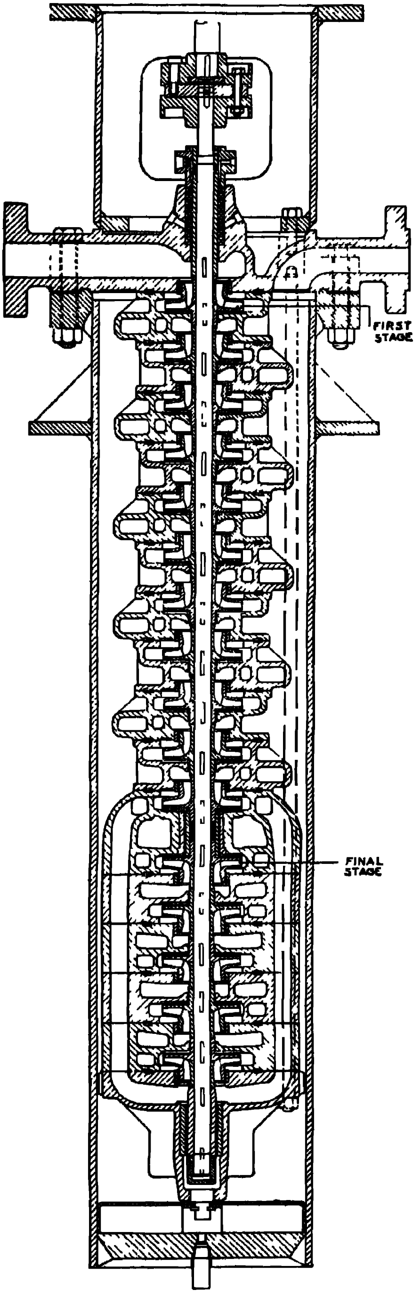


FIG. 8-168 A 16-stage vertical pump

turers can furnish complete information on units for such applications.

Sump Pumps. Sump pumps are small single-stage pumps used to drain shallow pits or sumps. They usually have single-suction open-type impellers. The pump itself is immersed in the liquid and driven by a shaft from a motor or belt pulley above the pit. A tube or column encases the shaft and forms a connecting support between the pump and the driver and for whatever shaft bearings may be required

Coolant Pumps. Coolant pumps are used for circulating coolant on machine tools. These are often of the immersion type in which the pump is immersed in the liquid (see Fig. 8-163) Others of the side-wall type (see Fig 8-164) are mounted on the side of a tank

Multistage Centrifugal Pumps. Multistage pumps are, in general, used for services requiring higher heads (pressures) than can be generated by single-stage pumps. Such services include h-p water-supply pumps, fire pumps, boiler-feed pumps, charge pumps for refinery processes, etc. Such pumps are available for pressures as high as 3,000 psi at capacities up to 3,000 gpm and above

Multistage pumps may be of the volute type or of the diffusor type Volute-type pumps (see Fig 8-165 usually have single-suction impellers arranged with half the impeller inlets facing one direction and half in the opposite direction to balance thrust Some two-stage (see Fig. 8-166) units have double-suction impellers, but this construction, except for special-purpose units, makes the casings of impractical size for more stages.

Diffusor-type pumps (see Fig 8-167) usually have single-suction impellers arranged with all impeller inlets facing in the same direction and impeller thrust neutralized by a differential pressure device known as a "balancing drum"

Pumps for very high pressures often have an inner volute-type casing or assembly of diffusor units placed within a forged-steel shell or barrel.

Multistage pumps for small or moderate capacity are available in a vertical design which for many services simplifies the piping and installation (see Fig. 8-168).

Propeller Pumps

The term "propeller pump" is applied to units with axial-flow or mixed-flow (*i e*, part axial and part centrifugal) impellers Such units are available in capacities from 100 gpm upward for heads up to about 100 ft per stage. Propeller pumps are usually vertical, although some single-stage horizontal units are built.

A common form of propeller pump has the pump element mounted at the bottom of a column which serves as the discharge pipe (see Fig 8-169) Such units are immersed in the liquid to be pumped and are commonly used for large-volume drainage and large-volume circulating service such as surface condensers, etc.

Another form of the pump has a shell surrounding the pumping element which is connected to the intake pipe (see Fig 8-170) In this form the pump is used on condensate service in power plants and for process work such as encountered in oil refineries

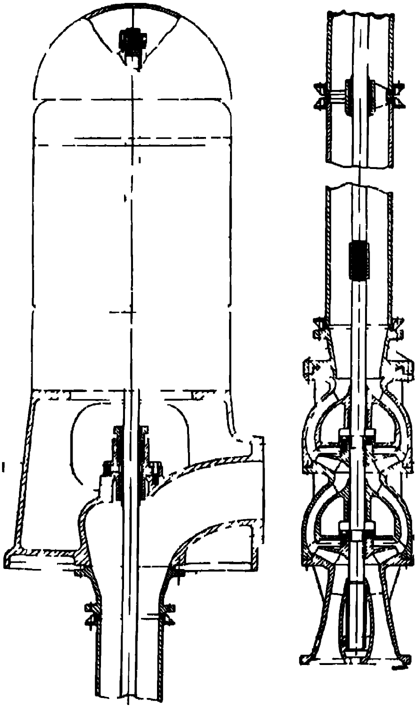


Fig 8-169 A propeller or mixed-flow pump

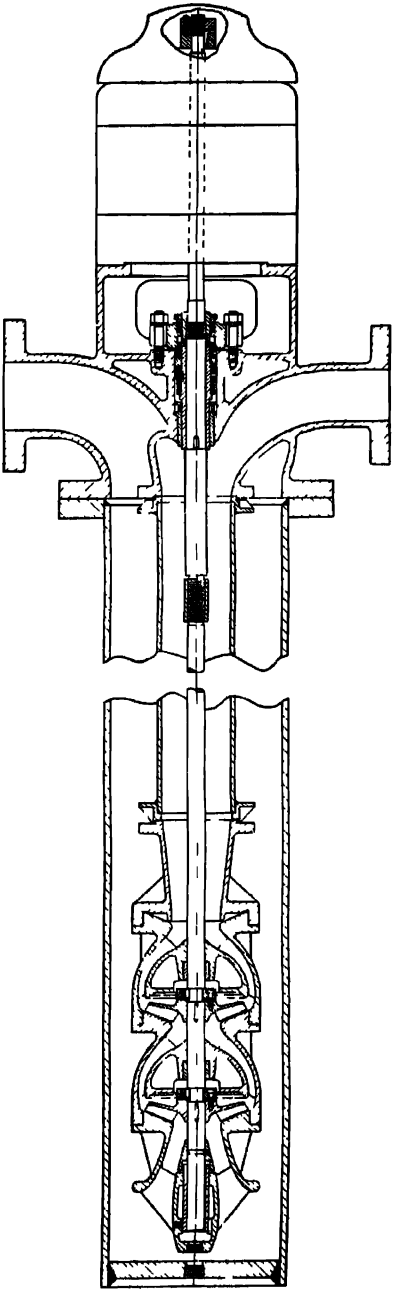


Fig 8-170 A two-stage mixed-flow pump with suction shell

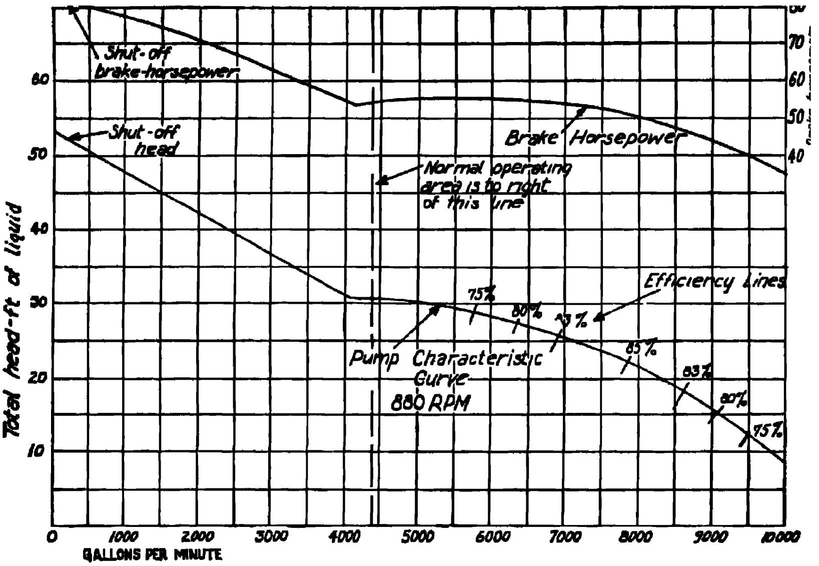


Fig. 8-171. Propeller-pump characteristic curve.

A characteristic curve of a typical axial-flow pump is given in Fig. 8-171. Pumps with mixed-flow impellers have a lower shutoff (i.e., closed discharge) head and lower horsepower at this point, but not so low as for a centrifugal pump.

Selecting a Pump

In selecting a pump, it is necessary to know the liquid to be handled, the total discharge head, the capacity, the suction lift or head, and in most cases, the temperature, viscosity, and vapor pressure of the liquid to be handled.

To illustrate these points, a typical example is given: An industrial plant wishes to install a pump to lift 200 gpm of water at 72°F from a sump to a tank on the roof. The water is to be delivered into the tank at 10 lb pressure. The tank is 58 ft above the sump, and the pump is 4 ft above the water level in the sump. The discharge pipe from the pump to the tank is 400 ft long and contains four standard elbows, one check valve, and one gate valve. A $2\frac{1}{2}$-in. discharge line is already installed which the plant superintendent would like to use if possible. The suction pipe is 4 in. in diameter, 25 ft long, and contains two elbows and a foot valve. For comparison, two solutions are given in Table 8-53: Solution A, using the available $2\frac{1}{2}$-in. discharge pipe, and Solution B, using 4-in. discharge pipe. Friction losses in this problem are based on using old, rough pipe. Solution A shows 295 ft total head which will require a pump driven by a 25-hp motor. Solution B shows 108 ft total head which will require a pump driven by a $7\frac{1}{2}$-hp motor. These two problems forcibly point out the savings that a discharge pipe of proper size makes possible.

Reciprocating Pumps

There are two general classes of piston or reciprocating pumps: steam pumps and power pumps. In general, the action of the liquid-transferring parts of these pumps is the same, a cylindrical piston, plunger, or bucket being caused to pass back and forth in a cylinder. The device is equipped with valves for inlet and discharge of the liquid being pumped, and the operation of these valves is related in a definite manner to the motions of the piston.

Reciprocating pumps are usually provided with an air chamber, as shown in Fig. 8-172. This serves to smooth out irregularities in the discharge of the pump and

Table 8-53

	Solution A 2½-in discharge pipe, 4-in suction pipe			Solution B 4-in discharge pipe, 4-in suction pipe		
Discharge head						
Length of discharge pipe		400′			400′	
4 ells—equivalent length of pipe	4 × 6.2′ =	24.8′		4 × 10.2′ =	40.8′	
1 check valve—equivalent length of pipe	1 × 22 =	22′		1 × 41 =	41′	
1 valve—equivalent length of pipe	1 × 1.32 =	1.3′		1 × 2.1 =	2.1′	
Total length for figuring friction		448.1′			483.9′	
Friction loss per 100 ft	46.3			4.3		
Total discharge friction loss	$\frac{46.3 \times 448.1}{100}$ =		207.5′	$\frac{4.3 \times 483.9}{100}$ =		20.8′
Static discharge pump to tank			58′			58′
Head, tank pressure	10 × 2.31 =		23.1′			23.1′
Total discharge head			288.6′			101.9′
Suction lift						
Length of suction pipe		25′			25′	
2 ells—equivalent length of pipe	2 × 10.2 =	20.4′		2 × 10.2 =	20.4′	
Foot valve—equivalent length ft		0			0	
Total length for figuring friction		45.4′			45.4′	
Friction loss per 100 ft	4.3			4.3		
Total suction friction loss	$\frac{4.3 \times 45.4}{100}$ =		2.0′	$\frac{4.3 \times 45.4}{100}$ =		2.0′
Static suction lift			4.0′			4.0′
Total suction lift			6.0′			6.0′
Total head						
Total discharge head		288.6′			101.9′	
Total suction lift		6.0′			6.0′	
Total head		294.6′			107.9′	

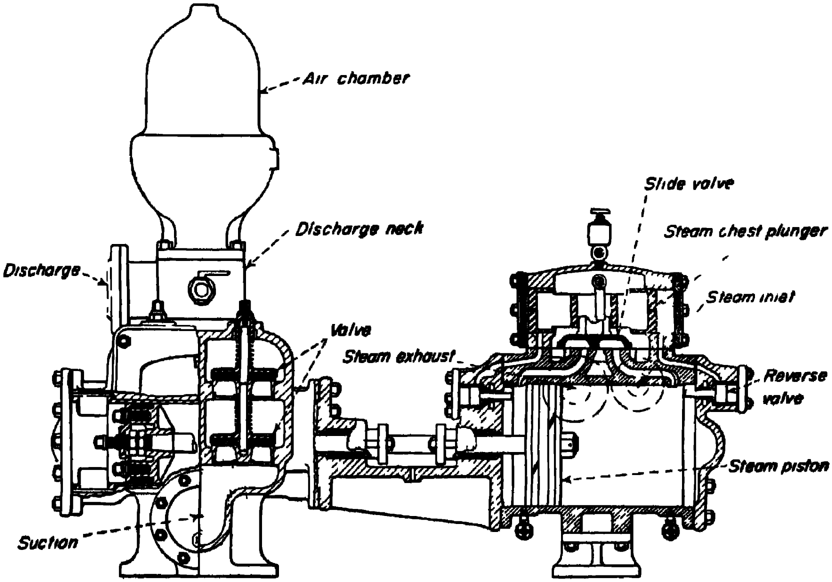


Fig. 8-172. Simplex direct-acting steam pump.

gives a uniform flow It is always used with single steam pumps and power pumps With low-pressure, direct-acting, duplex steam pumps, it is not needed With duplex pumps its volume should be four times the displacement of one piston per stroke, with all other types, eight times

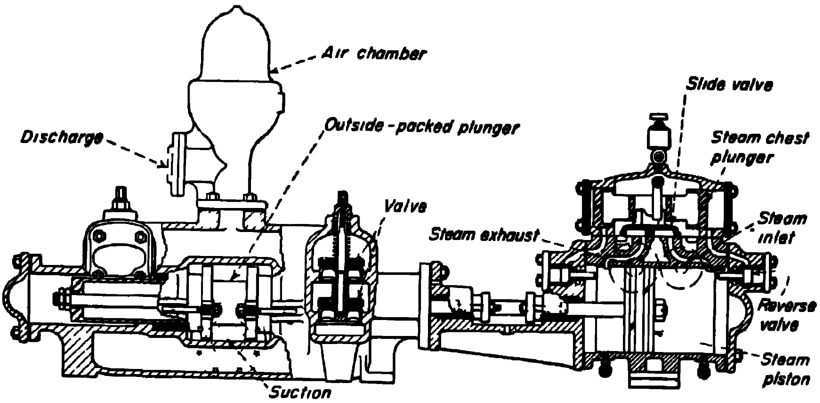


Fig 8 173 A simplex outside packed plunger pump

Types of Reciprocating Pumps The ordinary types of reciprocating pumps are the four pumps described below

1 *Simplex Double Acting* These may be direct-acting (*i e*, direct-connected to a steam cylinder) or power driven (through crank and flywheel from the crosshead of a steam engine) Figure 8-172 is a pump of this type designed for use at heads up to 200 ft

A simplex, outside-packed plunger pump of the type used with hydraulic presses and for similar service is shown in Fig 8-173

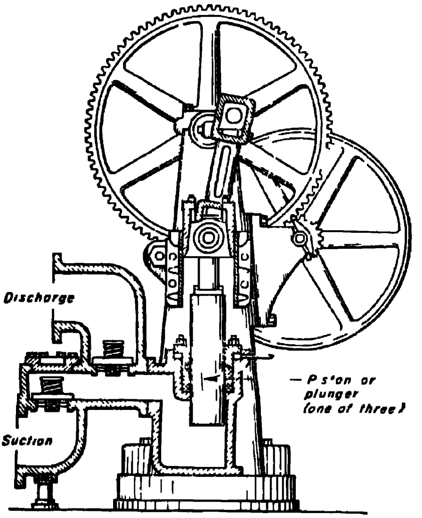


Fig 8-174 Triplex single acting reciprocating pump

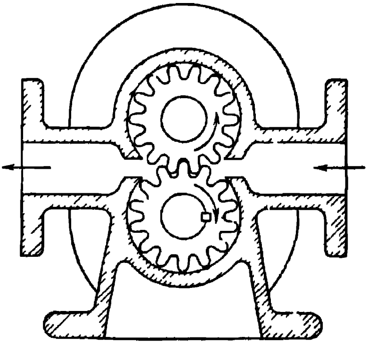


Fig 8-175 Gear-type rotary pump

2 *Duplex, Double Acting* These pumps differ primarily from those of the simplex type in having two water cylinders whose operation is coordinated These pumps may be either direct-acting, steam-driven, or power-driven with crank and flywheel

3. *Triplex, Single Acting.* These pumps have three single-acting plungers and cylinders and are used to give a uniform flow. They are usually of vertical design. The drive may be from a motor, belt, or steam cylinder and is through gears or crank and flywheel. This is the common type of power pump, an example of which, arranged for belt drive, is shown in Fig. 8-174, from which the action is readily traced.

4. *Triplex, Double Acting.* This is a double-acting arrangement of the pump discussed under 3. The design is generally used for horizontal triplex pumps. Other features are similar to 3.

Choice of a Reciprocating Pump. The advantages of a direct-driven pump lie in good efficiency over a wide range of operating conditions and flexibility of capacity, head, and speed. The disadvantages are usually high first cost, large floor space required, noisy operation, and more attention required.

The advantages given for a direct-driven steam pump do not apply to power-driven pumps. A triplex single- or double-acting pump does not have flexibility as to speed or capacity. It does have uniform delivery. Its use is indicated in pumping wells, in handling steady flow of liquids, and in serving machines operated by hydraulic pressure.

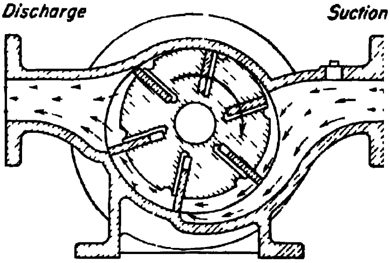


FIG. 8-176. Vane-type rotary pump.

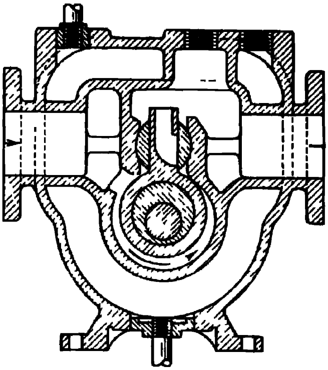


FIG. 8-177. Cam-and-piston-type rotary pump. This unit shown with jackets for heating of viscous liquids.

Simplex double-acting pumps are most suitable for water service and boiler feed.

Duplex double-acting pumps are universally adaptable. When made of the proper materials, particularly as to cylinder linings and valve material, and when of correct design, these pumps are frequently used for tars, oils, and other viscous liquids.

Triplex single-acting pumps are suitable for producing uniform flow when conditions are constant. Triplex double-acting pumps are used under the same conditions as triplex single-acting pumps but are generally of horizontal design.

Reciprocating pumps, particularly of the direct-acting, steam-driven design, are available in a great variety of designs and sizes to fill every need for which such a pump is suitable.

Another form of reciprocating pump is known as the "diaphragm pump." In these units a flexible diaphragm separates the suction chamber from the discharge chamber. The diaphragm moves up and down through the action of a yoke or rod, and a discharge valve forces the liquid from the suction chamber into the discharge pipe. Diaphragm pumps are used for emptying tanks and sumps and for dewatering excavations, etc

Rotary Pumps

Rotary pumps differ from centrifugal and reciprocating pumps in that they will deliver a positive quantity of liquid under conditions of varying head or pressure. These pumps will handle any liquid that contains no grit or abrasive material.

This type of pump consists of a stationary casing in which are located one or more rotating members. When one rotating member is used, it is mounted eccentrically on the shaft. The impeller in this type of pump is usually circular in section and is provided with a reciprocating blade or blades or a horizontal abutment. Figures 8-175, 8-176, 8-177, and 8-177*A*, show several types of rotary pumps.

Rotary pumps are available in two general classes: interior bearing and external bearing. The interior-bearing type is used for handling liquids of a lubricating nature, while the exterior-bearing type is used with nonlubricating liquids. The interior-bearing pump is lubricated by a part of the liquid being pumped, while the exterior-bearing type is oil-lubricated by an attendant.

Among the liquids handled by rotary pumps are water, mineral oils, vegetable oils, animal oils, greases, glucose, molasses, paints, varnish, shellac, lacquers, alcohols, catsup, brine, mayonnaise, sizing, soap, tanning liquors, vinegar, and ink. It should be kept in mind that most rotary pumps are not suitable for handling liquids carrying grit or abrasive material.

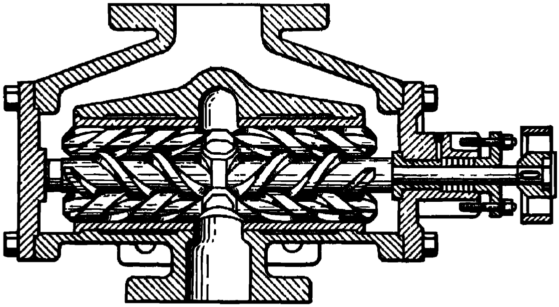


FIG. 8-177A. Screw-type rotary pump.

Air-pressure Pumps

Handling Liquids with Fluid Pressure. In addition to the liquid-handling devices, such as pumps, that depend on the mechanical action of pistons, plungers, or impellers to move the material, there is another group of devices for this purpose that employs gas or air pressure to move the liquid. This group includes air lifts, acid eggs, jet pumps, and pulsometers, among others.

Air Lift. The air lift is a device for raising liquid by means of compressed air. In the past it was widely used for pumping wells but has been less widely used since the development of efficient centrifugal pumps. It operates by introducing compressed air into the liquid near the bottom of the well. The air and liquid mixture, being lighter than liquid alone, rises in the well casing. The advantage of this system of pumping lies in the fact that there are no moving parts in the well. The pumping equipment is an air compressor which can be located on the surface.

Displacement Pumps. Displacement pumps operate by forcing the liquid that is to be moved from one point to another by means of air or gas pressure on the surface of the liquid. The acid egg is a device of this type frequently encountered by chemical engineers. It consists of an egg-shaped container which can be filled with a charge of the liquid that is to be pumped. This container is fitted with an inlet and outlet pipe for the liquid and a pipe for the admission of compressed air or gas. Pressure of air or gas on the surface of the liquid forces it out the discharge pipe. Such pumps can be manually operated or can be arranged for semiautomatic or automatic operation.

Jet Pumps. Jet pumps constitute another class of liquid-handling devices that makes use of fluid pressure as an operating medium. Ejectors and injectors are the two types of most common interest. The ejector, also called siphon, exhauster, or eductor, is designed for use in operations where the head pumped against is low and is less than the head of the fluid used for pumping. The injector is a special type of jet pump used for boiler feed and similar services.

The efficiency of an ejector or jet pump is low, around 1 or 2 per cent. The head developed by the ejector is also low, except in special types. The device has the disadvantage of diluting the fluid pumped by mixing it with the pumping fluid. In steam injectors for boiler feed and similar services, where the heat of the steam is recovered, the efficiency is close to 100 per cent.

The simple ejector or siphon is used, in spite of its low efficiency, for transferring liquids from one tank to another, for lifting chemicals, or solid-containing liquids of an abrasive nature, for priming other types of pumps, etc.

Packing and Shaft Seals

Proper packing and shaft seals are items of major importance, particularly for centrifugal pumps handling chemicals or high-temperature liquids. The success of the pumping installation often depends upon the proper selection of packing. Many types and designs are available, some of which are described below.

Stuffing-box arrangements vary with the liquid, the temperature, and the pressure. They can generally be classified as being packed solid or having a sealing cage located approximately in the middle of the box. In the latter case the sealing liquid may come directly from the pump, in which case it is said to be internally sealed; or if from an outside source, the box is said to have an external seal. The following arrangements will cover most pumping conditions:

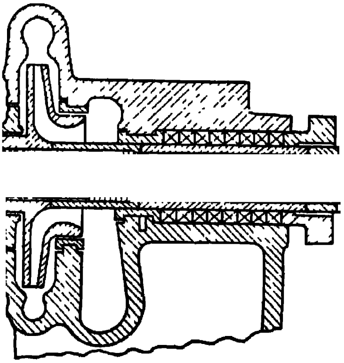


Fig. 8-178. Stuffing box packed solid without seal ring

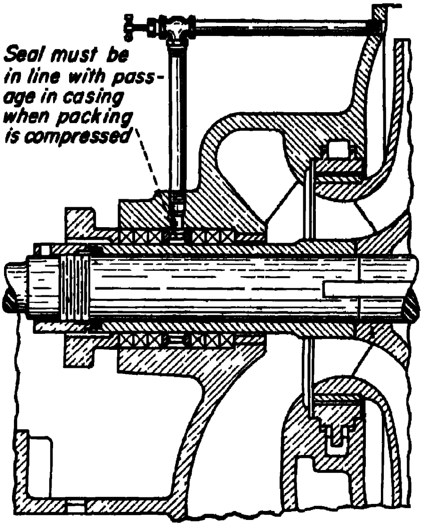

Seal must be in line with passage in casing when packing is compressed

Fig. 8-179. Stuffing box with internal seal.

1. **Positive Pressure on the Suction.** When there is a positive pressure on the suction, the seal cage may be omitted, and the stuffing boxes can be packed solid and adjusted so that a slight leakage cools and lubricates the packing (see Fig. 8-178)

2. **Suction Lift.** When a unit is working under a suction lift, it is important to prevent air from leaking into the pump through the stuffing box, for this will cause a reduction in capacity or loss of prime. To prevent air leakage and to lubricate the packing, sealing liquid is injected at a seal cage usually located at approximately the center of the box. When the sealing medium is the liquid pumped, the arrangement is known as an "internal seal" (see Fig. 8-179). When the sealing medium comes from an outside source, it is known as "external sealing" (see Fig 8-180).

3. **Flooded Suction.** An external or internal seal should be provided in order to cool and lubricate the packing (see Figs. 8-179 and 8-180).

4. **Gritty or Corrosive Liquids.** When the liquid handled is gritty or corrosive, the stuffing boxes should be externally sealed with a cool, clean liquid at approximately 10 psi above the suction pressure. The sealing liquid should be supplied at

constant pressure to prevent flexing action in the packing (see Fig 8-180) When gasoline, kerosene, gas, oil, or similar hydrocarbons are being handled with nominal suction pressure, a stick grease-type lubricator can be used to inject grease into the seal cage, instead of using globe valve shown in Fig 8-180

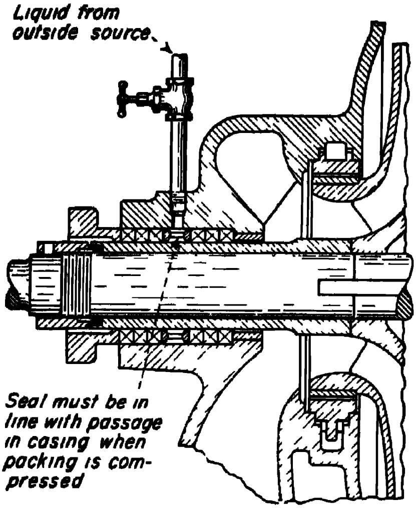

Liquid from outside source
Seal must be in line with passage in casing when packing is compressed

FIG 8-180 Stuffing box with external-seal system

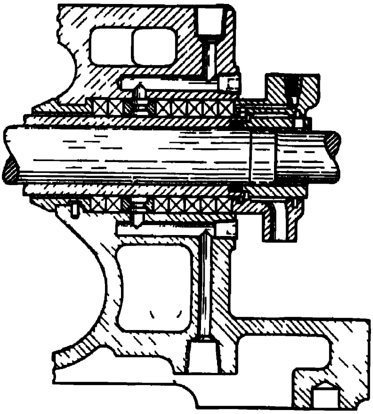


FIG 8-181 Circulating-type stuffing box

5. High Temperature and High Pressure. *(a) Circulating Box* When the temperature of the liquid to be handled is high enough to warrant internal cooling of the packing and the pressure falls within the limits for safe packing, a stuffing-box arrangement as shown in Fig 8-181 can be used in order to provide cooling of the packing Cool, clean liquid enters the sealing cage at one connection, circulates through the seal cage, and leaves at another connection

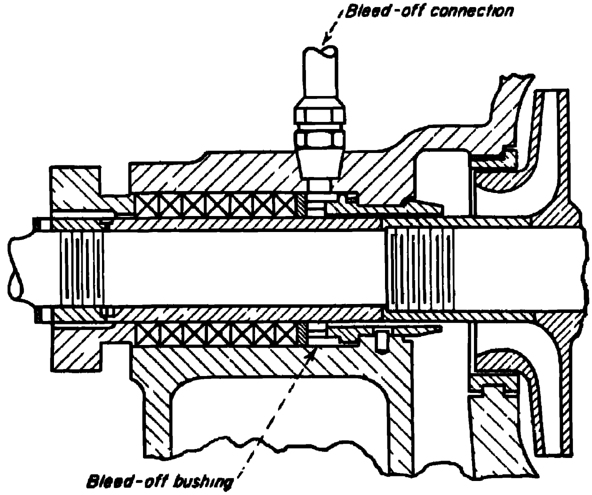


FIG 8-182 Stuffing box with bleed-off bushing

(*b*) *Bleed-off Box.* It sometimes happens that the suction pressure must be reduced to an amount that can be safely packed in order to ensure satisfactory operation of the stuffing box. This can be accomplished by locating a bleed-off bushing in the bottom of the stuffing box as shown in Fig 8-182. The bleed-off bushing can usually be used in conjunction with any of the stuffing-box arrangements previously described. The tapped bleed-off connection must be piped to a low-pressure point in the system. The pressure should never be reduced to such an extent that it will be below the vapor pressure of the liquid at the pumping temperature.

Smothering Glands. When handling liquids where the lubricating leakage through the packing might result in inflammable gas, noxious fumes, or steam, a smothering-type packing gland may be furnished (see Fig. 8-183). The water used for smothering may be taken from a cooling-water line through a flexible hose or dead soft copper tube.

Mechanical Seals. In order to overcome packing difficulties, several types of mechanical sealing devices have been designed and have proved very satisfactory under severe operating conditions (see Fig. 8-184).

Mechanical seals consist of a rotating and a stationary element having sealing faces that are brought into sliding contact under a pressure sufficient to prevent escape of the fluid being pumped.

Mechanical seals are generally used where the fluid handled has at least slight lubricating qualities and is free from abrasive and other foreign material that would tend to destroy the sealing faces.

Satisfactory operation of a mechanical seal depends on maintaining perfect contact of the sealing faces at all times.

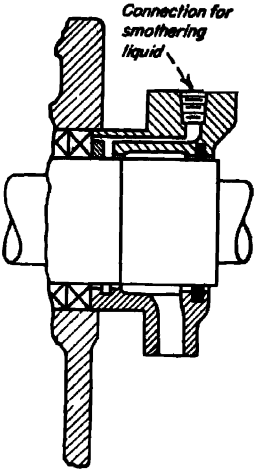


FIG. 8-183. Smothering-type stuffing-box gland

Double Mechanical Seals. There are a few conditions of service where it is desirable to provide a double seal to prevent the liquid being handled by the pump from coming in contact with the sealing parts.

Pumps handling corrosive and high-temperature liquids may require a double seal. In such cases, noncorrosive or low-temperature liquids are injected between the two seals at a pressure slightly higher than that obtained in the pump chamber next to the inner seal.

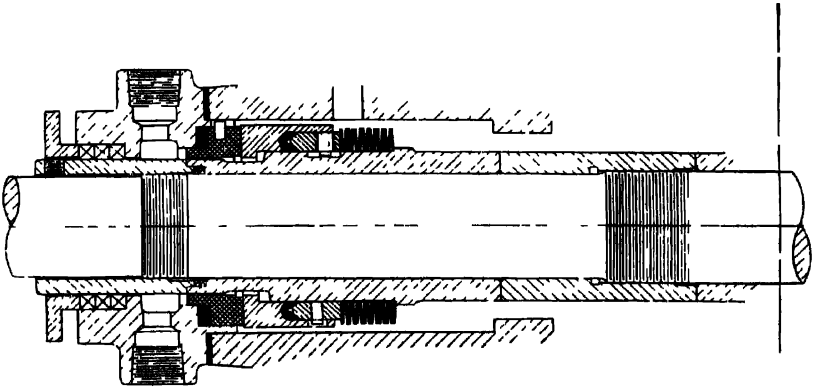


FIG. 8-184. Ingersoll-Rand shaft seal for Cameron pumps.

For double seals, it is necessary to have additional equipment such as sealing-liquid pump, tank, pressure regulator, and piping to the stuffing boxes.

Terms Used in Pumping

Liquids are said to be practically incompressible. This is near enough to the truth at low pressures, but at high pressures there is a slight change in density, which in some cases it is necessary to consider.

The pressure existing at any point in a liquid at rest is caused by the pressure exerted on its surface plus the weight of liquid above the point in question. If the liquid is in an open vessel, the pressure on the surface is atmospheric pressure. Pressure thus exerted on a liquid is equal in all directions and acts perpendicularly to any surfaces in contact with the liquid. All liquid pressures can be thought of as equivalent to a column of liquid of a height sufficient by its weight to produce the pressure in question. In pumping practice, pressures or heads are thus often expressed in feet of liquid.

Capacity. Capacity is the volume of liquid handled by a pump. It is usually expressed as volume per unit of time as gallons per minute (gpm) or barrels per hour (bph).

Velocity. Since a liquid is practically incompressible, there is a definite relation between the quantity flowing past a given point in a given time and the velocity of flow. This relation is expressed thus:

$$Q = Av$$
$$v \text{ (for circular conduits)} = 0.4085 \text{ gpm}/d^2 = 0.2859 \text{ bph}/d^2$$

Viscosity. In flowing liquids the existence of internal friction or the internal resistance to relative motion of the fluid particles must be considered. This resistance is called "viscosity." It varies greatly from one liquid to another and decreases with rising temperature. Viscous liquids tend to increase the horsepower required by a pump, to reduce the pump efficiency, head, and capacity, and to increase the friction in pipe lines.

Friction Head. Friction head (h_f) is the pressure (in terms of feet of liquid) required to overcome the resistance to flow in pipe and fittings.

Velocity Head. Velocity head (h_v) is the vertical distance a body would have to fall to acquire the velocity v. It corresponds to the static or pressure head which would cause that velocity.

$$h_v = v^2/2g = 0.0155v^2 \text{ (approx.)}$$
$$h_v \text{ (for circular conduits)} = 0.00259 \text{ gpm}^2/d^4 = 0.00127 \text{ bph}^2/d^4$$

Velocity head may be converted to equivalent pressure head by suitable means such as a Venturi tube. In any flowing liquid the sum of pressure head and velocity head remains constant.

Total Static Head. Total static head (h_{ts}) on a pump is the vertical distance (in feet) between the free level of the source of supply and the point of free discharge or the level (or equivalent level) of the free surface of the discharge liquid.

Total Suction Lift. Total suction lift (l_s) is the reading of a mercury column at the suction flange of a pump (corrected to the pump center line† and converted to feet of liquid) minus the velocity head (in feet) at the point of column attachment.*

$$l_s = l_{ss} + h_f \text{ suct}$$

Total Suction Head. Total suction head (h_s) is the reading of a gauge at the suction flange of a pump (corrected to the pump center line† and converted to feet of liquid) plus the velocity head (in feet) at the point of gauge attachment.*

$$h_s = h_{ss} - h_f \text{ suct}$$

Total Discharge Head. Total discharge head (h_d) is the reading of a pressure gauge at the discharge flange of a pump (corrected to the pump center line† and converted to feet of liquid) plus the velocity head (in feet) at the point of gauge attachment.*

---

* The text version of these formulas applies to an existing pump installation. The symbol version, following the text version may be used when estimating before a pump is installed.

† On vertical pumps the correction should be made to the eye of the suction impeller.

$$h_d = h_{sd} + h_f \text{ discharge} + \text{velocity} = \text{head loss at discharge of system}$$

Total Head. Total head (H) on a pump is the total discharge head minus the total suction head or plus the total suction lift.*

$$H = h_d - h_s \qquad \text{or} \qquad H = h_d + l_s$$

Net Positive Suction Head (NPSH). Net positive suction head is the head available at the entrance or eye of a pump impeller to move and accelerate the water entering the eye. It is the gauge pressure at the suction flange of the pump (corrected to the pump center line† and converted to feet of liquid) minus the gauge vapor pressure in feet of liquid corresponding to the temperature at the point of measurement plus the velocity head (in feet at the point of measurement).*

$$\text{NPSH} = \text{Atm} + h_s - \text{vp} = \text{Atm} + h_{ss} - h_f \text{ suct} - \text{vp} \qquad \text{or} \qquad \text{NPSH} = \text{Atm} - l_s - \text{vp} = \text{Atm} - l_{ss} - h_f \text{ suct} - \text{vp}$$

When the NPSH required by a pump is known and it is desired to find the static suction lift which the pump will handle, the following form of the formula is convenient:

$$\text{Max static suct lift} = \text{Atm} - \text{NPSH} - h_f \text{ suct} - \text{vp}$$

Suction Limitations on a Pump

The maximum theoretical suction lift would be a distance equivalent in feet of liquid to the atmospheric pressure less the head equivalent to the vapor pressure of the liquid, less the head equivalent to the pressure of any gas in solution, less the friction head, and less the head equivalent to the entrance loss. Practically, however, the suction lift that can be handled without cavitation and vibration in the pump is somewhat less than the theoretical. The actual suction lift obtainable depends not only on the above factors but also on the characteristics of the liquid, the total head, and the pump speed, capacity, and impeller design. Abnormally high suction lifts usually cause serious reductions in the capacity and efficiency of the pump and often lead to serious troubles from vibration and cavitation. Every pump manufacturer knows the suction limitations of each of his pumps. Therefore, when requesting quotations on a pump for a particular job, the suction conditions should be given so that the manufacturer can recommend a proper pump for those conditions.

Work Performed in Pumping

To move a liquid against gravity with a pump, work must be expended. A pump may actually raise the liquid, force it into a pressure vessel, or merely give it enough head to overcome pipe friction. No matter what the service required of a pump, all forms of energy imparted to the liquid in performing this service must be accounted for in establishing the work performed. In order that all these forms of energy may be algebraically added, it is customary to express them all in terms of head expressed in feet of liquid.

In order to determine the work required of a pump, it is necessary to know the total dynamic head and the weight of liquid to be pumped in a given time. Usually weight is not given but rather volume in gallons per minute or barrels per hour, along with the density or specific gravity. The weight can be calculated from this information, but this is not necessary since the horsepower formulas are usually given in terms of gallons per minute or brake horsepower. Theoretical horsepower of a pump is usually called the hydraulic horsepower.

$$\text{Hyd hp} = \frac{\text{gpm}(\text{H})(\text{s})8.33}{33{,}000} = \frac{\text{gpm}(\text{H})(\text{s})}{3{,}960} = \frac{\text{bph}(\text{H})(\text{s})}{5{,}660}$$

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* The text version of these formulas applies to an existing pump installation. The symbol version, following the text version, may be used when estimating before a pump is installed.

† On vertical pumps the correction should be made to the eye of the suction impeller.

$$\text{Hyd hp} = \frac{(\text{gpm})(H_p)}{1{,}714} = \frac{(\text{bph})(H_p)}{2{,}450}$$

$$\text{Brake hp} = \frac{\text{gpm}(H)(s)}{(3{,}960)(\text{pump eff})}$$

The actual or brake horsepower of a pump is greater than the theoretical or hydraulic horsepower by the amount of the losses incurred in the pump through friction leakage etc. The efficiency of a pump is, therefore, measured as

$$\text{Pump eff} = \text{hydraulic hp/brake hp}$$

Other useful formulas are

For motor-driven pumps:

$$\text{Kw per 1,000 gal per hr of liquid} = \frac{0.00315(s)(H)}{(\text{pump eff})(\text{motor eff})}$$

For direct-acting steam pumps:

$$\text{Duty in terms of ft-lb per 1,000 lb steam} = \frac{\text{theor hp}(1{,}980)1{,}000{,}000}{\text{total steam per hr, lb}}$$

Sometimes theoretical horsepower for figuring duty is calculated using static head instead of total dynamic head. In this case duty is in terms of foot-pounds of useful work done per 1,000 lb steam. As given above the formula is in terms of foot-pounds of actual work done per 1,000 lb of steam.

Symbols Used in This Subsection

A = flow area of conduit, sq ft
Atm = atmospheric pressure, ft of liquid absolute
bph = flow in barrels (42 gal) per hr
d = inside diameter of circular conduit, in.
g = acceleration due to gravity, ft/sec^2 at sea level and 45° latitude; this is 32.174 fps, and this value has been used in conversions
gpm = flow, gpm
H = total head, ft of liquid
H_p = total head, psi
h_d = discharge head, ft of liquid
h_f = friction head, ft of liquid
h_s = suction head, ft of liquid
h_{sd} = static discharge head, ft
h_{ss} = static suction head, ft
h_{ts} = total static head, ft
h_v = velocity head, ft
l_s = suction lift, ft of liquid
l_{ss} = static suction lift, ft
Q = quantity of flow, cfs
s = specific gravity (water at 60°F = 1)
v = velocity of flow, fps
vp = vapor pressure of liquid, ft of liquid absolute

WATER-HAMMER PROTECTION FOR PUMP DISCHARGE LINES

BY JOSEPH SEGEL

Mechanical Engineer (Retired)

DEFINITIONS AND ELEMENTARY THEORY

Water hammer is the rise in pressure that occurs when the water flow in a pipe line or closed conduit is reduced or stopped, regardless of how this flow is checked. Hammer can occur in any line carrying liquids. Usually, the problem is important only in water lines.

The classical water-hammer problem is that of protecting penstocks that supply water to water turbines. If the flow in the penstock is suddenly reduced by action of the gates controlling the water flow to the turbines, the penstock pressure at the gate immediately rises, then drops below the initial pressure, and continues to surge in regular periods until friction damps the surges out.

Pump-discharge-line pressures will surge in the same way if all the operating pumps are stopped at once, except that the initial surge which occurs at the pumping end of the line is downward. The rise in pressure following the downward surge may be sufficient to rupture the pipe or pull the line apart unless protection is provided.

FORMULAS

The formulas most useful in water-hammer problems are (ASME, ASCE symbols)

$$a = 4{,}720/\sqrt{1 + (Kd/Ee)}$$
$$h_{\max} = aV_0/g$$

Period of pipe line = $2L/a$ = one interval, seconds

where a = the velocity of the pressure wave in the pipe line, fps. (This is the velocity of sound in the water of the pipe line.)

K = the bulk modulus of the liquid in the pipe line, psi (approx. 300,000 for water)

E = Young's modulus for the pipe-line material psi (30 million for steel)

d = the diameter of the pipe

e = the wall thickness of the pipe; d and e are always in the same units, *i.e.*, both feet or both inches.

$h_{\max}$ = the maximum pressure rise (or drop) that can occur if the water flow is stopped instantly and there are no complicating features such as nonuniform pipe, branch lines, etc. The unit of $h_{\max}$ of the formula is in feet of the liquid flowing.

V_0 = the velocity in the pipe line just previous to the disturbance causing water hammer, fps

g = gravity, ft/sec²

L = the length of the pipe line, ft

Other terms used in the argument are

h = the hammer pressure actually developed, ft

H_0 = the operating or steady-state pressure in the pipe line just previous to the disturbance causing water hammer, ft

T = the actual time in which the flow is stopped, sec

The value of a normally is somewhere between 3,000 and 3,500 fps. An accurate value usually is not needed for estimating surges in pump discharge lines.

The maximum water-hammer pressure which can occur in a simple pipe system is h_{max}, but this is not the maximum pressure in the pipe line because h_{max} adds to H_0 the existing pressure in the line just previous to the disturbance. The maximum pressure existing is, then, h_{max} plus H_0, and if H_0 is near the design pressure, it may be noted that rupturing pressures may occur. The water-hammer pressure h_{max} is alternately both positive and negative. When negative, the minimal pressure is $H_0 - h_{max}$. Obviously, the pipe-line pressure cannot reduce below zero absolute. If the downward surge carries to zero, vacuum will form, and the collapse of this vacuum may cause a major water-hammer blow. It may be noted that h_{max} is independent of the length of the line and depends only on the water and the wave velocities. The value of h_{max} is usually between 40 and 45 psi per fps of water velocity destroyed.

The period of the pipe line $2L/a$ is, by inspection, merely twice the length of the line divided by the wave velocity. It is the time it takes the pressure wave front to travel from the point of origin to the end of the line or other reflection point back to the origin. The wave of rarefaction, or subnormal pressure wave, also takes $2L/a$ sec to make the round trip. The time by stop watch between two crests of the pressure wave is then $4L/a$ sec. The value of $2L/a$ is important because the actual hammer h is reduced below h_{max} if the time it takes the flow to stop is greater than $2L/a$. This is true because the subnormal phase of the wave reaches the point of origin before the pressure wave has ceased to build up and adds algebraically to the pressure wave. The value of h is for ordinary lines about equal to

$$h_{max}/T/2L/a = h_{max}2L/aT,$$

with the understanding that in this formula T cannot be less than $2L/a$. This formula holds reasonably well except for lines of high velocities and low pressures, in which cases h is much increased. In general, T should be five to ten times the value of $2L/a$ to eliminate destructive water hammer.

On power failure, the pressure drop at the pumps is never h_{max} except for very long lines of high H_0. Short lines, except those of high water velocities, have very little pressure drop since T is large compared to the $2L/a$. Such lines usually need no hammer protection. For intermediate lines the pressure drop on power failure is about one-half aV_0/g. The subsequent pressure rise may be slightly more than that unless a vacuum has formed, in which case nearly the full hammer develops.

PROTECTION AGAINST WATER HAMMER

If the water velocities are low, say, below 2 fps, and the lines are comparatively short (less than 1,500 ft), usually no protection is needed. Water-hammer pressures in other pump discharge lines following power failures can be kept within safe limits either by lengthening the flow-stopping time T to several intervals of $2L/a$ or by spilling water to relieve the pressure, or by both. The measures that may be taken to obtain these effects are listed below.

Stopping the flow in several pipe-line intervals:

Flywheels on pumps
Air chambers
Air aspiration
Air injection
Air valves (These also spill water)

Relieving the pressure by discharging water:

Slow-closing checks at the pumps
Relief valves, several types

There may be other systems of protection, or the line may be built strong enough to take the hammer without damage.

The protection equipment to be used depends mainly on the pressure at the pump end of the line H_0, the contour of the line, the terminal conditions, the maximum

water velocity, the length of the line, and the design pressure of the pipe as compared to the operating pressure.

It is perhaps obvious that, if there are adequate flywheels on the pumps, the discharge lines will not be subjected to appreciable water hammer, regardless of the factors mentioned above. Flywheels can be, but usually are not, used mainly because of costs and energy loss.

Properly proportioned and operated air chambers protect any line. They are, however, seldom used because of high costs and the difficulties of maintaining the air pressure in the chambers.

Air Aspiration. Air aspiration can be used on lines which are fairly level and which discharge into reservoirs having little elevation over the pump heads. A line of this type is illustrated in Fig. 8-185. When there is power failure and no protection, the pressure drops below atmosphere and may go to nearly absolute-zero pressure if the distance from the water level to the pump head exceeds the barometric height. Now, if ordinary brass or bronze check valves are installed in the pipe wall in such a way that air will be drawn into the pipe when the pressure drops below atmosphere, the forward motion of the water (on power failure) is then not retarded, and there will be several intervals before the flow stops. The return velocity—by pipe-line breathing, aided perhaps by the head from the reservoir or other elevation is now much reduced. This, coupled with the air cushion trapped in the pipe, prevents any destructive water hammer.

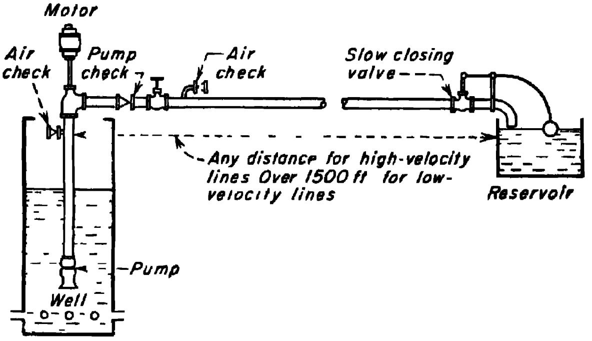
Motor
Air check
Pump check
Air check
Slow closing valve
Any distance for high-velocity lines Over 1500 ft for low-velocity lines
Reservoir
Pump
Well

Fig. 8-185. Aspirated-air protection. Air checks at pump house only.

This type of protection can be used on any system in which the elevation at the reservoir is not more than 40 ft above the pump head. At greater elevations—unless the water velocity is unduly high or the pipe is very long—the downsurge does not carry well below atmosphere, and protection is impaired. These checks should also be placed at other high points in the line or its connections where a vacuum might develop. If a portion of the line is above the hydraulic gradient (pipe-line pressure below atmosphere), the checks should, of course, be located at the points where the water pressure is normally positive.

The size of the air checks to be used is not critical. Ordinarily, the equivalent of a 2-in. check is sufficient for a 30-in. line. Excess check-valve capacity is no detriment. The checks should be installed in multiple since, if one fails to open, there will be at least partial protection. The check connections should be short to eliminate delay in air inflow upon power failure. It is necessary to protect the checks against frost in exposed locations.

Air-check protection is fully applicable to long flat lines. It is not so applicable to lines having a terminal rise unless the design pressure appreciably exceeds the operating pressure because the surges usually exceed the operating pressure.

The second air check shown in Fig. 8-185, although of some use in reducing hammer in the main line, is installed primarily to prevent destructive water hammer in the pump riser. If the riser exceeds the barometric height, a vacuum will form when the pump is shut down if there is no air inlet. If the pump is immediately started, there is nothing to retard the water flow until the check valve is reached, and a full water-hammer blow results. Air in the riser prevents this.

Air Injection. If the pump discharge lines are of medium length and one-half aV_0/g subtracted from H_0 is not well below atmosphere, they can be protected by the use of an air tank as shown in Fig. 8-186. The tank air pressure is maintained at 10 psi (approx.) below the pipe-line pressure by means of an air compressor controlled by a pressure switch. On power failure the pipe-line pressure drops, air enters, and the water continues flowing for a sufficient length of time to reduce the hammer surge.

The size of the air tank should be such that, on the downsurge following power failure, the air continues to displace water for a period of approximately three times $2L/a$ sec. The tank size needed then varies directly with the expected maximum flow (cfs) and the length of the line and, inversely, as the pressure. A line 3,000 ft long, flowing water at 30 cfs and at a pressure of 50 psig would be adequately protected by an air tank of 50-cu ft volume. The connection between the tank and the pipe line may be 2 in. diameter per 30 in. diameter of water pipe.

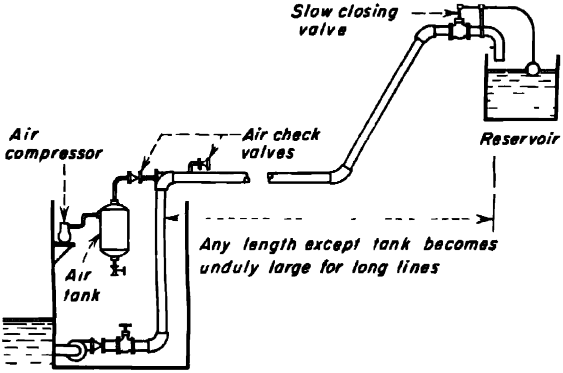

FIG. 8-186. Injection-air protection.

The installation needs operating attention only to service the compressors and its control and to drain leakage water from the tank.

Both this system and that using aspirated air are unsuitable if it is imperative to keep air out of the water line. Both systems are also unsuitable if the operating pressure is very little below the design pressure because some surges above operating pressure will occur in all lines except those which are flat.

Aspirated or injected air, if it can be tolerated, is also effective in reducing cavitation which is fundamentally repeated water hammer on a small scale.

Air Valves. If the line will aspirate air on power failure, and it is important to keep air out of the line and also to prevent any hammer pressure above the operating pressure, the air valve may be used. This valve opens on a drop in pressure below atmosphere but closes slowly under dashpot control when pressure returns. The valve, therefore, discharges air first and then water on the return surge, thereby reducing the surge pressure to safe limits. It is advisable to install an air check in parallel with the air valve as auxiliary protection should the air valve be inoperative because of freeze-up or for any other reason. The air valve should not be installed unless there is a substantial static pressure, say, 30 ft of water or more in the line, since otherwise the valve will not close automatically.

The location for the air valve is the same as that for air-check protection, *viz.*, the high spots in the line near the pump or pumps and elsewhere.

The size and number of air valves may be selected on the basis that one 6-in. valve at each high spot will protect a 30-in. water line.

Slow-closing Checks. If pump operating pressures, pipe-line elevations, water velocities, length of lines, and other conditions are such that air aspiration or injection is not suitable, some way of relieving the return water-hammer surge by spilling water becomes necessary. Slow-closing checks in the pump discharge lines may be used for the purpose. These checks are provided with controls to prevent the checks from closing immediately on a reversal of water flow. These valves are usually hydraulically operated, and the closing rate can be predetermined.

Slow-closing-check protection can be used on lines of the type shown by Fig. 8-187. Upon power failure, the line acts as two separate lines. The column of water separates at B, and the section A to B surges separately from that part of the line from B to C. The section A-B is protected by spilling water backwards through the checks and the pumps. For full protection, the closing time of the checks should be not less than five times the period ($2L/a$) of the pipe line A to B. This time is not excessive if the line A-B is short, say, from 1,500 to 3,000 ft. Shorter lines ordinarily need no protection. Longer lines require such slow closing that the backflow of water may damage the pumps.

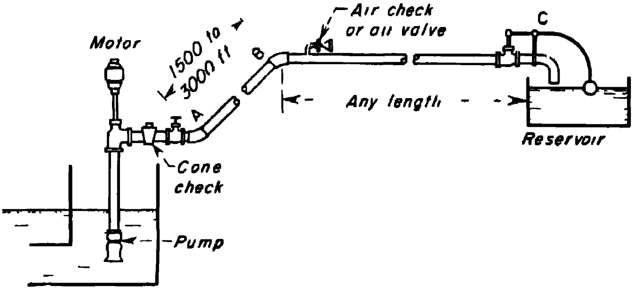

FIG. 8-187 Slow-closing pump-check and aspirated-air protection

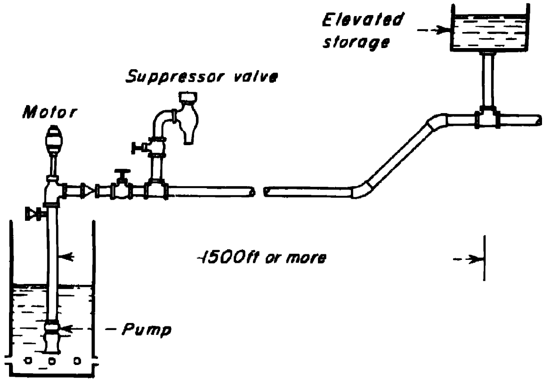

FIG. 8-188 Surge suppressors protect entire line

The lines like A-B of Fig. 8-187 and any other line in which the pressure does not drop below atmosphere on power failure can be protected by surge suppressors. These valves are installed on the pump discharge pipes and on the line side of the check valves. The suppressor valves open if all the pumps stop, and the water then discharges to waste. The effect of the valve action is the same as that of the slow-closing checks, but the pumps are now by-passed and, therefore, are not distressed. A typical installation is shown by Fig. 8-188. These valves open on interruption of electric current to solenoids. A pressure switch set to open the solenoid circuit when the water pressure drops is a usual auxiliary. Consequently, the valves open if the pumps stop for any reason. The opening and closing times of the valve are adjustable. The opening cycle of the valve may be set at 2 to 4 sec. The closing time should be at least five times the pipe-line period ($2L/a$) for ordinary short lines and ten times for long lines or short lines having high water velocities. The valve is hydraulically operated and closes automatically, regardless of the presence or absence of current in the solenoid circuit. Electricity supplies the trigger action only.

The water-discharge capacity of the valve or valves may be somewhat less than the maximum estimated flow in the line to be protected. It is advisable to install two or more valves rather than one large valve. If there is a change in size in the pumping main, such as usually exists in a well field, one suppressor should be installed at or near the point where the full-sized main begins.

These valves, if properly selected, installed, and operated, will prevent any pressure rise in excess of the design pressure.

Spring-loaded relief valves are not used for water-hammer protection. They open too late, and they may set up a cyclic disturbance exceeding even the original water-hammer surge.

Hydraulically operated relief valves without the solenoid feature are adapted to control surges that do not have steep wave fronts. These valves open on a pressure rise and close slowly when the pressure drops. They are mostly useful to protect lines having booster pumps where the accumulated shut-in pressure of the pumps in series exceeds the design pressure of the lines. These valves may be installed in out-of-the-way locations where some hammer protection is desirable, but electric current is not available.

INDUSTRIAL WATER-COOLING SYSTEMS*

BY L. T. MART

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FUNCTION

In nearly all industrial processes large quantities of heat must be dissipated. This is most commonly accomplished by circulating water through heat exchangers. The water thus heated, if conserved and reused, must be cooled, and this is accomplished in the water-cooling system. In actual practice the cycle is continuous, and the heat added to the water in the heat exchangers equals the heat transferred from water to air in the water-cooling system. Cooling and reusing water is quite popular and offers many advantages over the use of water from wells, lakes, or streams.

TYPES OR CLASSES

There are many styles of water-cooling systems. One general classification is the spray-pond type, and another is the water-cooling tower. The latter can be classed as natural or mechanical draft, depending upon whether fans are or are not used for moving air through the structures. Atmospheric water-cooling towers fall in two general classes, *viz.*, the spray tower and the deck-type tower. Mechanical-draft towers may be classed as forced or induced draft, depending upon the location of the fans. Nearly all industrial towers are of wood construction, whereas those located in large cities and restricted fire zones must be noncombustible and therefore of steel or masonry.

THEORY OF PERFORMANCE

The generally accepted theory of the cooling of water assumes that heat is transferred from water to air both by convection (sensible heat) and evaporation (latent heat). The scientific explanation generally accepted was well offered by Walker, Lewis, and McAdams[1]

This theory was further developed by Merkel in 1925 and by many others since. The basic equation can best be expressed as follows:

$$L\,dT = Ka\,dV\,(h'' - h) \qquad \text{or} \qquad (KaV/L) = \int_{T_1}^{T_2} [dT/(h'' - h)]$$

where K = mass-transfer coefficient based on vapor-content potential
a = sq ft of exposed water surface per cu ft of tower
V = active tower volume, cu ft per sq ft
L = water rate, psf
T = water temperature
h'' = enthalpy of saturated air at water temperature
h = enthalpy of main air stream

This equation offers a basis for selecting the difference in enthalpies as the driving force to account for the transfer of both sensible and latent heat in cooling-tower performance and also shows why performance is determined by wet-bulb temperatures.

The left-hand side of the equation (KaV/L) represents the tower coefficient, while the mechanical or graphical integration of the right-hand side gives an evaluation of the performance conditions.

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* Superior numbers refer to specific references listed at the end of this subsection.

DESIGN AND APPLICATION ENGINEERING

Wet-bulb and Dry-bulb Temperature. The dry-bulb temperature is that temperature which is ordinarily read on any (dry) commercial thermometer. The wet-bulb temperature is that lower temperature to which air can be cooled when fully saturated with water vapor. The wet- and dry-bulb temperatures are equal when air is fully saturated. Relative humidity is a measure of the actual per cent of moisture in air by weight to the total amount possible. The relationship between wet bulb, dry bulb, and relative humidity is given in Table 8-54.

Table 8-54. Per Cent Relative Humidity (Barometric Pressure 29.92 in. Hg)

Dry-bulb temperature	25	30	35	40	45	50	55	60	65	70	75	80	85	90	100
	Wet-bulb temperature														
60	44 5	45 7	46 9	48 0	49 0	50 1	51 2	52 1	53 1	54 1	55 1	56 1	57 1	58 1	60
65	48 0	49 4	50 9	52 0	53 1	54 2	55 6	56 8	58 0	59 0	60 0	61 1	62 0	63 0	65
70	51 5	53 0	54 4	56 0	57 1	58 5	59 9	61 0	62 1	63 4	64 9	65 9	66 9	68 0	70
75	55 0	56 5	58 1	59 8	61 1	62 8	64 0	65 2	66 8	68 0	69 1	70 1	71 5	72 9	75
80	58 5	60 0	61 9	63 9	65 1	66 9	68 2	69 9	71 0	72 4	74 0	75 2	76 5	77 7	80
85	61 8	64 0	65 8	67 7	69 3	71.0	72 8	74 5	76 0	77 1	78 8	80 0	81 3	82 8	85
90	65 0	67.2	69 2	71 4	73 0	73 7	77 0	78 5	80 0	81 9	83 0	84 5	85 9	87 1	90
95	68 5	70 9	73 1	75.3	77.1	79 1	81 0	83 0	84 7	86 0	88 0	89 5	91 0	92 0	95
100	71 9	74 9	76 9	79 4	81 5	83 9	85 7	87 9	89 1	91 0	92 0	94 0	95 9	97 1	100
105	75 6	78 0	80 5	83.8	85 9	88 0	90 0	92 0	93 9	95 9	97 3	99 0	100 9	102 0	105
110	79 0	82.0	84 9	87 9	90 0	92 1	94 4	96 8	98 5	100 0	102 0	103 6	105 1	107 0	110
115	82 9	86.0	89 0	92 0	94 5	96 9	99 1	101 1	102 9	105 0	107 0	108 5	110 4	112 0	115
120	86 2	90.0	93 0	96 0	98 9	101 5	103 2	106 0	107 9	109 9	111 9	113 9	115 1	116 9	120

Total Heat of Air. At any given wet-bulb temperature the total heat of air remains constant regardless of humidity or dry bulb. With ingoing and outgoing wet-bulb temperature known, the total heat carried off by each pound of air can be calculated for any given heat load if ingoing and outgoing wet bulbs are known. The properties of air are shown in Table 8-55.

Table 8-55. Psychrometric Table

Temperature	Enthalpy, Btu per lb dry air (based on wet-bulb temperature)	Volume, cu ft mixture per lb dry air (based on dry bulb temperature)	Weight, lb per 1,000 cu ft mixture (based on dry bulb temperature)	Saturation press, psi (based on dew-point temperature)	Saturation, humidity ratio, grains water per lb dry air (based on dew point temperature)
50	20 25	12 99	76 98	0 1780	53 38
55	23 15	13 15	76 05	0 2140	64 34
60	26 37	13 32	75 08	0 2561	77 21
65	29 96	13 49	74 13	0 3054	92 40
70	33 96	13 68	73 10	0 3628	110 2
75	38 46	13 87	72 10	0 4295	131 1
80	43 51	14 08	71 02	0 5067	155 5
85	49 24	14 30	69 93	0 5960	184 0
90	55 70	14 53	68 82	0 6980	217 1
95	63 05	14 79	67 61	0 8149	255 6
100	71 40	15 07	66 36	0 9487	300 5

Design Wet Bulbs. Water can be cooled only to the wet-bulb temperature after which no heat transfer can take place. The commercial cooling system never cools to that theoretical limit but is designed to accomplish a portion thereof. Accurate information on weather and design wet bulb is necessary in connection with water-cooling problems. Suggested safe design wet bulbs are given in Table 8-56. Two

Table 8-56. Design Wet-bulb Temperatures Based on Three Summer Months

City	State	Average noon temperature	Not exceeding 100 hr
lbany	New York	65	73.0
lbuquerque	New Mexico	61	67.0
marillo	Texas	65	71.0
tlanta	Georgia	70	78.0
ugusta	Georgia	73	79.0
aker	Oregon	54	65.0
altimore	Maryland	68	80.0
ellefonte	Pennsylvania	65	74.0
irmingham	Alabama	73	78.0
ismark	North Dakota	60	69.0
oise	Idaho	57	67.0
oston	Massachusetts	64	74.0
rownsville	Texas	76	80.0
uffalo	New York	62	74.0
amden	New Jersey	67	78.0
harleston	South Carolina	76	80.0
heyenne	Wyoming	54	65.0
hicago	Illinois	65	76.0
incinnati	Ohio	68	78.0
leveland	Ohio	63	75.0
oncordia	Kansas	67	76.0
allas	Texas	71	78.0
enver	Colorado	56	65.0
es Moines	Iowa	66	75.0
etroit	Michigan	64	74.0
evils Lake	North Dakota	59	68.0
astport	Maine	57	70.0
lkins	West Virginia	64	73.0
l Paso	Texas	64	70.0
vansville	Indiana	69	79.0
argo	North Dakota	61	70.0
resno	California	63	71.0
t. Smith	Arkansas	71	78.0
rand Junction	Colorado	58	66.0
reen Bay	Wisconsin	62	75.0
reenville	South Carolina	69	76.0
elena	Montana	55	64.0
oughton	Michigan	58	72.0
ouston	Texas	74	80.0
uron	South Dakota	63	73.0
ndianapolis	Indiana	66	77.0
ackson	Mississippi	74	80.0
acksonville	Florida	74	80.0
ansas City	Missouri	68	77.0
noxville	Tennessee	70	77.0
a Crosse	Wisconsin	65	78.0
aguna Beach	California	65	68.0
exington	Kentucky	67	77.0
incoln	Nebraska	67	74.0
ittle Rock	Arkansas	72	80.0
os Angeles	California	64	68.0
ouisville	Kentucky	69	78.0

Table 8-56. Design Wet-bulb Temperatures Based on Three Summer Months (*Cont.*)

City	State	Average noon temperature	Not exceeding 100 hr
Medford	Oregon	59	63.0
Memphis	Tennessee	72	80.0
Miami	Florida	76	80.0
Milwaukee	Wisconsin	63	75.0
Mobile	Alabama	75	80.0
Moline	Illinois	67	78.0
Mondena	Utah	56	64.0
Montgomery	Alabama	73	79.0
Newark	New Jersey	66	77.0
New Haven	Connecticut	65	76.0
New Orleans	Louisiana	75	81.0
New York	New York	66	77.0
North Platte	Nebraska	64	73.0
Oklahoma City	Oklahoma	71	78.0
Omaha	Nebraska	66	77.0
Phoenix	Arizona	69	76.0
Pittsburgh	Pennsylvania	65	75.0
Portland	Maine	61	71.0
Portland	Oregon	59	68.0
Pueblo	Colorado	59	66.0
Raleigh	North Carolina	71	78.0
Rapid City	South Dakota	60	69.0
Reno	Nevada	55	64.0
Sacramento	California	63	72.0
Salt Lake City	Utah	57	64.0
San Antonio	Texas	74	78.0
San Diego	California	64	72.0
San Francisco	California	57	64.0
Savannah	Georgia	75	80.0
Sault Ste. Marie	Michigan	59	71.0
Seattle	Washington	56	66.0
Shreveport	Louisiana	74	80.0
Sheridan	Wyoming	58	65.0
Sioux City	Iowa	65	75.0
Spokane	Washington	55	64.0
Springfield	Missouri	68	76.0
Springfield	Illinois	68	78.0
St. Louis	Missouri	69	77.0
St. Paul	Minnesota	63	75.0
Toledo	Ohio	64	74.0
Tulsa	Oklahoma	71	78.0
Vicksburg	Mississippi	74	80.0
Washington	District of Columbia	68	79.0
Wichita	Kansas	68	76.0
Williams	California	57	73.0
Williston	North Dakota	58	67.0
Wilmington	North Carolina	73	79.0
Yellowstone Park	Wyoming	51	60.0
Yuma	Arizona	72	81.0

columns of wet-bulb figures are shown. The first is recommended for average year-round industrial use, and the second is for freon air conditioning which must produce maximum load during hottest weather. For steam-jet refrigeration even higher wet bulbs are recommended. Factors affecting actual wet-bulb choice are many, and they include (1) operating load factor, (2) total annual days of operation, (3) capital costs, (4) elevation above sea level, (5) overload protection, and added minor influences.

Pipe Sizes for Circulating Water. Two factors must be taken into consideration when determining the size pipe to use in distributing water. The larger the pipe, the more the initial expense and the less the friction loss.

If the distribution system is of considerable length and required to operate over long periods, larger pipe should be used as the pumping costs will thus be low. If the system piping is short and if the operating season is brief, then the smaller pipe sizes are recommended.

Some suggested pipe sizes and capacities are given in Table 8-57. Column A should be used for a continuous distribution system, Column B for an intermittent use, and Column C for a water-distribution system that only operates a small part of the time, *e.g.*, an air-conditioning layout.

Table 8-57. Pipe Sizes and Allowable Gpm—Standard-weight Steel Pipe

Pipe size, in.	Inside diameter	Gallons per minute		
	Standard-weight steel pipe	A Continuous flow	B Intermittent flow	C Occasional flow
		30 ft friction loss per 1,000 ft of pipe	50 ft friction loss per 1,000 ft of pipe	75 ft friction loss per 1,000 ft of pipe
½	0.622	1.2	1.65	2.00
¾	0.824	2.55	3.35	4.2
1	1.049	4.76	6.30	7.89
1¼	1.380	9.85	13.15	16.80
1½	1.610	14.9	19.6	22.5
2	2.067	28.8	37.7	47.1
2½	2.469	46.1	59.9	74.7
3	3.068	98.4	131.5	167.8
3½	3.548	113.5	150.0	186.6
4	4.026	164.0	217.5	270.5
5	5.047	300.0	395.0	490.0
6	6.065	485.0	640.0	796.0
8	7.981	1,000.0	1,305.0	1,626.0
10	10.02	1,810.0	2,385.0	2,960.0
12	12.00	2,910.0	3,820.0	4,790.0
14	13.25	3,800.0	5,000.0	6,220.0
16	15.25	5,500.0	7,200.0	9,000.0
18	17.18	7,500.0	9,900.0	12,300.0
20	19.18	10,000.0	13,100.0	16,300.0
24	24.00	18,000.0	23,500.0	28,500.0
30	30.00	33,000.0	41,800.0	58,000.0
36	36.00	53,000.0	70,000.0	87,000.0

Sound. Sound-level measurements and noise control have become increasingly important. Cooling towers installed in industrial locations seldom present a serious problem since the ambient or background sound level is usually of about equal magnitude to that of the cooling tower. This is not true of cooling towers located in congested downtown, business, or hotel districts or in residential areas.

The response of the ear to sound is not proportional to the intensity but is more nearly proportional to the logarithm of the intensity as is shown in Fig. 8-189. The characteristic would be a straight line if the response of the ear were strictly logarithmic. Figure 8-189 also shows the loudness levels of familiar sounds.

The decibels scale, which is logarithmic, has been widely used for measurements in electrical communication, and it has logically been adopted for sound measurements. A decibel number represents a ratio rather than a definite quantity, so some reference point is necessary. This value for sound measurements is 10 watts per sq cm at a frequency of 1,000 cycles and actually represents approximately the threshold of hearing for a person whose ears are slightly more sensitive than average.

A decibel rating represents a ratio of powers which may be expressed by the equation

$$db = 10 \log (P_2/P_1)$$

where P_2 and P_1 represent the two powers being compared, and the decibel rating of two sound levels is not the sum of the two components.

The noise produced by a cooling tower is caused by the falling water and also by the fan on mechanical-draft towers. Fan noise may be minimized by installing oversize fans operating at lower speeds. Baffles constructed of sound-absorbent material may be used to absorb the sound or to deflect it.[2,3]

Delignification of Wood. The presence of sodium carbonate in the circulating water results in delignification of the wood structure of the tower. This is because the lignin which binds the wood fibers together is dissolved by the sodium carbonate, leaving the wood surface in a white, fibrous condition. Prolonged exposure reduces the structural strength of the wood. The rate of delignification is roughly proportional to the concentration of sodium carbonate present, and a concentration of 200 to 300 parts per million will result in damage to the wood structure and fill.

Delignification appears first on those parts of the tower which are alternately wet and dry since the evaporation at such points rapidly increases the concentration of dissolved solids. For this reason the action is far worse in atmospheric towers. Wood parts in mechanical-draft towers are continuously wet, and delignification is thus much slower.

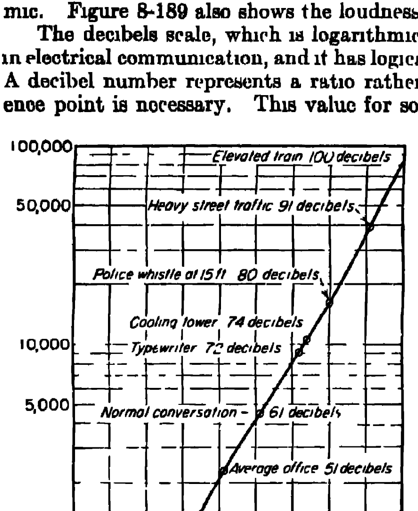


Fig 8-189 Loudness level in decibels for various sounds

The presence of sodium carbonate in harmful amounts is usually indicated by a high pH reading of 9 to 11. It should be understood, however, that the delignification is caused by the presence of sodium carbonate and not by the high pH value of the water. The sodium carbonate may be neutralized by the use of sulfuric acid which converts it into sodium sulfate. Such a treatment lowers the pH value of the water which should be maintained at 7.3 to 7.5.

Sodium carbonate seldom occurs naturally in the make-up water but is a product of water treatment. High concentrations are usually found when a zeolite water treatment is used It is therefore important to use sulfuric acid in connection with sodium zeolite water treatment to prevent delignification of the wood

Steam Condensing. When steam is condensed, the latent heat must be removed. This amounts to 960 Btu per lb of dry saturated steam. In actual operation the total heat may be more or less, depending upon the steam condition, as due allowance should be made for moisture or superheat being present

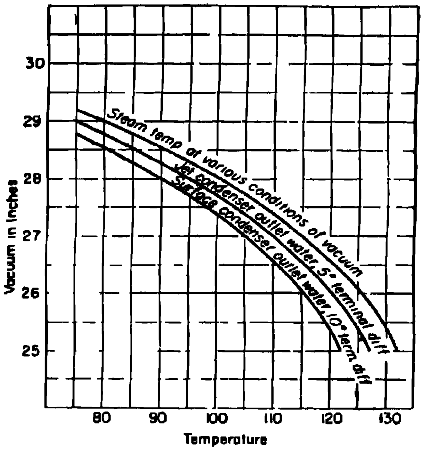


Fig 8-190 Relation between vacuum and temperatures for steam condensers.

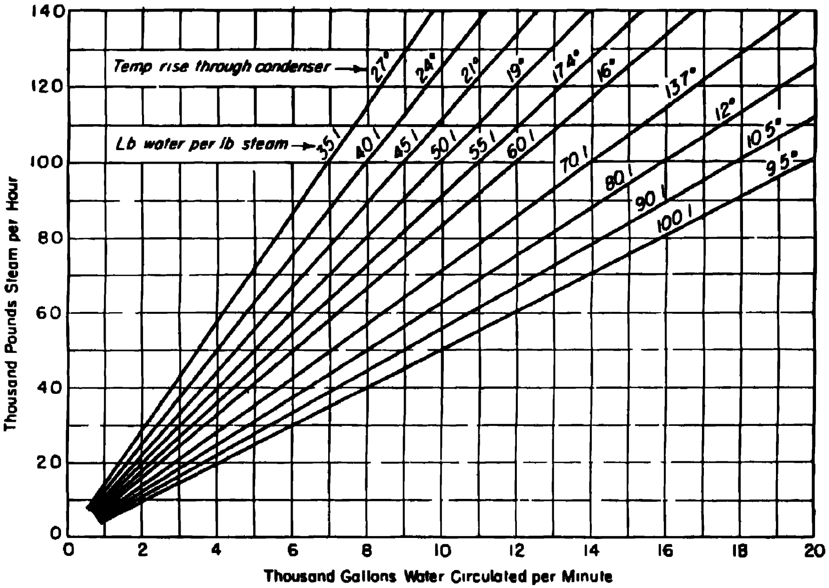


Fig 8-191 Flow relationship of steam to water for steam condensers, based upon 960 Btu per lb of steam

Allowable hot-water temperature leaving the steam condenser must be somewhat less than the condensing steam temperatures This is known as the "terminal difference" (Fig 8-190) For reciprocating-steam-engine work, a vacuum of 26 in is practical, and for steam-turbine operation vacuums of 27 to 28 in. are practical. When jet-type steam condensers are used, the outgoing hot water should be approximately five degrees cooler than the exhaust steam. When surface-type condensers are used, a difference of 8 to 10° is considered good practice (Fig. 8-191).

Excessive terminal difference indicates one of several things, including

1. Air or other noncondensable vapors in the condenser
2. Dirty or scaled-up cooling surface
3. Insufficient water velocity which reduces the heat-transfer rate
4. Too small a condenser

WATER-COOLING SYSTEMS

Spray Pond. The spray pond (Fig. 8-192) requires large areas and is practical only where ground is cheap and where drift is not objectionable to surroundings. If the soil is such that a concrete basin is required, the cost is prohibitive. If surroundings and drift conditions force the use of a louver fence, the spray pond is not recommended. Ponds are very low in first cost, and maintenance over a period of years is almost negligible. No serious icing problems occur in winter.

FIG. 8-192. Typical spray pond for cooling water with atmospheric air.

Systems are usually designed for from 5 to 6 lb pressure at the nozzles, and nozzles are elevated about seven feet above the surface of the water. Total pumping head required averages 25 ft on smaller systems and upward to 30 ft on large ponds.

For good practice space requirements will approximate 1.3 to 1.6 sq ft per gal of water sprayed. Large ponds should be relatively long and narrow. Actual width of the spraying section should be 25 ft for small single-run spray systems and should be held to not over 100 ft in large systems. An added clearance area of 15 to 20 ft on all sides should be allowed for drift prevention (Fig. 8-193).

Table 8-58. Recommended Spray-pond Standards

Wet bulb	Vacuum, in. Hg	Steam temperature	Surface condensers				Jet condensers			
			Hot water	Cold water	Range	Ratio lb H_2O/steam	Hot water	Cold water	Range	Ratio, lb H_2O/steam
80	27.0	114.8	103	91	12	80	108	96	12	80
80	27.25	111.8	101	90	11	86	106	95	11	86
80	27.50	108.4	99	89	10	95	104	94	10	95
75	27.25	111.8	100	87	13	73	106	93	13	73
75	27.50	108.4	98	86	12	80	104	92	12	80
75	27.75	104.8	96	85	11	86	102	91	11	86
70	27.50	108.4	97	83	14	68	104	90	14	68
70	27.75	104.8	95	82	13	73	102	89	13	73
70	28.00	100.8	93	81	12	80	100	88	12	80

The water is broken up into droplets only once in a spray pond and falls through a relatively short distance. This limits the practical cooling efficiency and usually results in more water being circulated than in a cooling tower.

Maximum practical cooling efficiency does not exceed 60 per cent in small single-run systems and seldom exceeds 50 per cent in large ponds if we consider performance to be measured as follows:

$$\text{Cooling efficiency} = \frac{\text{hot water} - \text{cold water}}{\text{hot water} - \text{wet bulb}} = \frac{\text{actual cooling range}}{\text{max poss. cooling range}}$$

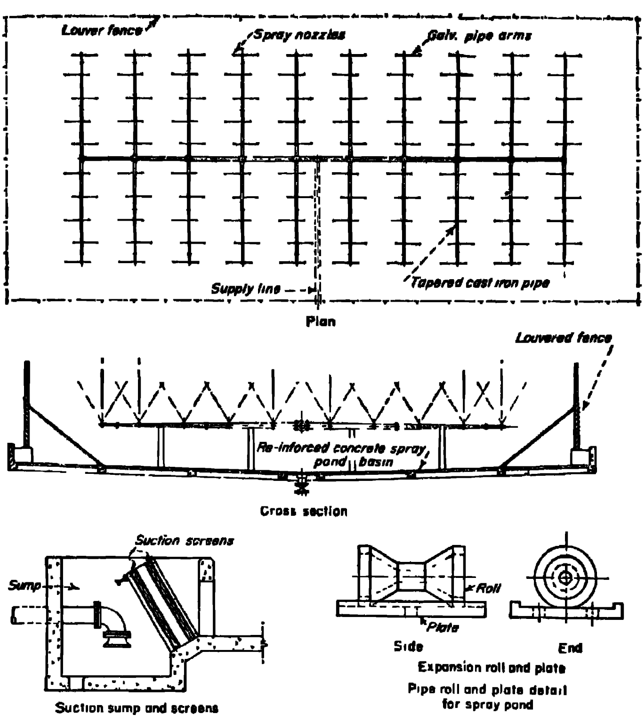


FIG. 8-193. Arrangement of spray pond showing piping, nozzles, louver-fence, water-sump, and pipe-expansion details.

Some recommended spray-pond standards for steam condensing are indicated in Table 8-58.

Atmospheric Spray Tower (Fig. 8-194). The atmospheric spray tower is very popular and practical in smaller sizes only. It is simple in design and easily erected by the average engineer or mechanic. A factory superintendent is not required to install it. Performance possibilities are limited to much the same as for a spray pond. Initial cost is low. Maintenance is easy, and repairs are negligible. Total pumping head is from 20 ft on the smaller units to 26 ft on larger sizes. Drift can be objectionable in high winds. When installed upon building roofs, it should be set back from the sidewalks to prevent drift on streets or other surroundings below.

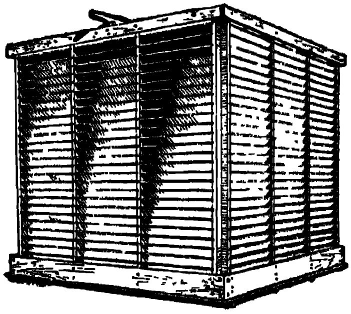


FIG. 8-194. Atmospheric-spray water-cooling tower.

Atmospheric Deck or Tray Tower (Fig. 8-195). In the atmospheric-deck type of cooling system, the hot water is introduced across the top by suitable distributing nozzles or flumes. It then cascades downward, splashing and dripping from deck to deck. Air passes horizontally through the structure, dependent upon existing breezes.

The effective width is usually about twelve feet, and height ranges from 28 to 36 ft. The total pumping head varies from 30 to 40 ft. Inclined louvers are required on all sides to prevent excessive drift. Velocity of the air through a well-designed unit will not exceed one-quarter to one-third the existing breeze available at the site. During periods of zero wind, some air moves vertically upward if the oncoming water is hotter than the air.

This style of tower performs well during periods of good breeze, but results are very poor during low-wind periods or with wind direction lengthwise rather than crosswise. Good exposure to prevailing breezes is essential.

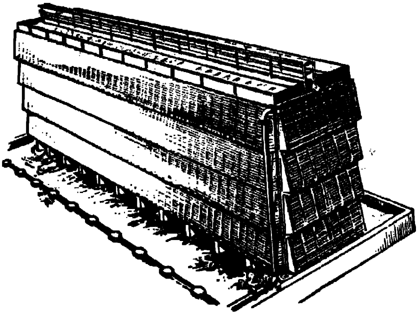


FIG. 8-195. Atmospheric-deck water-cooling tower.

Plan area required is a minimum for atmospheric towers. While very popular for many years, atmospheric towers are fast being outmoded by the modern improved mechanical-draft tower.

Table 8-59. Approximate Effective Areas Required for Atmospheric-deck Type of Cooling Towers

Cooling range, degrees	Square feet per gallon, approach to wet bulb		
	5°	10°	15°
5	0.864	0.312	
6	0.984	0.424	
7	1.10	0.520	
7½	1.15	0.562	0.230
8	1.20	0.604	0.264
9	1.30	0.672	0.332
10	1.38	0.744	0.392
11	1.44	0.808	0.452
12	1.50	0.856	0.496
13	1.57	0.912	0.536
14	1.61	0.952	0.576
15	1.66	1.00	0.624
16	1.70	1.04	0.656
18	1.77	1.10	0.724
20	1.83	1.18	0.780

Table 8-59 gives a rough approximation of the size or sectional area of tower required per gallon of water circulated. For instance, with a 15° range and 10° approach, 1 sq ft is required. A 2,000-gpm tower would require 2,000 ft of exposed area, and if the tower is 30 ft high, the length would be between 65 and 70 ft. Effective area is measured by height times length inside the louvers. The above table is for 70° wet bulb. With higher wet bulbs a reduced area is required for a given range and performance. With lower wet bulbs an increased area is required. Actual areas and wet-bulb correction factors vary considerably from the above figures, depending upon the make and style of tower being considered.

The Mechanical-draft Water-cooling Tower. Mechanical-draft water-cooling towers can be divided into two general classes. Forced draft pushes air into and through the tower, whereas induced draft pulls air from the tower. Most manufacturers prefer to build the induced-draft style. Mechanical-draft towers can also be classed as counterflow or crossflow, depending upon the direction of air flow as it passes through the active cooling surface.

Mechanical-draft towers have nearly constant air-to-water ratios and give uniform cooling results regardless of direction or velocity of wind. Performance tests can be easily conducted, whereas it is very difficult to test a spray pond or an atmospheric cooling tower because of variation in wind velocity and direction.

Mimimum space is required for mechanical-draft towers. Their external appearance is excellent, and there is no free moisture or drift loss to wet the surrounding ground or buildings. Because of drift elimination, they can usually be located much closer to the equipment they serve, thus reducing the cost of water lines to and from the equipment.

Table 8-60. Recommended Mechanical-draft Tower Standards

Wet bulb	Vacuum, in. Hg	Steam temp.	Surface condensers				Jet condensers			
			Hot water	Cold water	Range	Ratio	Hot water	Cold water	Range	Ratio
80	27.0	114.8	104	90	14	68	109	95	14	68
80	27.25	111.8	102	89	13	73	107	94	13	73
80	27.50	108.4	100	88	12	80	105	93	12	80
75	27.25	111.8	101	86	15	63	107	92	15	63
75	27.50	108.4	99	85	14	68	105	91	14	68
75	27.75	104.8	97	84	13	73	103	90	13	73
70	27.50	108.4	98	82	16	60	105	89	16	60
70	27.75	104.8	96	81	15	63	103	88	15	63
70	28.00	100.8	94	80	14	68	101	87	14	68

Table 8-60 indicates some suggested practical performance standards for steam-condensing operation. Three different vacuum conditions are indicated for each of three wet bulbs. The higher wet bulb is suggested for the Gulf Coast areas; the intermediate wet bulb is satisfactory for the central part of the United States; and the lower wet bulb is for northern locations.

Table 8-61. Approximate Effective Areas Required for Mechanical-draft Towers

Cooling range, degrees	Approach to wet bulb		
		10°	15°
5	0.313	0.174	
7½	0.412	0.237	0.154
10	0.489	0.287	0.195
15	0.598	0.373	0.200
20	0.670	0.438	0.311

Table 8-61 gives a rough approximation of the size or sectional area of tower required per gallon of water circulated. For instance with a 15° cooling range and a 10° approach 0.373 sq ft is required per gallon. Thus for a 10,000-gal tower, 3,730 sq ft are necessary, and if the tower picked has a width of 40 ft, the length must be somewhat over 90 ft, the exact dimension depending upon the manufacturer's standards.

Table 8-61 has been prepared for a 70° wet bulb. With higher wet bulbs a reduced area is required for a given range and performance. With lower wet bulbs an increased area is required. Actual areas and wet-bulb correction factors vary considerably from the above figures, depending upon the make and style of tower being considered.

Crossflow Water-cooling Towers. In the crossflow style (shown by Fig. 8-196) the water falls vertically downwards, and the air flows horizontally across the cooling section of the tower, whereas in the counterflow style of tower the air flows vertically upward. The crossflow principle has been popular in European practice for many years in both atmospheric and mechanical-draft equipment. In 1938, the first large-capacity crossflow cooling tower was introduced into American practice, and in a short time its popularity was unquestioned. In the larger industrial towers, instead of a single cooling tower per fan, the customary practice is to use one fan for each two cooling units. This results in what is known as the double-flow cooling tower shown by Fig. 8-197. The crossflow style of tower includes practically all the advantages but none of the disadvantages of counterflow towers.

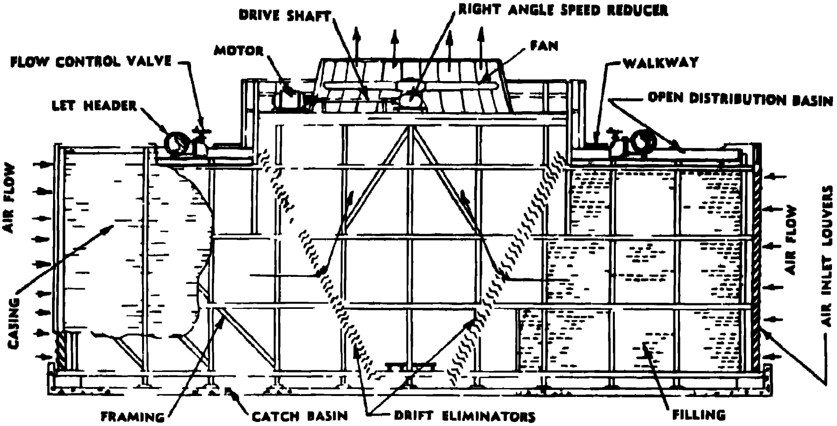


FIG. 8-196. Sectional elevation of a crossflow, also called double-flow, cooling tower.

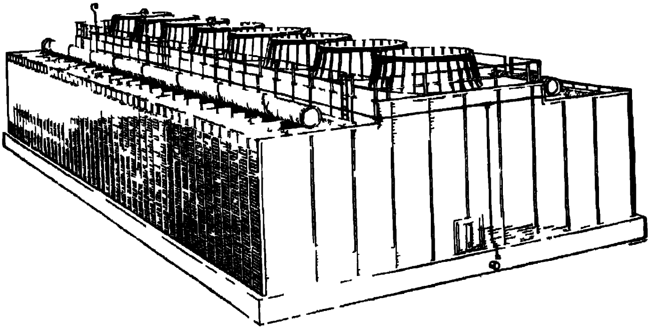


FIG. 8-197. Perspective view of a double flow water-cooling tower.

The crossflow tower is of minimum height and required pumping head. Being low and wide, it is easy to brace structurally for winds and earthquake. Accessibility to the mechanical equipment upon the top of the tower is easier than with conventional higher designs. Distribution of the hot water to the cooling surfaces of the tower is accomplished with a total head of only 6 to 8 in. of water.

The water-distributing system can be easily inspected at all times and while the tower is in operation. Mud can easily be removed, clogged nozzles can be cleaned, and most important of all, it is easy to adjust the float-control valves to make each tower cell take its proportionate quantity of water. These features are not possible in towers of counterflow design.

In conventional towers, the wider the framework, the higher must be the structure, and because of practical limits in the extreme larger sizes, two rows of fans are required, whereas with the crossflow tower the two cooling-tower cells may be set at any distance apart, and there is no limit to the maximum diameter of fans which can be used. Larger fans mean larger cells with fewer inlet water lines and fewer sets of electrical wiring and starting equipment, thus simplifying operation and maintenance. An extremely low draft loss is claimed for this style of tower because of less resistance to air flow past the slat fill as compared to the vertical or counterflow principle. The drift eliminator also provides a smaller draft loss because the eliminator area is much larger than the actual air-flow section of the tower proper. Because of the low draft loss, greater air per horsepower input to fans is obtained than with conventional towers.

OPERATION AND MAINTENANCE OF WATER-COOLING SYSTEMS

Spray pond, atmospheric towers, and mechanical-draft towers all require a regular maintenance schedule to keep them operating efficiently. The water-distribution problems are about the same in any water-cooling system.

Starting. When first starting operation or after a shutdown all water basins should be clean and free of any dirt or foreign matter, and they should be watertight. All pipe connections, valves, and nozzles should be checked to make certain they are in good working order. The cold-water basin should be filled with water, and then the system should be started operating. The water-distribution system should be inspected regularly to make certain the nozzles are spraying properly and that the evaporation and drift loss are not excessive. Low nozzle pressure indicates insufficient water, whereas high nozzle pressure indicates excessive water.

Make-up Water. Make-up water must be added continuously to a cooling system while in operation. Two different losses occur: (1) the loss due to evaporation and (2) the loss due to drift or free-moisture removal.

Evaporative Loss. Evaporative loss cannot exceed 1 per cent for each 10° of cooling. This assumes full heat dissipation by evaporation, which never occurs as part of the heat is dissipated by convection. In practice, evaporation dissipates from 50 to 90 per cent of the total heat. The evaporative portion is greatest in hottest weather and with longest cooling ranges.

Drift Loss. Drift loss or carry-over of entrained free moisture of any one standard of design varies through wide ranges in atmospheric cooling systems, depending upon wind velocity, wind direction, and structural design. Well-designed spray ponds will show no loss at zero winds and 3 to 4 per cent at extreme winds. They will average less than $\frac{1}{2}$ of 1 per cent. Deck towers will come within these limits, but 5 to 8 per cent maximum drift loss in high winds is not uncommon with older designs. In mechanical-draft towers the drift loss remains fairly constant. Well-designed units will average about $\frac{1}{10}$ of 1 per cent loss, and when the loss exceeds $\frac{2}{10}$ of 1 per cent, drift becomes objectionable. Drift-loss comparisons are shown by Fig. 8-198.

Algae. Algae can be expected to grow in almost all basins, especially those which are exposed to sunlight. If algae that have formed on basin floors and walls become too thick, they will break loose to the surface and clog the water-distribution system. By using chlorine, copper sulfate, potassium permanganate, or some other suitable chemical, algae may be minimized or completely eliminated. A reliable water-treatment engineer should be consulted for the proper treatment.

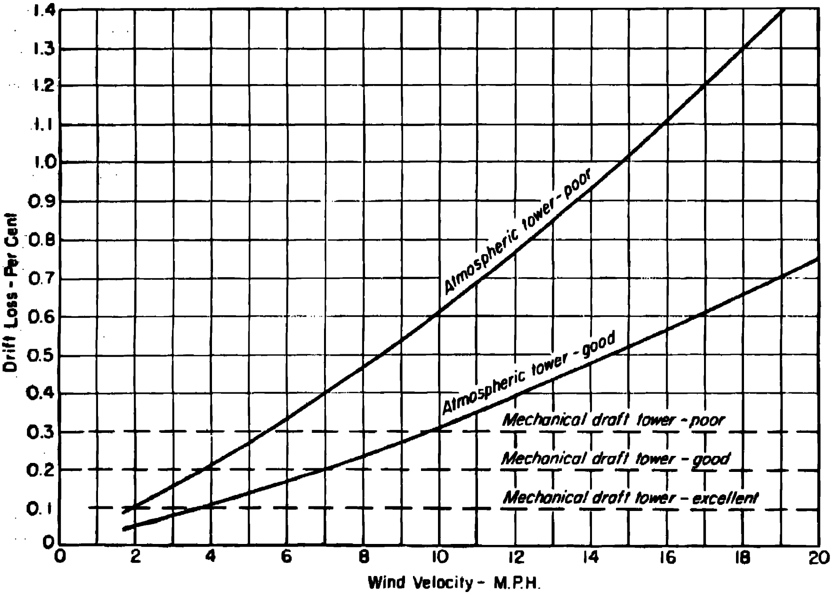
Scale. Scale-forming materials are found in practically all water. Scale will reduce heat-transfer rates, and scale forming in pipes and nozzles will reduce the effectiveness of water-cooling systems. A water-treatment engineer should be consulted to keep scaling at a minimum. Scale should be removed at regular intervals.

Concentration. As water is evaporated from the system the soluble solids remain in the water and increase in concentration. These solids will hamper the efficiency of the cooling system. Solution wet bulbs will be raised, and the system efficiency can be seriously reduced. Concentrations of solids can be controlled either by "blowing down" or by a "continuous overflow" to the sewer. A good water-treatment engineer can recommend the best way to control these solids.

Water Flow. The amount of water being circulated should be checked occasionally. Water flow can be checked by means of orifice meter, weir, pitot tube, pump characteristic curves, or spray-nozzle capacity curves. Reduced water quantities may indicate clogged or worn pumps, clogged nozzles, scaled-up tubing or piping.

When shutting down it should be made certain that all parts are properly drained and that the distribution system is protected from freezing and corrosion. While shut down, the entire system should be thoroughly inspected and repaired if necessary so it will be ready for immediate operation.

Structural Maintenance. When the system consists of a cooling tower, a careful maintenance and inspection schedule should be maintained to make certain that the structure is always in first-class condition. The following comments apply to both atmospheric and mechanical-draft units.



. 8-198. Drift-loss comparison of atmospheric and mechanical-draft towers.

Before starting all corrosive parts should be cleaned with special corrosive-resistant paint.

Any weak, broken, or warped lumber or structural members should be replaced.

All loose bolts should be tightened. Bolts in wood structures should not be pulled up too tightly as some allowance must be made for swelling of wood when wet.

Any loose, dry algae from hot-water basins or troughs and also from drift eliminators, deck filling, or air-inlet louvers should be removed.

All parts of hand rails, ladders, or stairways should be carefully examined or inspected.

Any painting of the exterior should be done while the structure is thoroughly dry. Painting of wood is not recommended as a preservative but for appearance only. Steel-cased towers should be painted annually.

Slat filling should be checked carefully, as any sagging of the deck slats or their support framing results in serious loss of cooling efficiency. Accurate spacing and position of slats is important.

Any possible structural failures of wood or hardware should be checked for, and wood should be examined for any signs of delignification.

If the system is in continuous operation, periodic inspections at intervals of 2 to 3 months apart are recommended.

Maintenance of Mechanical-draft Equipment. Where mechanical-draft equipment is used, a more rigid system is required, and attendants should inspect all moving machinery at least once per 8-hr shift.

Motors should be greased, painted, or otherwise maintained in accordance with the motor manufacturer's written instructions.

The alignment of motors, drive shafts, and gearboxes should be carefully checked at established intervals. Slight misalignment results in rapid failure of universal joints and is frequently the cause of excessive vibration. Universal joints should be oiled or greased as recommended, and drive shafts should be properly protected with guards to prevent damage to fans in case of failure.

Gearboxes should be drained and old oil replaced at intervals as recommended by the manufacturer. The instructions on the gear box should be followed. Too much end play of shafts indicates excessive bearing wear, and worn bearings should be replaced. Removable shims are usually provided to permit take-up of bearings if end play develops. Air vents should be kept open, and any water which may have accumulated in the gearbox should be drained off.

Table 8-62. Some Trouble Tips

Trouble	*Possible causes*
Unusual noise or vibration	Bad motor bearing
	Worn universal joint
	Loose motor or gearbox yoke
	Motor and gearbox not aligned
	Bad bearing in gearbox
	Loose fan hub
	Loose bolts in fan cylinder
	Loose bolts in tower structure
	Fan blades pitched unevenly
Excessive drift	Warped, broken, or missing eliminators
	Clogged or worn nozzles
	Excessive nozzle pressure
	Basin too full
	High winds
Increased temperature range	Decreased water circulation
	Dirt, algae, scale, or mud
	Valves partly closed
	Clogged strainers
	Worn pumps
Higher water temperatures	Reduced air flow—restrictions nearby
	Worn or dirty nozzles
	Extreme wet bulb conditions
	High concentration of solids
	Sagging of fill slats
	Algae on eliminators or decking
Rotting or disintegration of tower	Wood decay due to presence of fungi
	Insect attack
	Delignification
	Age
Poor vacuum	Mud or scaling in condenser
	Noncondensible vapors in condenser
	Reduced water quantity

Fans should be repainted as often as the need arises. The pitch of the fan blades must be uniform and should be carefully checked. A fan out of balance should be rebalanced at once. Fan cylinders should be kept painted if made of metal. All bolts should be tightened. The fan cylinder or ring requires frequent inspection and tightening. Clearance between fan ring and blade tips should be checked periodically.

In general, the greatest maintenance difficulties lie in the mechanical parts of the tower. Costly repairs and shutdowns can be prevented by a strict inspection and maintenance schedule. Spare mechanical parts should be carried in stock. All spray

ponds, atmospheric deck, or mechanical-draft towers require much the same operation and maintenance technique. However, individual instructions issued by the manufacturers for each type of water-cooling system should be carefully followed for the best results.

Winter Operation. Freezing weather demands extreme care and watchfulness to prevent serious damage. For atmospheric-deck-type towers a winter distributing header is recommended and is installed at a lower elevation. Ice accumulates on louvers and decks especially in the lower tower section and on the air-inlet side. Serious structural overloads occur, and resultant damage can easily occur.

Mechanical-draft towers have serious icing difficulties, but these can be eliminated or controlled by proper operation. Two-speed motors equipped with reversing switches should be standard equipment. Fully enclosed or splashproof motors are recommended. Splashproof motors equipped with space heaters provide good results.

Most forced-draft towers accumulate ice about the fans or fan rings due to moisture or spray. Serious damage to the mechanical equipment results if ice is not caught in time. Induced-draft towers cannot ice up the fans, and ice accumulation is confined to the inlet louvers or adjacent fill thus at least protecting the mechanical equipment.

It is practical to operate fans at one-half speed from fall to spring with a great saving in horsepower. During extreme freezing weather fans may be shut down continuously or intermittently as load conditions require. If ice still accumulates, fans may be reversed for short intervals, thus melting the ice.

If operating troubles develop, several sources should be investigated. Most probable of these are included in Table 8-62.

EVALUATING A WATER-COOLING TOWER

When cooling-tower manufacturers submit proposals to a buyer, the purchaser is confronted with various types of towers and various-priced towers. Each tower is said to give a certain performance. While almost all plant engineers know something of water-cooling towers, too often they have nothing too tangible to use as a guide in selecting the best cooling tower. The following material suggests some items that should be taken into consideration when purchasing a cooling tower.

Structural Theory. Structurally, a cooling tower may be considered as a shell or housing enclosing apparatus for mechanically forcing air through droplets of water for the purpose of cooling water. The housing must be capable of supporting

1. Internal loads of water passing through or being cooled in the cooling tower, the piping and tower filling (multiple strips used to mix mechanically the falling water with the air so as to facilitate heat transfer), the fans for forcing the air through the tower

2. External loads of wind, earthquake accelerations, and temperature effects; wind loads are usually assumed as (*a*) 20 psf of exposed vertical surface for cooling towers located on or near the ground level; (*b*) 30 psf for cooling towers located on roofs or other exposed locations. Earthquake loads are usually considered as horizontal forces (acting in any direction) equal to 0.1 of the vertical loads at or above the points or level of the structure under consideration.

The structural framework of a cooling tower consists of a series of framed bents (running both longitudinally and transversely in the tower). The horizontal ties or beams of these bents support the filling, water-spray or -distribution system, fan enclosure, and drift eliminators. The vertical posts act as supports for the beams and as struts in the wind or earthquake bracing system of the bent. The external casing and air-inlet louvers are usually attached to and supported by the exterior posts.

Any ordinary structural material that can be properly protected or is inherently resistant to decay when subjected to constant immersion in water (or alternate wetting and drying) can be used for cooling-tower construction. Redwood, cypress, Wolmanized fir, steel, and concrete are all used in varying degrees for that purpose.

Redwood[4] and cypress[5] (heartwoods) are the most commonly used materials because these timbers are impregnated with natural toxic ingredients which protect them from decay and insect attack. Since redwood is more generally used, its properties will be discussed briefly.

Redwood has a relatively high strength-weight ratio. It is highly resistant to warping and curling because of its low shrinkage and expansion with change of moisture content. Select heart redwood is usually used for the structural framework. Clear heartwood is used for exterior casing, louvers, trim, and other parts requiring good appearance and watertightness. Working stresses used in the design of cooling-tower structures for the grades of redwood outlined above are as shown in Table 8-63.

Table 8-63

Extreme fiber in bending	960 psi
Tension	960 psi
Compression with grain	800 psi
Compression across grain	250 psi
Horizontal shear	56 psi
Modulus of elasticity	1,200,000 psi

Since the cooling-tower frame is subject to severe exposure and possible vibratory loads, the connections or joints must be structurally adequate and designed for long life. These requirements dictate that bolted joints must be used throughout the framework with approved timber connectors[6] (such as the Teco ring type) used at all joints carrying horizontal loads. The nailing of the cooling-tower structure does not guarantee permanency and should not be used. Hardware should be of cast iron, bronze, or other corrosion-resistant material to ensure a well-balanced, long-lasting cooling tower.

Detailed Specifications for Mechanical-draft Towers. When requesting proposals from manufacturers be sure to include the following data:

1. Description of service intended
2. Location of installation or desired wet bulb
3. Total load in Btu or pounds of steam
4. Desired vacuum and style of condenser
5. Capacity in gallons per minute
6. Allowable hot- and cold-water temperatures
7. Characteristics of electrical current or power available
8. Cost of electricity or other power if any
9. Space limitations if any
10. Class of erection labor required
11. Desired delivery and erection schedule
12. Is location on ground or roof
13. Is railroad siding available
14. Must any local codes be adhered to such as fire or earthquake
15. Will electric power and water be available at site
16. Distance from site to nearest town
17. Specify whether bronze or galvanized bolts and nails
18. Specify who will furnish
    - *a.* Receiving and unloading
    - *b.* Erection supervision
    - *c.* Erection labor
    - *d.* Foundation and anchor bolts
    - *e.* Water piping and valves to tower
    - *f.* Electrical wiring and starters
    - *g.* Painting if any

Suggested General Specifications for Purpose of Purchase.

1. All lumber shall be of heart California redwood, free from sap and any serious defects. It shall be equal to No. 1 heart common or better as graded by the California Redwood Association.

2. The successful bidder will furnish a stress diagram. He will furnish detailed information of all joints and sizes of main frame members and bolts and also details of all load calculations.

3. Structural timber members shall be designed according to the basic unit stresses (reduced 30 per cent for continuously wet operating conditions) as established in the chapter on Structural Timbers, pages 99 *ff.* of the "Wood Handbook" of the Forest Products Laboratory, edition of 1935. Connections shall be designed to develop the stresses in the members with no allowance for nails or spikes. All dependence for the developing of the joints shall be upon the bolts or ring connectors. The strength of the bolted or ring connector joints shall be determined by the procedure and values established in the chapter on Timber Fastenings, pages 119 *ff.* of the aforementioned "Wood Handbook." The general design of the timber frame shall be according to the chapter on Wood Beams, Columns and Arches, pages 149 *ff.* in the above-mentioned "Wood Handbook."

4. Concrete members shall be designed according to the recommendations of the latest edition of Joint Code of Building Regulations for Reinforced Concrete Structures of the American Concrete Institute.

5. Cast iron used structurally shall conform to the standard specifications for gray iron castings, serial designation A-48-32-T, of the American Society for Testing Materials.

6. Cast steel used structurally shall conform to the standard specifications for steel castings, serial designation A-27-24, of the American Society for Testing Materials.

7. Unit stresses for combined dead and live loads and wind may be increased 33⅓ per cent over the provisions stated above for static loading conditions, provided that the section obtained thereby is not less than that required for dead and live loads along with the static unit stresses.

8. Cooling towers shall be designed to withstand a wind pressure of 30 psf, acting from any direction. Adequate anchorage shall be provided for uplift either during construction or in the completed tower.

9. If installed upon a roof, a ladder with guard will be provided. If installed upon the ground, a stairway will be provided. If in excess of 100 ft in length, two stairways will be provided, one to be at each end of the tower.

10. Deck-fill slats (flat side horizontal), if of ½-in. nominal thickness, shall be supported at intervals of not less than 18 in. If of 1-in. nominal thickness, they shall be supported at intervals of not less than 36 in. Allowable span for inlet louvers and eliminator slats may be double this amount where inclined at not less than 45 deg to the horizontal.

11. Motors shall be of the splashproof type unless otherwise required.

12. A speed reducer using spiral bevel gears is preferred. This shall be contained in a cast housing, oil-lubricated and with antifriction roller-type bearings.

13. All material and workmanship shall be guaranteed for a period of 1 year, and manufacturer will replace free of charge f.o.b. shipping point any defective part.

14. A performance acceptance test may be conducted at any time within 1 year after completion of erection. The manufacturer may have a representative present. Test will be conducted at purchaser's expense, and readings shall be taken in accordance with the ASME code for water-cooling equipment.

15. The purchaser may employ a competent licensed professional engineer at his expense to check structural drawings furnished, and successful bidder agrees to alter his structural design to meet the specifications of the purchaser if found necessary.

16. Casing shall be not less than 2 in. (nominal) in thickness. It may be of single- or double-wall construction.

17. Fill or deck slats of interlocking nailless design are to be preferred.

18. The cooling tower shall have a guaranteed drift loss of not exceeding 0.2 of 1 per cent under condition of 5-mph wind if of atmospheric design or under any and all conditions if of mechanical-draft design.

19. Hot-water-flow control valves shall be furnished for each section or cell of the tower.

20. The successful bidder shall furnish a printed manual with full instructions for care, operation, and maintenance of the tower.

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HYDRAULIC TURBINES

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The national importance of water power in the United States to supply the needs of industry, transportation, and domestic use is indicated in Fig. 8-199. It will be noted from this chart that, until 1895, the hydraulic-turbine industry was small and of little national importance. In that year, however, the first large water-power development at Niagara Falls was started; and since that time, the demand for water power has progressed at a rapid rate. From 1922 to 1931, installed horsepower increased at the rate of about 800,000 hp per year The slope of the curve rises to a still higher rate of about 870,000 hp per year from 1937 to 1945. Because of conditions growing out of the Second World War, only about 15,000 hp of new capacity

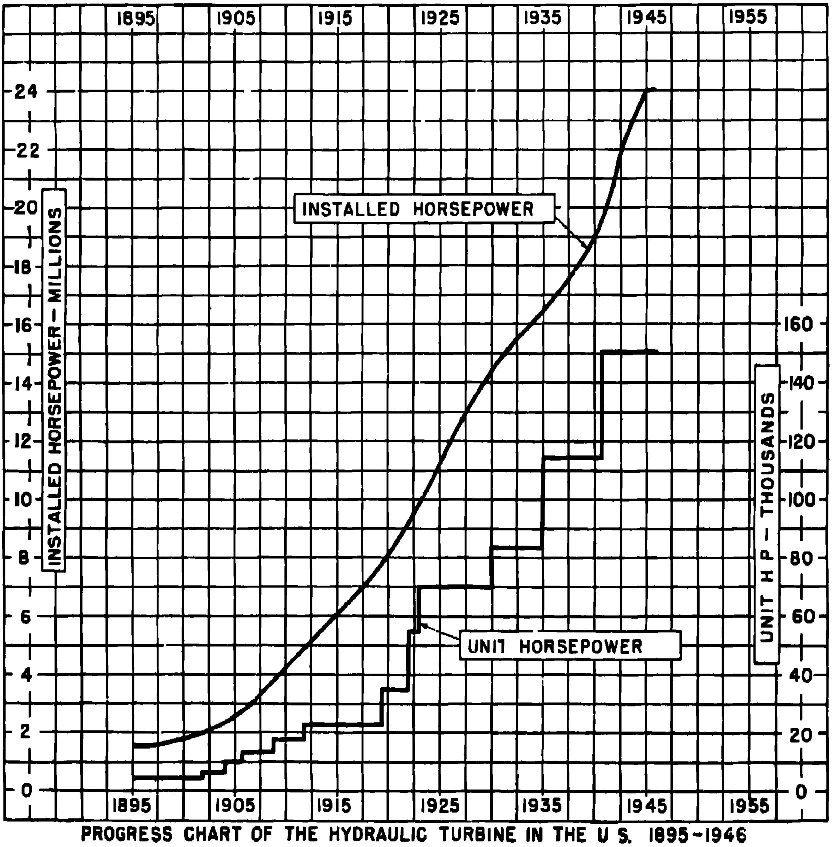

FIG. 8-199 Progress chart of the hydraulic turbine in the United States, 1895 to 1946. (*Federal Power Commission.*)

went into service in 1946; but from present data available about 500,000 hp went into service in 1947 and over a million horsepower is expected in 1948.

The increase in output from single units (unit horsepower) has been very great, starting in 1900 with the 5,000-hp units installed in the original Niagara Development and progressing to the 150,000-hp units first installed in the Grand Coulee Station on the Columbia River in Washington in 1941. These are the most powerful units in the world. Nine units are now in operation, and six additional units are under construction. In comparing hydroelectric power with steam-electric power, the important role played by the former is shown by the fact that, of the total power generated in central stations, about 35 per cent is furnished by hydroelectric plants as compared to 65 per cent from steam plants.

TYPES OF HYDRAULIC TURBINES

There are two general types of turbines, impulse and reaction. In the impulse type, all the energy contained in the water is converted to velocity head at the nozzle, where the buckets of the wheel receive the jet and convert the velocity in the jet to mechanical power by impulse action. In the reaction-type turbine, the energy is converted to power in the runner by means of vanes turning the water, thus causing a high pressure on the face of the vanes and a lower pressure on the back, causing a reaction. The reaction type is further subdivided into the following classifications:

1. *The Francis turbine* is of the mixed-flow type in that the water enters the runner in a radial direction and discharges the water in an axial direction. Such runners are usually constructed with from 12 to 19 vanes supported by the crown of the runner which is connected to the turbine shaft and guided by the band near the discharge side of the runner.

2. *The propeller turbine* also belongs to the reaction type but consists of from three to eight blades, the flow through the runner being in a general axial direction both at inflow and discharge. The runner blades are supported by the hub and have no band, thus in general resembling a ship propeller, from which it derives its name. Turbines of this type, having fixed blades, are usually referred to as propeller turbines.

3. *Kaplan turbines* are similar to fixed-blade propeller turbines, except that the blades are automatically adjusted during operation in order to develop high efficiencies over a wide range in loads.

Specific speed. As the head, horsepower, and speed may differ widely in different installations, a comparison of the various types of hydraulic turbines may best be made by comparing their specific speeds. Specific speed is the speed in revolutions per minute at which any turbine would operate at its best efficiency if the dimensions of such turbine are geometrically reduced so that the turbine would develop 1 hp under a net head of 1 ft. As a formula

$$N_s = \text{rpm} \times \sqrt{\text{hp}}/H^{5/4}$$

where N_s = specific speed
rpm = revolutions per minute
hp = horsepower developed
H = net effective head

The specific speed of the runner selected depends on the head. For high-head developments, low-specific-speed runners are used; and as the head decreases, the specific speed may be increased. An approximate formula to determine the specific speed for various heads is as follows:

For Francis-type turbines:

$$N_s = 660/\sqrt{H}$$

For propeller-type turbines:

$$N_s = 6{,}750/H$$

The size and shape of runners of different specific speeds differ materially as indicated by Fig. 8-200, which shows a comparison of five runners of different specific speeds. All these runners are drawn to the same scale, and each runner would develop the same horsepower output under the same net head but would operate at speeds proportional to the specific speeds indicated. The runner shown at the top, having a specific speed of 20, would be suitable for heads up to about 1,000 ft; the second runner, having a specific speed of 35, for heads up to about 300 ft; the third runner, having a specific speed of 53, for heads up to about 150 ft; the fourth runner, having a specific speed of 85, for heads up to about 60 ft; and the fifth runner, of the propeller type, having a specific speed of 143, for heads up to about 48 ft.

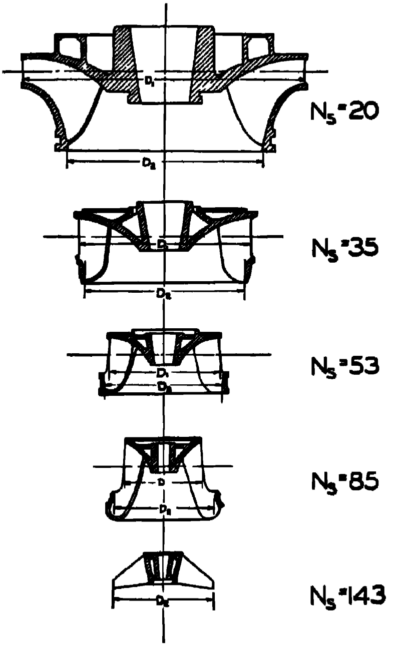

Fig. 8-200. Runners of equal power but with different specific speeds.

It is a characteristic of design that, as the specific speed of a runner increases, the horsepower-efficiency curve becomes steeper, *i.e.*, the part-load efficiencies decrease. This characteristic is indicated by Fig. 8-201, which shows typical horsepower-efficiency curves for the same runners indicated in Fig. 8-200. It may be noted from these curves that for high-head plants with runners of low specific speeds, flat horsepower-efficiency curves are obtained; whereas for the runners that would be used in low-head plants, where higher specific speeds are used, steep horsepower-efficiency curves result. Thus, in the case of the fixed-blade-propeller runner, the horsepower-efficiency curve is quite steep, and low efficiencies are obtained at part loads. In order to correct this inherent weakness in the characteristics of fixed-blade-propeller turbines, the Kaplan adjustable-blade turbine is used. This runner was named after Victor Kaplan of Austria, who first suggested it. It consists of runner blades supported in bearings in the runner hub. The blades can move while the turbine is in operation by means of an operating gear within the runner hub. This gear is connected by an internal shaft within the main turbine shaft, and the internal

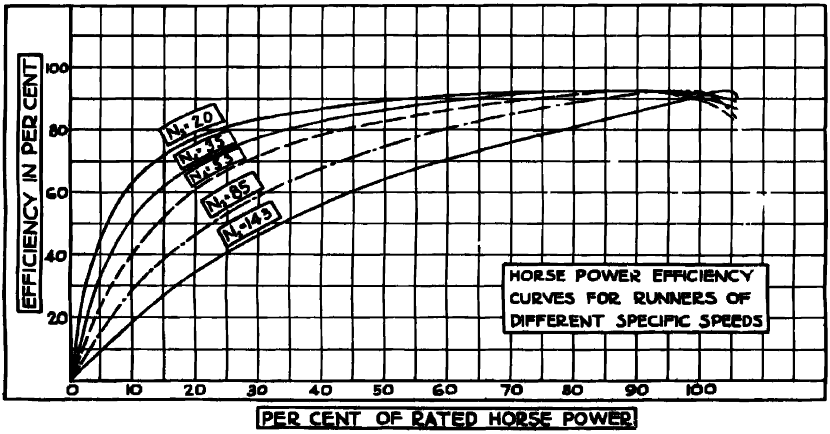

Fig. 8-201. Horsepower-efficiency curves for runners in Fig. 8-200.

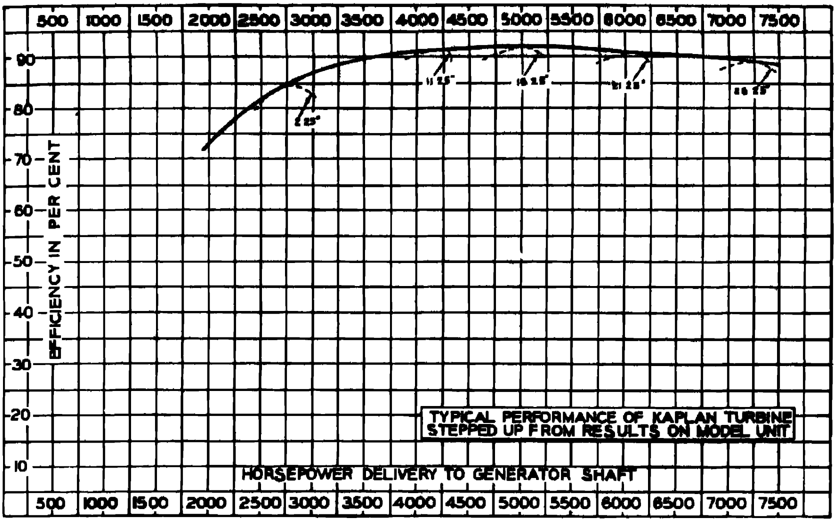

FIG. 8-202. Horsepower-efficiency curve for Kaplan-type turbine.

shaft is connected to a piston in a servomotor revolving with the main shaft. Oil pressure controlled by the governor is admitted to the revolving servomotor in order to adjust the runner blades. These runner blades and the turbine guide vanes are automatically and simultaneously adjusted in accordance with the load demand on the unit in order to provide the best possible hydraulic relation between the guide vane opening and the runner blade position. A typical horsepower-efficiency curve for the Kaplan-type turbine is shown in Fig. 8-202. Each of the dotted lines shows the performance of the unit if operated as a fixed-blade runner at the blade angle indicated. By automatically moving the runner blades at the same time as the guide vanes, the unit may be operated at the peak of each separate efficiency curve, so that the envelope curve (shown solid) becomes the actual operating curve of the unit. The Kaplan-type turbine, therefore, permits the adoption of high speed, yet maintains the high efficiencies over a wide range in load.

ACTUAL INSTALLATIONS OF DIFFERENT TYPES OF TURBINES

Impulse Type. Figure 8-203 shows a runner of a very large impulse turbine installed in the Big Creek No. 2-A Development of the Southern California Edison Company. The runner illustrated develops 35,000 hp under a net head of 2,200 ft at a speed of 250 rpm. A single unit consists of two overhung runners, each runner being provided with a single nozzle, so that the complete unit develops 70,000 hp.

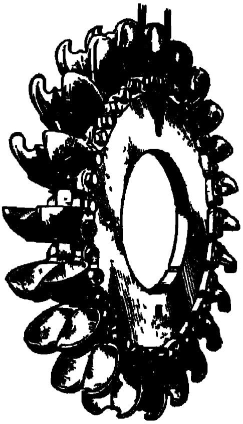

FIG. 8-203. Impulse runner, Big Creek 2-A Development of the Southern California Edison Company.

High-head Francis Type. Figure 8-204 is a shop view of an assembled turbine built for the Hoover Dam Plant of the U.S. Bureau of Reclamation, located on the Colorado River between Arizona and Nevada. Each unit is designed to develop 115,000 hp under a net head of 480 ft at a speed of 180 rpm. Twelve units of this size are now installed in the Hoover plant, together with one smaller unit of 55,000

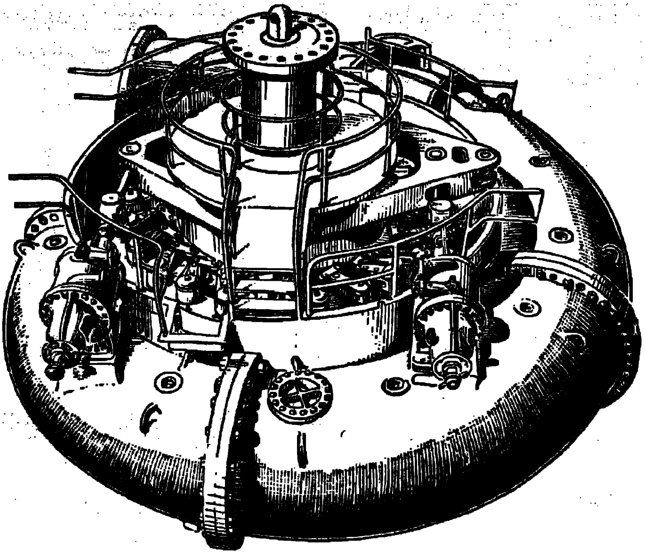

FIG. 8-204. Hoover Dam turbine, 115,000 hp, in shop.

hp. There is still space in this plant for installing three additional main turbines and one additional unit of the smaller size.

Medium-head Francis Type. An example of the Francis-type turbine operating under a medium head is given in Fig. 8-205, a shop view showing runner and shaft being lowered into the unit. This turbine was built for the Loup River Public Power District, Columbus Power Plant, Nebraska. Three units are now in operation, each designed to develop 18,000 hp under a net head of 112 ft at a speed of 150 rpm. The illustration shows only the assembled guide vanes and head cover, but this unit is provided with a plate-steel volute-type casing.

Low-head Propeller Type. As an example of a low-head development, Fig. 8-206 shows the runner of a fixed-blade

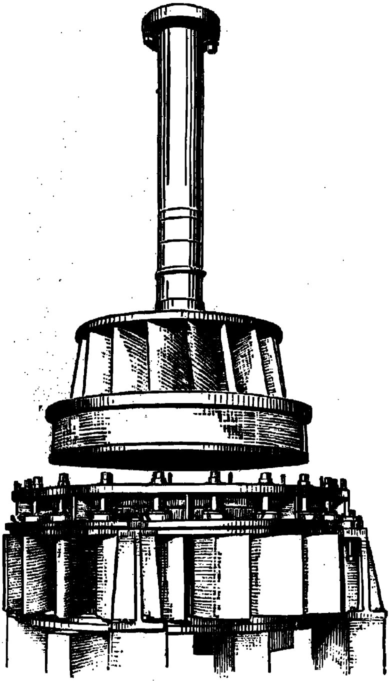

FIG. 8-205. Turbine of 18,000 hp capacity, Loup River Public Power District.

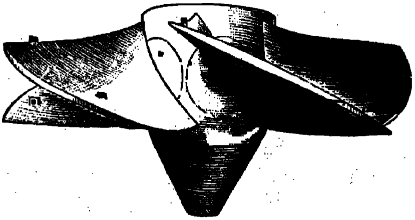

FIG. 8-206. Propeller runner, 45,000 hp, fixed-blade type, Wheeler Dam Development of Tennessee Valley Authority.

propeller-type turbine built for the Tennessee Valley Authority, Wheeler Dam Plant, on the Tennessee River in Alabama. Six units are now installed in this plant, and two additional units are under construction. Each unit is rated at 45,000 hp under a head of 48 ft at a speed of 85.7 rpm. The runner illustrated has an outside diameter of 22 ft.

As an example of a Kaplan-type turbine for low-head operation, Fig. 8-207 is a sectional view through one of the turbines built for the Tennessee Valley Authority, Watts Bar Dam, on the Tennessee River in Tennessee. Five of these turbines are now in operation, each unit developing 42,000 hp under a net head of 52 ft at a speed of 94.7 rpm.

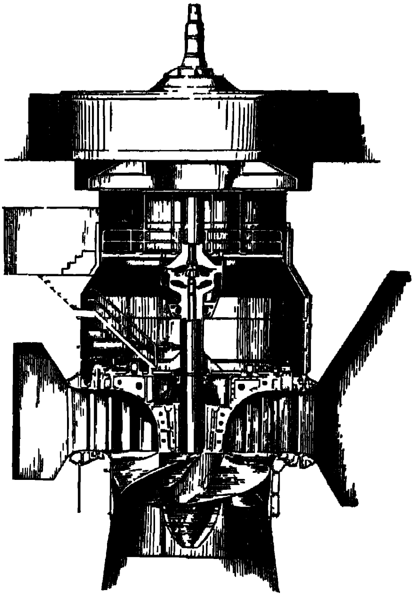

Fig. 8-207. Section of 42,000-hp Kaplan-type turbine, Watts Bar Development of Tennessee Valley Authority.

LABORATORY TESTING

Research work is carried on by all the turbine manufacturers in specially designed hydraulic laboratories in order to be able to predict the performance of the large units in the field. Models exactly homologous to the large units, including the intake, casing, and draft tube, are tested over a wide range in power and speed. The model runners are usually from 10 to 16 in. in diameter. During the testing program, changes can readily be made in the runners, guide vanes, casing, and draft tube in order to obtain the best possible combination before starting construction of the large units. From the test data secured in the laboratory, the performance of the large units in the field can be accurately predicted. From the tests made on the models, curves may be plotted between the speed factor phi and unit horsepower (the power the runner would develop under a 1-ft head and a 1-ft runner diameter) and between phi and efficiency for various gate openings. The speed factor phi is the ratio of the peripheral speed of the runner to the spouting velocity of the water under the full head, or expressed as a formula

$$\phi = (D_2 \times \text{rpm})/(1{,}836 \times \sqrt{H})$$

D_2 represents the diameter at the throat of the runner in inches, and H is the net effective head on the unit

The model results can be stepped up to the field conditions by the following formulas at the phi selected for the field conditions:

$$\text{hp}_2 = \text{hp}_1 \times (D_2/D_1)^2 \times (H_2/H_1)^{3/2}$$

where hp_2 = expected horsepower from prototype or large unit
hp_1 = horsepower developed by model
D_2 = runner throat diameter for prototype unit
D_1 = throat diameter of model
H_2 = net effective head on prototype unit
H_1 = net effective head on model

The efficiencies of the prototype will be higher than the model efficiencies because of the smaller internal losses of the water flowing through the larger passages. The

Moody formula, which is generally accepted. may be used to determine expected field efficiencies as follows:

$$E_p = 100 - (100 - E_m) \times (D_m/D_p)^{0.20}$$

where E_p = expected efficiency of prototype
E_m = maximum efficiency obtained on model at its best phi
D_m = throat diameter of model
D_p = throat diameter of prototype

The gain in efficiency for the prototype unit, or $E_p - E_m$, is then applied to the efficiency of the model at the phi selected for the prototype unit. The performance of the unit in the field for various heads can be predicted in the same manner by using the phis corresponding to the heads considered.

CAVITATION

Cavitation research is of vital importance both to the turbine industry and to the operating companies, and specially equipped laboratories are used for these studies. Cavitation may be defined as the formation of vapor-filled cavities at points on the runner blades or at any point within the water stream which disturbs the continuity of flow or causes the water to leave the blade surfaces. Cavitation reduces both turbine output and efficiency, and because of the collapse or "implosion" of the cavities where the flow reaches points of higher pressure, it causes pitting of the runner or adjacent parts.

At a given elevation above sea level and a given water temperature, the absolute pressure available is the water barometric pressure. At the runner discharge, this pressure is reduced by

1. The elevation of the runner above tail water
2. The velocity head
3. The curvature of the flow lines

This third element is dependent upon the runner design, particularly on the proportional magnitude of the blade area and the blade spacing, and may differ materially on two runners of the same specific speed. Hence, for any given runner, it is necessary to determine the correct elevation with respect to tail water so as to assure sufficient absolute pressure at the discharge of the runner blades under the required head to avoid cavitation. In the special cavitation laboratories, the elevation of the tail water can be varied while the total head on the model turbine is maintained constant. As the tail water is lowered, observations are made of the discharge, power output, and efficiency; and the break in these curves indicates the cavitation limit. The cavitation coefficient is known as sigma and is given by the formula

$$\sigma = (H_b - H_s)/H$$

where H_b = height of water barometric column at proper elevation above sea level and normal water temperature
H_s = greatest elevation of runner above tail water at which cavitation does not occur with turbine operating at its rated capacity
H = total effective head on unit

It is to be noted that, if cavitation begins while the runner is still below tail water, H_s is negative and represents the least depth of runner below tail water at which cavitation does not occur.

Values of sigma must be determined over the entire range of power and head through which the runner is to be operated. In fixing the elevation of the runner in the field, a factor of safety of from 3 to 4 ft should be allowed, *i.e.*, the value of sigma used in the field should be well above the actual value determined in the laboratory.

From studies made of the behavior of various installations in operation, with particular reference to the avoidance of pitting, the curves shown in Fig. 8-208 may be used as a general guide to the selection of plant sigma values. This curve is presented as an approximate method of determining the elevation of the runner for a proposed installation; but the final elevation should be fixed from the results of cavitation tests made on the model of the runner to be used in the actual installation. As an example for the use of this curve, if we assume a hydroelectric installation where the elevation above sea level is 1,000 ft and the net effective head on the turbine is 300 ft, we can arrive at the approximate elevation of the runner above the low tail-water level in the following manner.

From the approximate formula for determining specific speed as shown on page 579, it will be noted that, for a head of 300 ft, the specific speed should be about 38. For this specific speed, from curve shown in Fig. 8-208, it is noted that the plant sigma should be 0.09. For this elevation above sea level and assuming a water temperature of 70°F, the value of the water barometer H_b is 31.6; and from the formula for sigma shown on page 584, the distance of the runner above tail water H_s is obtained as follows:

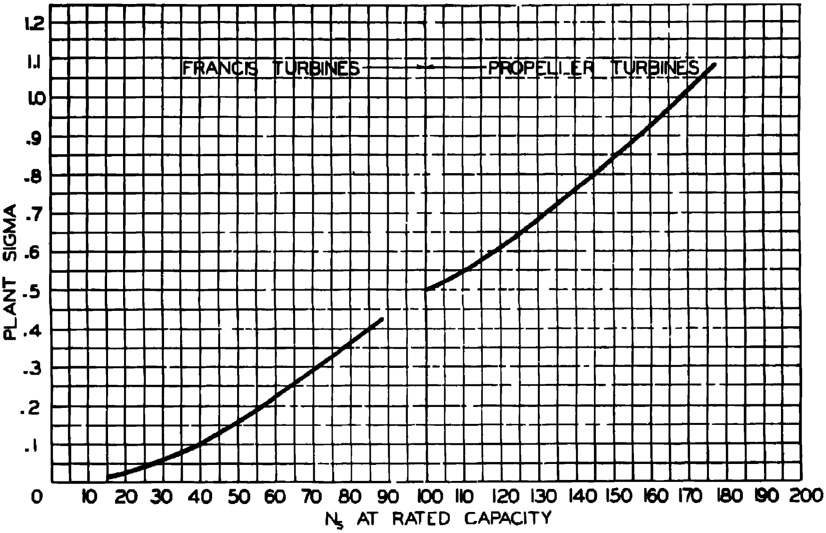

Fig. 8-208. Plant sigma vs. specific speed.

$$H_s = 31.6 - 0.09 \times 300 = 4.6 \text{ ft}$$

In a similar manner, if we assume the installation of a Kaplan-type turbine where the total net head is 48 ft and the plant is located 1,000 ft above sea level, a specific speed of 141 could be used; and using the same formulas, it is found that the runner should be located about five feet below low tail-water elevation.

FIELD TESTING

It is desirable, when possible, to conduct performance tests of turbines at site, as the data thus obtained permits the most economical operation of the units for the various flows and heads to be encountered. Field tests are usually conducted in accordance with the Test Code for Hydraulic Prime Movers of the ASME, originally issued in June, 1938. The present treatise will, therefore, only briefly describe the values to be measured in order to determine turbine efficiency.

The turbine efficiency is the ratio of the brake horsepower delivered by the turbine to the water horsepower available, or

$$E = \text{bhp}/\text{whp}$$

Water horsepower is given by the formula

$$\text{whp} = (Q \times w \times H)/550$$

where Q = flow, cfs
w = weight of water, lb per cu ft
H = effective head, ft

In the case of hydroelectric units, the brake horsepower is determined from the generator output divided by the generator efficiency. The generator efficiency may be determined by tests made in the shops of the builder or after installation, either by direct measurement of input and output or by the separate loss method.

The effective head on the turbine is the difference between the total energy contained in the water immediately before its entrance into the turbine and the total energy in the water immediately after its discharge from the draft tube. As the total energy at any point is the pressure head plus the velocity head, the net effective head on the turbine is given by the following:

$$H = H_p + (V_1^2/2g) + D - (V_2^2/2g)$$

where H_p = pressure head measured at casing intake, ft
V_1 = velocity at point where pressure head is measured, fps
D = distance from point of measurement of pressure head to elevation of the water in the tailrace near the draft-tube discharge at points reasonably free from local disturbances, ft
V_2 = velocity at the section of measurement in the tailrace, fps

The pressure head H_p is usually obtained from calibrated pressure gauges or direct-reading U tubes or elevation tubes. The velocity V_1 is determined by dividing the total flow in cubic feet per second by the area in square feet at the point of measurement, and the velocity V_2 is determined by dividing the total quantity in cubic feet per second by the area of that part of the tailrace channel which may fairly be charged to the unit under test.

The measurement of flow in cubic feet per second is usually considered the most difficult of all factors to be determined. Many different methods of quantity determination have been used, depending on local conditions, such as the Venturi meter, weir, pitot tube, current meters, salt velocity, and Gibson methods. The last two methods have been in general use for a number of years and are considered the most accurate for determining the flow of water. Therefore, a brief description will be given herein in connection with each of these two methods.

Salt-velocity Method. The salt-velocity or Allen method for measuring the flow of water was developed by Professor Charles M. Allen of Worcester Polytechnic Institute, Massachusetts. This method has been very widely adopted in a large number of tests made in this country and many foreign countries. The method is based on the fact that salt in solution increases the electrical conductivity of water. The salt solution is introduced near the upper end of the conduit, and the passage of the solution across one or more pairs of electrodes at other points in the conduit is recorded graphically by electrical instruments. The passage of the salt solution between two points is accurately timed, and the volume of the penstock between the same points is accurately measured. The discharge in cubic feet per second is then equal to the volume of the penstock in cubic feet divided by the time in seconds.

In making such a test, it is necessary to provide for the introduction of the salt solution and for two sets of electrodes and a source of electrical energy, generally 110 volts direct current or alternating current. A recording ammeter or wattmeter is connected in the circuit of the electrodes; and before the salt solution is introduced, the initial steady reading of the meter will be from ½ to 1 amp. When the salt solution reaches the first set of electrodes, the current passing between the bars increases (owing to the greater conductivity of the water), reaches a maximum value, and decreases again to normal as the charge of salt passes by, thus producing a humped curve on the chart. A similar curve is drawn by the meter when the salt solution passes the second set of electrodes. The time is also recorded on the same chart by a clock or seconds pendulum connected electrically to a pen or by a hot spark. The time between the centers of gravity of the two curves produced is the average time it has taken the water to travel the distance between the two sets of electrodes. The volume in cubic feet between the two sets of electrodes is accurately determined by physical measurements, and the quantity flowing in cubic feet per second is the volume divided by the time.

Gibson Method. The Gibson method of water-flow measurement was developed some years ago by Norman R. Gibson, Consulting Engineer for the Niagara Falls Power Company. This method makes use of two well-known principles, the first of which is Newton's second law of motion, that the force is equal to the product of the mass and the rate of change of velocity of that mass, or

$$F = M(dv/dt)$$

This equation may also be written $F\,dt = M\,dv$, which expresses the fact that the impulse or product of force by time is equal to the change in momentum or the product of the mass by the velocity change.

The second principle is the corollary of the first, *viz.*, the relation between the change of pressure and the change of velocity of a column of water expressed in terms of velocity of the pressure waves which are propagated during the change from one end of the column to the other.

To apply this principle to the determination of water quantity flowing in a pipe of uniform cross section, the gates of the turbine at the lower end are closed gradually in a given time, so that the velocity of the water in the pipe is diminished and is finally stopped when the turbine gates are fully closed. During this closure, the pressure in the pipe rises, and this pressure rise is the manifestation of the force exerted to stop the flow of the mass of water in the pipe. From the fundamental impulse equation given above, if the mass of water in the pipe is known and the time taken to stop its flow and the average force exerted in doing so are measured, the velocity of the water prior to the interruption of its flow may readily be determined.

Thus

$$PwaT = (Law/g) \times V$$

from which

$$V = gPT/L$$

where a = area of pipe, sq ft
L = length of pipe, ft
V = velocity of water, fps
T = time to stop flow, sec
P = average pressure rise during time T, ft
w = weight of water, lb per cu ft
g = acceleration due to gravity

The average pressure rise P and the time T are recorded by specially designed Gibson apparatus so that it is necessary only to measure the length L of the penstock to determine the velocity V.

In practice, the test procedure consists of placing the desired amount of load on the unit and, for a period of time, observing the head water elevation, tail-water elevation, casing pressure, speed, gate opening, and the various electrical measurements necessary to obtain the generator output. At a given signal, the turbine gates are closed at a slow rate, and the load is taken up by another unit operating in synchronism with it. The rate of closure need not be uniform, although this operation should take place as smoothly as possible. During the closure, the flow in the penstock is being retarded and produces a change in pressure. A continuous record of this pressure change and the time of closure is made by the Gibson apparatus.

In most installations, the sections of the penstock are not of uniform area, but the penstock is made up of a series of sections of different lengths and different areas. The formula is, therefore, usually expressed in terms of discharge, the net area of the pressure-time diagram, the constants of the apparatus, and the physical dimensions of the penstock, by the following formula:

$$Q = K_g A_d / SF$$

where Q = discharge, cfs
K_g = calibration constant of recording apparatus = g/y

y = vertical height, in., on diagram corresponding to 1 ft of pressure change in the penstock

A_d = net area of pressure-time diagram, sq in.

S = time constant of diagram = horizontal length, in., corresponding to 1 sec of time

F = constant of the penstock. If pipe is of uniform cross section, $F = L/a$. If penstock is of nonuniform cross section, F is the integration of length divided by area

Because of the fact that some leakage always occurs through the turbine guide vanes when in their closed position, this leakage quantity is not cut off by the closure of the gates. The total turbine discharge obtained by the above formula must therefore be corrected by adding the amount of such leakage.

OPERATION OF UNITS IN THE FIELD

Field tests as previously described are of very considerable value to the operating forces at the plant, as from such test data the units can be operated as near their points of maximum efficiency as permitted by the loading. Curves of power discharge may be plotted so that the turbine itself may be used as a measuring device or indicating flowmeter. It is also possible, when making the field tests, to calibrate a flowmeter consisting of a closed U tube, each arm of which is connected by piping to a piezometer in the water passage, where the velocities differ by a reasonable amount. Readings are taken during the efficiency tests on the turbine, and curves may be plotted between the total deflections of the U tube and the discharges determined during the test. Such a calibration curve may be used in station operation for determining quite accurately the flow under any load condition and whether the efficiencies are being properly maintained. A drop in efficiency might be due to wear at the seal rings between the runner and stationary parts or possibly to foreign material wedged in the runner vanes to such an extent as to decrease the flow at a given gate opening.

The data obtained in field tests are also of value in setting the gate limits on the turbine to prevent overloading If turbines are operated appreciably beyond their rated horsepower, there is a drop in efficiency and an increase in discharge beyond the full rated flow. This practice results not only in loss of efficiency but also in pitting on the turbine runners and adjacent parts due possibly to exceeding the cavitation limits.

INSPECTION AND LUBRICATION

Regular inspections of the visible parts of the turbine are usually made once each day or preferably once each shift. Such inspections cover shaft swing, bearing feed, stuffing box, and operating gear.

If the shaft swing increases appreciably, it may be due to bearing wear from lack of proper lubrication or to unbalance in the generator or turbine. Oil-type bearings are usually supplied by two motor-driven pumps, the regular pump being driven by an a-c motor and the emergency pump by a d-c motor from the storage battery. In case of failure of the a-c motor, the d-c motor pump should automatically start up. These controls should be carefully checked at each regular inspection.

In many installations of vertical-shaft turbines, water-lubricated bearings are used. Such bearings consist of strips of lignum vitae or compressed phenolic materials driven into slots in the bearing housing and then bored or ground to the finished diameter to give proper clearance on the shaft. Where the available water supply is reasonably free of foreign materials, such water-lubricated bearings are entirely satisfactory and require less auxiliary equipment than the oil-type bearings. The main stuffing box may be located above the bearing and is, therefore, more accessible for inspection and adjustment than the stuffing box with the oil-type bearing, which must be located below the bearing Furthermore, for water-type bearings, no oil pumps or sump pumps are required One very important consideration, however, for an installation with water-lubricated bearings is an assured water supply at all times. The water may be taken from the ordinary station system if reasonably clear. Otherwise, filtered water should be used. A flowmeter is installed in the feed line with a low flow alarm connected electrically to the station switchboard so as to give the operator immediate notice of water supply below normal. In such an event, the unit should be shut down, either manually or preferably automatically, to prevent damage to the bearing.

Excessive shaft swing may also be due to unbalance in the turbine runner. The runner is usually balanced in the manufacturer's shops before shipment, so when first installed it should operate smoothly. Unbalance at a later date may be due to foreign material becoming wedged between the vanes.

In the main stuffing box, the packing should be adjusted to allow a small leakage at all times to prevent the packing from cutting the shaft sleeve.

The operating gear, consisting of all pins, levers, and links connecting the operating ring to the servomotors, should be free of lost motion due to wear; otherwise the speed regulation will be impaired. If such wear is discovered, the pins and bushings should be repaired or renewed.

Lubrication. The guide-vane bushings and all pin connections between the levers, links, and operating ring are lubricated by a high-pressure grease system. This usually consists of a high-pressure grease gun, operated by air pressure from the station system and connected through reinforced flexible hose to a hand-operated measuring valve and trigger control. Connection is made by the operator to special fittings at each bearing or sliding contact, and a measured amount of grease is forced to each point requiring lubrication. In some plants, automatic lubrication is provided by piping the grease gun to all points and automatically greasing at fixed time intervals.

It is recommended that all parts of the operating gear be lubricated at least once a day and preferably once every shift.

For Kaplan-type turbines, the runner hubs are filled with oil to lubricate the blade-shank bearings. The oil is carried up through a central hole in the turbine shaft to a revolving oil chamber so as to maintain sufficient oil pressure in the hub to prevent leakage of water into the hub. The oil level at the revolving oil chamber should be checked at regular intervals. A falling of this oil level shows loss of oil from the hub, which should be replaced. If such loss is excessive, the unit should be shut down, and the packing at the runner blade shanks adjusted or replaced.

A rise in the oil level at the revolving chamber indicates that water is leaking into the runner hub and forcing the oil up through the shaft hole. Such a mixture of oil and water in the hub would cause damage to the blade-shank bearings, and the water should be drained or forced out of the hub by adding more oil at the filling valve in the shaft.

Inspection of Internal Parts. The first inspection of the internal parts of the turbine should be made within 6 months after starting the machine in initial operation, and thereafter complete inspection of internal parts should be made at least once a year.

The clearance between the runner seals and stationary seals should be checked with feelers. This is of particular importance in connection with high-head turbines, as excess clearance at the runner seals would result in considerable loss in efficiency. Such wear is of less importance in connection with low-head turbines.

Of particular importance in connection with these inspections is the pitting which may occur in connection with the runner and other parts. In the case of Francis-type turbines, such pitting will usually be found just beyond the discharge orifice of the runner vanes, or what is usually known as the "spoon" of the vanes. This is due to the fact that, when the water has left the discharge orifice, it is no longer guided, and the water may leave the back of the vanes, causing points of high local vacuum, resulting in pitting. In connection with Francis-type turbines, it is also found that pitting will sometimes occur on the back of the vanes near the intake edge and extend onto the runner band. This is due very often to the rapid curvature of the runner band near the entrance edge of the vanes.

In the case of propeller-type turbines, pitting is apt to occur on the back of the blades near the outer periphery and toward the trailing edges of the blades. Pitting also occurs in this type of turbine near the runner hub where the blades must be of considerable thickness to transmit the load to the hub.

Pitting also sometimes occurs on the stationary parts near the runner discharge where the velocity is extremely high. Pitting or mechanical wear is also sometimes found on the distributor plates immediately above and below the guide vanes and also on the guide vanes themselves. Such wear is more apt to occur if units are operating at small loads or as synchronous condensers with the casing full of water.

If pitting or wear of the internal parts is severe, repairs should be made by chipping the pitted areas to clean metal and welding with stainless steel. It has been found from experience that stainless steel will resist pitting for a considerably longer period than ordinary cast steel or plate steel. It is for this reason that in many turbines the parts of the runner vanes or blades most likely to pit are prewelded in the shop with stainless steel.

When inspections are made of the internal parts, it is also advisable to check any possible wear in the guide-vane bushings by means of feelers. Such wear is due to the thrust of the water against the guide vanes in the direction toward the center of the unit, or possibly may be due to lack of proper lubrication of these parts.

At the time of these yearly inspections, a complete record should be kept of the wear at the various points and particularly of any pitting which is occurring on the runner vanes or other parts. It is advisable to number the runner vanes and to make sketches showing the location of the pitting and its depth on each vane, so that, in future inspections, comparisons may be made of the increase in pitting or wear each year.

MECHANICAL-DRIVE TURBINES

BY H. D. BEVAN

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PRINCIPLE

The mechanical-drive turbine operates on the same basic principle as the turbine of a large turbogenerator unit. It converts heat energy into mechanical energy. In its simplest form it consists of a single wheel with blades or buckets mounted on the periphery. Steam is expanded through accurately proportioned nozzles which convert the energy in the steam into velocity energy. The high-velocity steam then strikes the blades and causes the wheel to rotate. Figure 8-209 shows the action of the steam in two types of single-stage mechanical-drive turbines. These units, and practically all mechanical-drive turbines, are of the impulse type.

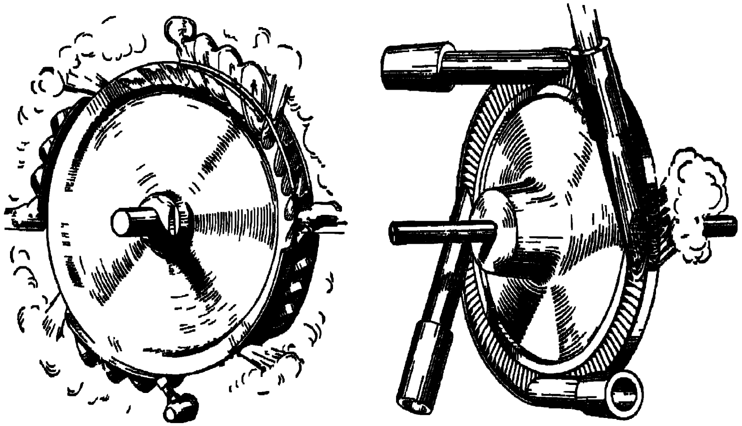


FIG. 8-209

DETAILED DESIGN

Turbine designs are flexible, and more or less standard machines are built for a wide variety of drives and steam conditions. The design and the cost of a mechanical-drive turbine depend upon the conditions under which it will be used. To submit figures on a turbine, the manufacturer must know:

1. Horsepower required and type of driven machine.
2. Steam pressure which will be available at the turbine inlet and condition of steam at inlet, viz., whether it is dry and saturated, moist and per cent quality, or superheated and deg F.
3. Pressure at turbine exhaust.
4. Speed at which the driven machine will operate and type of governor or control mechanism that will be required.

If occasional low initial pressure, high back pressure, or overload is likely to be encountered, these should be stated so the manufacturer will supply sufficient nozzle capacity. All these could be taken care of by one or more extra hand-valve-controlled nozzles which would permit the turbine to be designed for best efficiency at rated conditions.

If new boilers with higher pressure or temperature are likely to be installed later, consideration should be given to designing the turbine so it can be operated with the new conditions. Under some circumstances it is more practical to install new parts in the turbine when the future changes take place. The manufacturer can make recommendations if he knows the requirements.

5. Any other details or special requirements such as operation as a bleeder turbine, as a mixed-pressure turbine, maximum total steam consumption desired, etc.

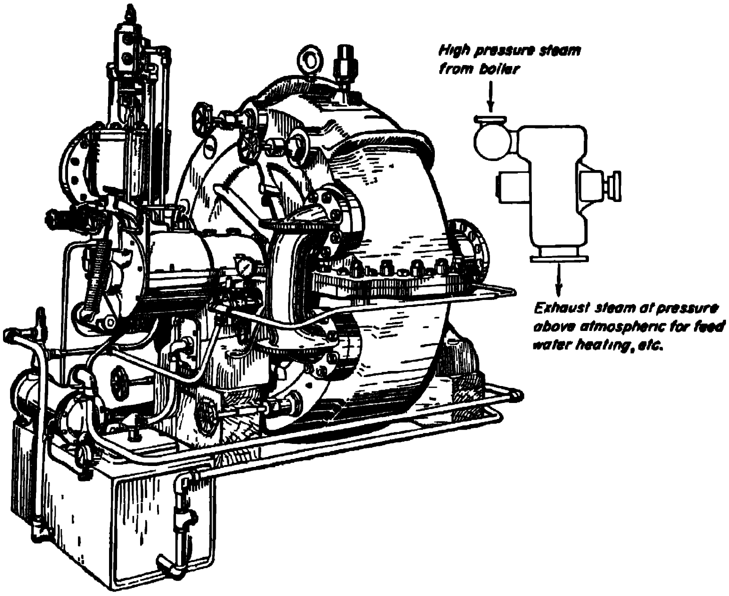

High pressure steam from boiler
Exhaust steam at pressure above atmospheric for feed water heating, etc.

Fig. 8-210. Single-stage, back-pressure turbine with oil relay governor, forced-feed lubrication, and hand-valve-controlled nozzles

TYPES OF MECHANICAL-DRIVE TURBINES

Mechanical-drive turbines are used extensively in the boiler plants of all industries, in process plants, chemical plants, oil refineries, paper mills, breweries, etc. They commonly drive equipment such as

Centrifugal pumps—boiler feed, fuel oil, condenser circulating, hot well, fire, general service, etc.

Fans and blowers—forced and induced draft, gas, etc.

Electric generators—auxiliary exciters, emergency electric-generator units, etc.

Miscellaneous apparatus paper machines, coal pulverizers, line shafting, centrifugal compressors and exhausters, centrifugal refrigeration machines, etc.

Back-pressure Turbine. Most mechanical-drive turbines are of the back-pressure or reducing-valve type. They are used where there is a demand for low-pressure steam for feed-water heating, general heating, or for some commercial process.

In installations of this kind the turbine acts as a power-generating reducing valve It reduces the pressure of the steam from that in the boiler to that required for heating or process At the same time it skims off power as a by-product but in so doing extracts only a small percentage (about 10 per cent) of the total heat in the steam. The value of the steam as a heating medium is, therefore, not impaired, and it is not contaminated by oil. The steam charge for this by-product power is approximately 10 per cent of the steam passing through the turbine. This results in extremely high over-all thermal efficiency.

The smaller drives of this type are single-stage turbines (Fig. 8-210). If the heat balance of the plant requires a lower steam rate than is obtainable with a single-stage machine, the more efficient multistage turbine with two or more sets of expanding nozzles and corresponding wheels may be used for some of the larger drives. Figure 8-211 shows the steam path through a machine of this type.

Condensing Turbine. In the condensing turbine steam is expanded through the turbine to a condenser (Fig. 8-212). While the steam rate is lower, a large part of the heat in the steam is given up to the condenser, and heat returned to the boiler in the condensate is a relatively small part of the heat content of the exhaust steam. Figure 8-213 shows the relative heat recovery from a condensing turbine compared to that from a noncondensing or back-pressure turbine.

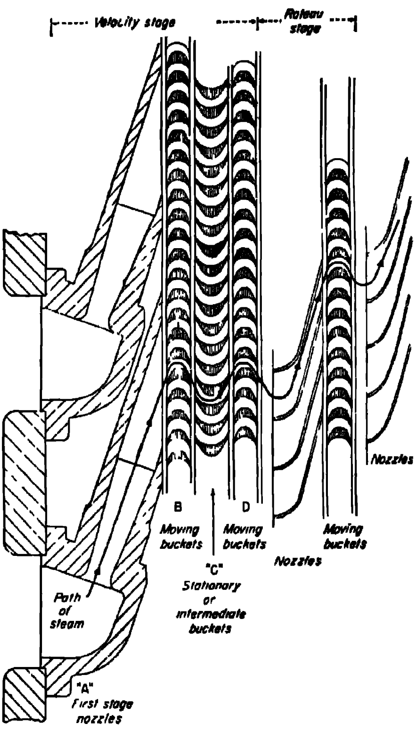


Fig. 8-211. Steam path through a multistage turbine.

Since the steam is expanded to a lower pressure in a condensing machine, there is more heat available for making power than with a back-pressure turbine. To utilize this energy efficiently it is generally necessary to use a multistage turbine. Such a turbine is considerably more expensive than a single-stage unit, and there is an added investment for a condenser, pumping equipment, etc., which is difficult to justify for small installations.

For installations where it is desirable to operate condensing at low first cost, single-stage condensing turbines may be installed. Care must be used in selecting a single-stage turbine for condensing operation, however, since the high heat drop and consequent high steam velocity through the turbine may cause rapid blade wear.

Condensing turbines are sometimes installed to operate in conjunction with back-pressure turbines. The back-pressure machine is operated when steam for heating is needed, and the condensing turbine is used when no exhaust steam is required.

Bleeder Turbine and Mixed-pressure Turbine. The bleeder turbine combines the advantage of the back-pressure turbine with that of the condensing turbine (Fig. 8-211). High-pressure steam is admitted to the turbine and, after being expanded to the required pressure through one or more stages and making power, is extracted from the turbine casing for process or heating. If the quantity of steam extracted is not sufficient to develop the full horsepower required, additional steam is expanded through the low-pressure stages of the turbine to a condenser, and this portion of the power is made with a low water rate. Such a machine will take care of plants where the power and process steam loads do not vary up and down together.

The bleeder turbine is fitted with a valve gear to maintain the required bleeder pressure automatically.

In some installations, instead of having a condensing low-pressure end, the bleeder turbine is built to exhaust at a back pressure above atmospheric but below bleeder pressure (Fig. 8-215). This type of turbine could be used in a plant requiring process steam at 75 lb gauge pressure, for example, but with an additional heating load at 5

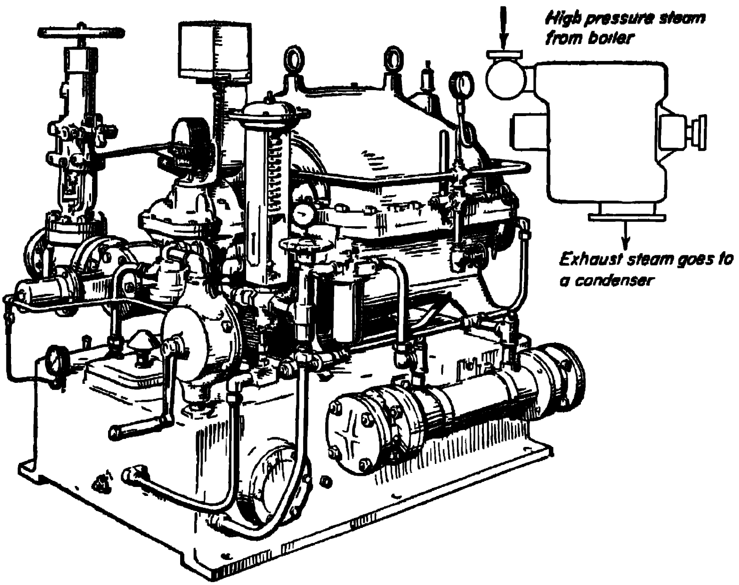

High pressure steam from boiler
Exhaust steam goes to a condenser

Fig. 8-212 High-speed condensing multistage turbine with combined trip and throttle valve direct-acting variable speed oil governor with manual- and pressure-operated speed control electric trip on emergency overspeed governor hand oil pump for starting and steam-sealed glands

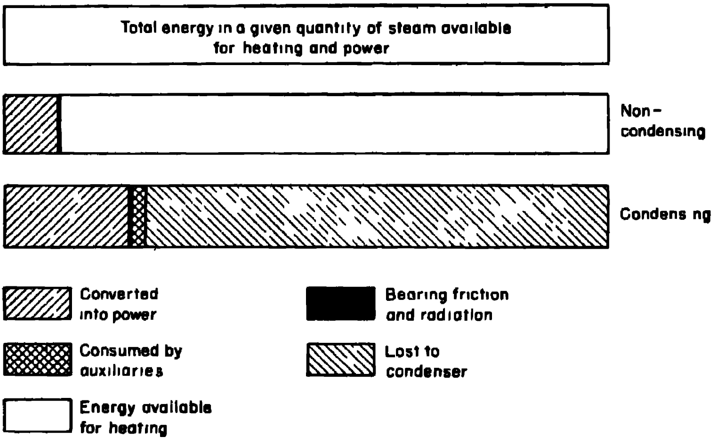


Fig. 8-213

lb gauge pressure Such a machine, however, would not be profitable for a bleeder pressure of 15 lb with a secondary requirement for steam at 5 lb For such conditions a straight back pressure turbine exhausting at 15 lb should be used

The bleeder turbine is not often built in sizes below 500-hp since its initial cost is too high when compared with the savings that can be obtained with lower horsepower ratings

The bleeder turbine is also built as a mixed pressure or mixed-flow unit (Fig 8-216) This is similar to the straight bleeder turbine, but in addition to bleeding steam, the valve gear is arranged to admit any surplus steam from the process line to the turbine low-pressure stages. With this flexible installation it is possible either to bleed steam to or absorb steam from the process line. The valve gear takes care of

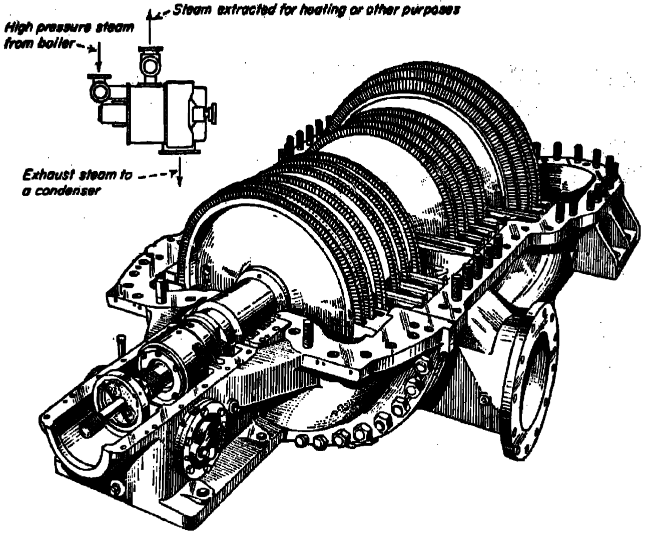
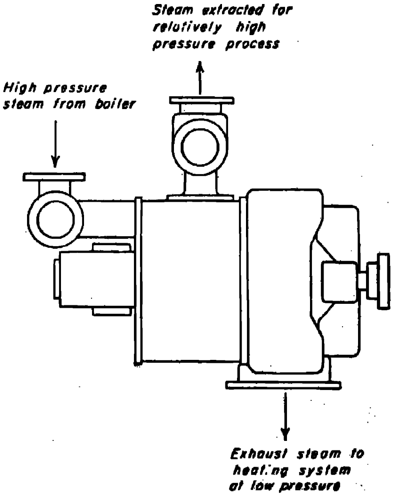

Steam extracted for heating or other purposes
High pressure steam from boiler
Exhaust steam to a condenser

FIG. 8-214. Partly assembled 12-stage bleeder turbine which bleeds steam after the fifth stage.

this extraction and absorption automatically. The steam passed in or taken out of the turbine is determined by the steam pressure in the process line.


Steam extracted for relatively high pressure process
High pressure steam from boiler
Exhaust steam to heating system at low pressure

Low-pressure Turbine. Low-pressure turbines, in general, operate with initial steam pressures at or slightly above atmospheric and exhaust to a condenser (Fig. 8-217). The energy available in 1 lb of steam at atmospheric pressure expanded to 28 in. Hg vacuum is approximately the same as for steam expanded from 150 lb gauge to atmosphere.

Plants with a reasonably constant amount of surplus low-pressure steam can convert it into mechanical power with this type of turbine. The turbine is much larger than a high-pressure machine, has more stages, and requires a condenser. Therefore, it has a high installation cost. It can often make very attractive savings in spite of this fact, however.

If the low-pressure steam supply is erratic, a mixed-pressure machine should be used. In the mixed-pressure turbine, low-pressure steam is absorbed and converted into power; if there is not enough low-pressure steam to carry the load, high-pressure steam is automatically admitted to the turbine to maintain the power output.

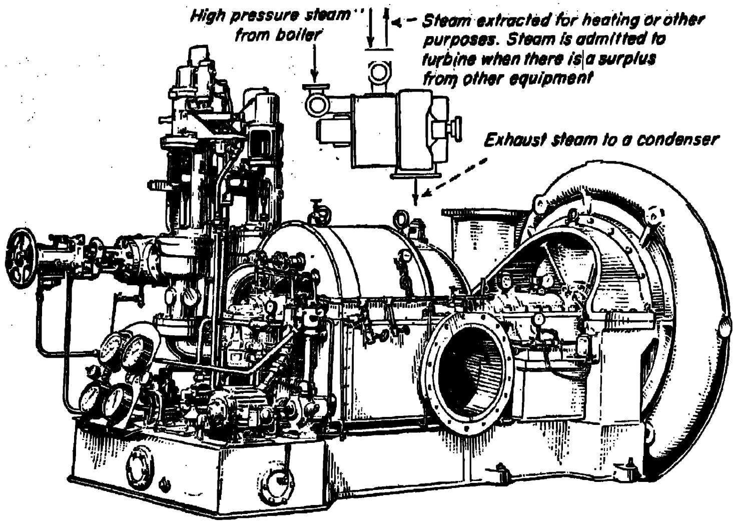

High pressure steam from boiler
Steam extracted for heating or other purposes. Steam is admitted to turbine when there is a surplus from other equipment
Exhaust steam to a condenser

Fig. 8-216. Mixed-pressure bleeder turbine direct-connected to a centrifugal blower.

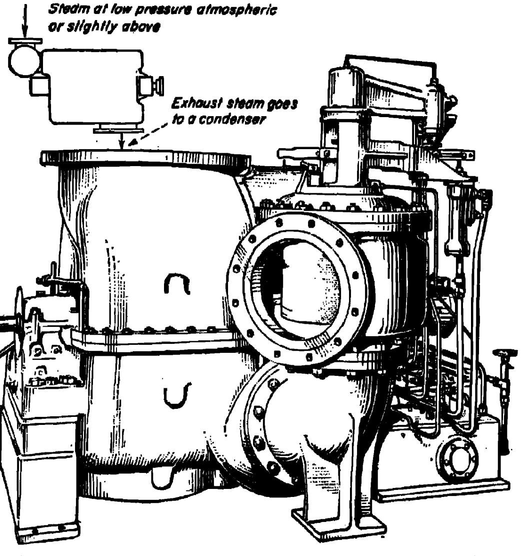

Steam at low pressure atmospheric or slightly above
Exhaust steam goes to a condenser

Fig. 8-217. A 980-hp, 4,000-rpm, low-pressure turbine operating with 2 lb gauge inle steam pressure and exhausting to a condenser.

MAXIMUM PRESSURE AND TEMPERATURE CONDITIONS

Most steam turbines are so constructed that in emergencies they will withstand increased initial pressures of 10 per cent or 50 lb gauge, whichever is lower. They are also constructed so that in emergencies they will withstand increased initial temperatures of 25°F for periods not exceeding 15 min in duration and which do not occur more than twice in 24 hr.

Manufacturers design machines for various initial steam-pressure and temperature classifications as shown in Table 8-64.

Table 8-64*

Initial steam pressure, psi gauge	Initial steam temperature, deg F					
	Up to 500	501–600	601–750	751–825	826–900	901–950
250 and lower	x	x	x			
251 400	x	x	x	x		
401 600		x	x	x	x	
601 900			x	x	x	x
901–1500				x	x	x

* Nema Publication 43-88.

PERFORMANCE

Water Rate. Steam-turbine performance is indicated by water rate (or steam consumption) which is expressed in pounds per brake horsepower per hour. Calculations usually involve total steam flow in pounds per hour, which is brake horsepower multiplied by water rate.

Manufacturers determine water rates from actual test data. Since steam conditions, horsepower, speed, and turbine type all affect water rates, no attempt is made here to indicate them. Manufacturers' data should be consulted for this information.

Heat in Exhaust Steam. For heat-balance calculations it is necessary to determine the Btu contained in the turbine exhaust steam. The amount of heat exhausted from a turbine cannot be determined directly from steam tables. The heat per pound of exhaust is equal to the total heat of the steam when it enters the turbine, minus the heat loss due to radiation and minus the heat utilized in making power. A convenient formula for determining the heat in the exhaust steam is

$$H_L = H_1 - \left(H_R + \frac{2{,}545}{\text{Water rate, lb/bhp-hr}}\right)$$

where H_E = Btu per lb of exhaust steam

H_1 = total heat of steam, Btu per lb at turbine-inlet conditions

H_R = heat radiated, Btu per lb

For H_R, a close approximation is 3 per cent of the difference between the total heat of the steam at the turbine inlet steam condition and the total heat at the turbine exhaust pressure after adiabatic expansion (constant entropy).

Temperature of Exhaust Steam. Turbine-exhaust steam temperature is determined on the Mollier chart at the intersection of the exhaust-pressure curve and Btu per pound of exhaust steam.

Efficiency. Theoretical water rate in pounds per horsepower-hour is equal to

$$2{,}545/(H_1 - H_2)$$

where 2545 = Btu equivalent of 1 hp

H_1 = total heat of steam in Btu per lb at turbine-inlet conditions.

H_2 = total heat of steam in Btu per lb at turbine-exhaust pressure after an adiabatic expansion (at constant entropy)

$$\text{Thermal efficiency} = \frac{\text{theoretical water rate}}{\text{actual water rate}} \times 100$$

Rated-load Water Rates. Water rates are usually given at rated-load and rated-steam conditions. Nozzles are selected for a steam-chest pressure which allows for the pressure drop through the turbine governor valve. Therefore, the steam-chest pressure is close to the inlet pressure when the turbine is operating at rated conditions, and high nozzle efficiency is obtained.

Partial-load Water Rates. At loads under rating the turbine will overspeed unless the steam flow is reduced by the governing valve. When the flow is reduced, the steam-chest pressure, the effective heat in the steam, and the nozzle efficiency are lowered, and water rate is increased at the lighter load.

Hand-valve Control at Partial Load. Hand valves to close individual nozzles can be supplied. Closing one or more of these valves at partial load reduces the total nozzle area, and therefore, more governor-valve opening is required. This raises the steam-chest pressure ahead of the nozzles so it is closer to the design pressure, and improved efficiency and water rate are obtained. A turbine can be designed with hand-valve-controlled nozzles to permit it to carry a peak load without sacrifice of efficiency at the lower normal load.

The dotted line in Fig 8-218 is a typical water-rate curve for a mechanical-drive turbine operating at constant speed with governor or throttling control. The effect of closing successive hand-valve-controlled nozzles is shown by the solid line

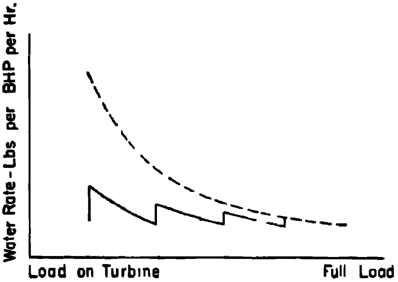


FIG. 8-218.

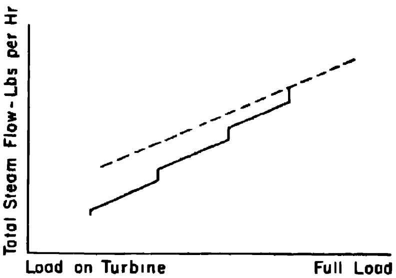


FIG. 8-219.

Figure 8-219 shows total steam flow for the same machine; solid line shows the effect of closing hand valves.

SELECTION

Heat Balance. Selection of a turbine generally depends on heat balance. Steam is required in all boiler plants for heating feed water Industrial boiler plants built to supply process steam can use turbines to drive auxiliaries such as boiler feed pump, forced- and induced-draft fans, pulverizers, etc. It is necessary only to choose turbines which will not supply exhaust steam in excess of feed-water heating or other process-steam requirements.

In chemical plants, oil refineries, etc., where initial steam pressures are 400 lb or above, mechanical drives are sometimes employed which exhaust to process-steam lines at pressures as high as 200 lb A single-stage turbine can be used, but since first cost is higher where back pressures are above 75 lb gauge, these types cannot be used economically, as a rule, in sizes below 200 hp

In plants of this type, fire hazard may be reduced by using turbine drive.

Starting and Emergency Drives. Central stations and industrial plants generating electric power frequently install steam-turbine drives for starting or emergency service These machines will not be operated for long periods, so water rate is seldom an important consideration. Such turbines may drive boiler feed pumps, emergency exciters, etc.

This type of drive should be simple, rugged, and capable of being started without preliminary warming up In general, all mechanical drives should be selected on this basis if possible, and the more complicated multistage type of turbine should be used only where a low water rate is required.

APPLICATION

Turbines are most efficient when operated at relatively high speeds. Standard designs range in speed from 1,000 to 5,000 rpm Below 1,000 rpm the ordinary mechanical governor furnished on a turbine will not control the speed closely. Also, at the lower speeds water rates are high In general, for slow-speed driven equipment it is necessary to use a high-speed turbine with a reduction gear. Turbine speeds above 5,000 rpm are common, but special construction at higher cost may be required.

Governors. Turbine speed is controlled by a governor that positions the turbine inlet or governor valve. Some driven machines run at constant speed, others operate at varying speed depending on load, etc. Different governors are available for taking care of driven-machine speed-control requirements.

Direct-acting Centrifugal Governor. This governor is of the flyball type and is commonly called a constant-speed governor (Fig. 8-220). It is standard equipment on most mechanical-drive turbines. It consists of governor weights A mounted on the turbine shaft. Centrifugal force of the governor weights is opposed by a spring. An increase in speed above that for which the governor is set causes the weights to move apart, and their movement closes governor valve B through the governor lever.

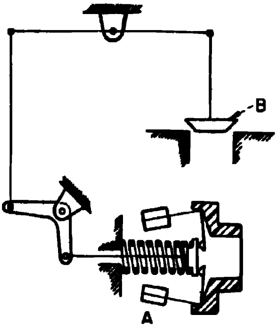


Fig. 8-220.

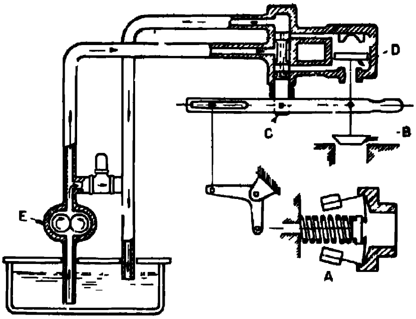


Fig. 8-221.

This governor is usually sufficient to control satisfactorily the smaller turbines that drive pumps, fans, pulverizers, etc.

Oil Relay Centrifugal Governor. The oil relay governor operates on the same principle as the direct-acting type (Fig. 8-221). However, instead of moving the governor valve directly, the governor weights actuate an oil pilot valve C which puts oil pressure on piston D to open or close governor valve B. As soon as the governor-valve position is changed, the pilot valve returns to its neutral position, and the governor valve does not move again until there is another impulse from the governor weights This compensation, together with the short travel of governor weights and pilot valve, results in speed regulation of about three per cent. In addition to being used where close speed regulation is needed, the oil relay governor is required on turbines with large governor valves and often for initial steam pressures above 650 lb.

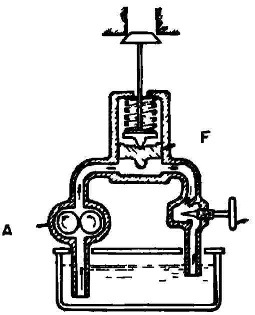


Fig. 8-222.

Oil pressure for operating this governor is supplied by a turbine-shaft-driven oil pump.

The direct-acting and oil relay centrifugal governors can be arranged to permit a small speed adjustment while the turbine is in operation, but this will not exceed 10 to 20 per cent.

Direct-acting Variable-speed Oil Governor. The direct-acting variable-speed oil governor is used on turbines driving machines that require variable-speed operation (Fig. 8-222). The speed of the turbine can be set at any operating point (within the speed range of the governor) while the unit is running. A turbine-shaft-driven oil pump A_1 supplies oil pressure to governor valve operating piston F and discharges back to a reservoir through adjustable orifice valve G. With the orifice valve set for the desired speed, the oil pump maintains just enough oil pressure at F to keep the governor valve sufficiently open to carry the load on the turbine. If the speed increases slightly, there is an immediate and corresponding increase in the oil pressure which causes the governor valve to close slightly and restore the speed. Speed is changed by adjusting orifice valve G.

Speed control may be obtained by the following methods:

Hand control as in Fig. 8-222
Motor-operated control from a remote point
Pressure-operated control by a regulator
Mechanical linkage control

Speed regulation with this type of governor is approximately ten per cent.

The direct-acting variable-speed oil governor can be arranged to permit a speed range as high as five to one, provided the turbine shaft speed is reasonable at the low operating point.

Oil Relay Variable-speed Governor. The oil relay variable-speed governor is used on turbines driving large centrifugal compressors, pumps, etc., where variable-speed operation is necessary and when close speed regulation is required (Fig. 8-223). The principle of operation is the same as that of the direct-acting variable-speed oil governor, but instead of moving the governor valve directly, piston *F* operates pilot valve *C* through which oil is applied to piston *D*. Two turbine-shaft-driven oil pumps are required by this governor. Methods of speed control are the same as for the direct-acting oil governor.

This governor can be built for a wider speed range than the direct-acting type.

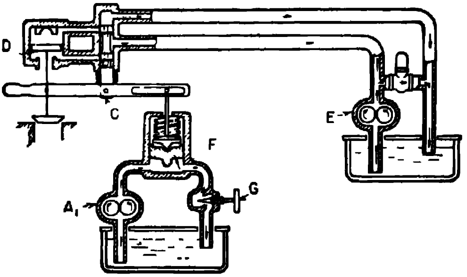


FIG. 8-223.

Emergency Overspeed Governor. All turbines should be equipped with an emergency overspeed trip to reduce the steam supply to the turbine and prevent overspeeding. Usually this governor is set to trip at 10 to 15 per cent overspeed.

When an oil governor is used, it is desirable to supply circulating oil to the turbine-shaft bearings from the governor oil pump.

Selection and application of mechanical-drive turbines vary according to driven-machine requirements.

Boiler-feed-pump Drive. The turbine should be simple, reliable, and capable of being started without preliminary warming up. Usually, efficiency is not paramount since exhaust steam will be utilized for heating feed water, or the unit will be an emergency stand-by to an electrically driven pump. The turbine drive should be rated a little above required pump brake horsepower to allow for gradual pump-impeller wear.

Hand Valves. In general, hand valves are not useful on boiler-feed-pump-drive turbines. These units operate over a wide load range, and an operator should not be expected to open and close hand valves whenever the load changes. Water rates at partial loads, therefore, should be figured without the use of hand valves. However, a hand valve may be justified for an infrequent overload condition. Also, there are installations where an extra hand-valve-controlled nozzle is necessary to develop the required power when starting the plant at low boiler pressure.

Control. The turbine should be equipped with a standard constant-speed governor, and it should also have an emergency overspeed trip.

Automatic control may be obtained by fitting the turbine with a differential pressure governor. This type of governor maintains a constant differential between discharge pressure of the boiler feed pump and steam pressure in the boiler. In principle, it consists of a diaphragm-operated steam-supply valve with pump discharge pressure connected to the upper side of the diaphragm and steam pressure connected to the lower side with a spring to balance the two pressures. An increase in pump discharge pressure on the diaphragm will tend to close the steam-supply valve and reduce turbine speed and, therefore, pump capacity. At loads under rating, the pump will require less power for a given capacity if the speed is reduced.

It is preferable that water and steam connections of the pressure governor be connected to water and steam lines close to the boiler in order to keep the differential as low as possible and to minimize the effect of pipe friction for varying boiler loads.

If a turbine-driven boiler feed pump feeds only one boiler, this governor may be connected across the boiler water-level control valve. The water side of the diaphragm would be connected near the valve inlet and the steam side between the outlet and the boiler.

With the three governors, the pump would be controlled by the differential pressure governor, the constant-speed governor would be set at a speed a little above the maximum required and the emergency overspeed trip about 10 per cent higher. If the differential pressure governor becomes inoperative and opens wide, the constant-speed governor will act as a speed limit and prevent the overspeed governor from shutting the unit down. The differential pressure governor can be built into the turbine, or it may be installed in the steam line ahead of the turbine.

Where a turbine-driven boiler feed pump is to operate in parallel with a motor-driven pump, the two units must run at approximately the same speed in order to divide the pumping load, and a differential pressure governor will not be useful on the turbine.

Boiler Fan Drive. Many forced- and induced-draft fans are turbine-driven. Where the exhaust steam can be used for feed-water heating or process work, turbines are economical drives. In addition, variable speed, which is usually essential, can be obtained easily with turbine drive by throttling the steam supply.

The drive must be selected for the horsepower and speed of the fan, and it must be capable of developing the required power with the low steam pressure which may prevail when the boiler is started. Also, it is frequently necessary to have additional capacity so that the fan unit will perform satisfactorily with lower than design air or gas temperatures.

Smaller fans, and particularly forced-draft fans, are often direct-connected to turbine drives. Induced-draft fan drives and duplex units with forced- and induced-draft fan wheels on the same shaft run at speeds around or below 1,200 rpm, and it is frequently necessary to use a high-speed geared turbine, so the steam required to drive the turbine will be within the heat balance limitation.

The reduction gear must be selected for the type of fan it is to drive. Induced-draft-fan-drive service is more severe than forced-draft-fan drive. When a coupled gear is used, turbine and gear should be mounted on a common base plate so that alignment will be maintained.

Hand Valves for Overload. One or more extra hand-valve-controlled nozzles can be provided to give the extra power required when the fan handles air or gas at low temperature and for starting with low steam pressure. This will permit good efficiency at the normal rating.

Partial-load Hand Valves. Most fans operate at loads under normal rating. Considerable steam saving can be made by closing hand valves at partial loads when the horsepower requirements are lower and fan speeds are correspondingly reduced.

Control. It is desirable to control fan capacity by varying the speed because at reduced loads the fan power requirement is less than if it runs at full speed with the control obtained by dampering. To lower the speed of a turbine it is necessary only to reduce the steam flow. This can be done by hand operation of a valve in the steam line, by combustion-control operation of a valve in the steam line, or by changing the speed setting of a variable-speed turbine oil governor by the combustion control. Usually hand control is used only on smaller installations. Satisfactory results are obtained with an automatic combustion-control valve installed in the steam line, but generally a damper is required also. Use of a damper is minimized with the variable-speed turbine oil governor because the combustion control sets the speed of the turbine directly, and steam flow to the turbine is controlled to maintain speed (within the capacity of the turbine) regardless of changes in load and initial steam pressure or back pressure. The variable-speed oil governor is relatively expensive and usually requires a fairly large drive to justify the extra expense.

Generally, it is not desirable to connect the combustion-control system direct to the main turbine governor valve. The travel of this valve is short, for the full speed range and lost motion in the connecting linkage will not permit the small valve adjustments required at light loads.

Some kind of control should be provided to shut off fuel in the event that the induced-draft fan stops. In general, the induced-draft-fan drive will be running as long as there is steam pressure in the steam chest ahead of the turbine nozzles. A pressure control can be connected to the turbine steam chest which will operate to trip the fuel supply if this pressure goes below a predetermined point.

Pulverizer Drive. Turbines that drive coal pulverizers are direct-connected or geared depending on the amount of exhaust steam that can be absorbed. When a geared unit is employed, the gear must be selected specifically for pulverizer drive since this is rougher than the usual drive. Furthermore, coal pulverizers require a drive with a high starting torque, and this starting torque should be specified.

Hand Valves. One or more extra hand-valve-controlled nozzles should be supplied. The high starting torque must be provided for, and furthermore, the turbine must be capable of developing this torque with low boiler pressure when the plant is started. These hand-valve-controlled nozzles allow best efficiency at the normal load and at the same time provide necessary overload capacity. Partial-load hand valves are not usually required.

Control. A pulverizer runs at constant speed, and usually a mechanical constant-speed governor and separate emergency overspeed trip are sufficient to control the driving turbine.

Provision should be made to shut down the pulverizer (and the forced-draft fan) in the event that the induced-draft fan stops. For the pulverizer turbine this may be accomplished by installing a valve in the steam-inlet pipe arranged to shut off the steam automatically when the induced-draft fan stops.

Auxiliary Exciter and Emergency Electric-generator Drive. Auxiliary exciter and emergency electric-generator drives usually are required only for emergency power. The turbine should be as simple as possible and should be capable of starting without preliminary warming up. A single-stage turbine should be used unless the unit is likely to be operated for considerable periods of time when excess exhaust steam might be a problem. A multistage turbine would not be used, ordinarily, on an exciter drive but might be used to drive an emergency electric generator. However, this type cannot be started so quickly as the simpler single-stage turbine. Some single-stage machines can be started automatically if necessary, with a solenoid valve in the steam line ahead of the turbine. For such an installation the exhaust valve should always be kept open and the casing and steam-inlet pipe drained at all times.

Hand Valves. These types of units may have to start and operate with reduced boiler pressure, so one or more extra hand-valve-controlled nozzles should be provided.

Control. A mechanical constant-speed governor is usually satisfactory for generator drives up to 50-kw capacity unless close regulation is required. For larger capacities, an oil relay governor is required in order to avoid excessive speed change and voltage variation with the heavier electrical loads.

Paper-machine Drive. Turbine units can drive either variable-speed or constant-speed paper-machine line shafts. The drive usually consists of a high-speed turbine connected to a reduction gear which can be coupled or belted to the line shaft. If belted, the driving pulley should be mounted on a separately supported jackshaft with a flexible coupling connecting it to the slow-speed gear shaft. This eliminates belt pull on the reduction-gear bearings. Sometimes a drive is installed in the middle of a line shaft with the slow-speed gear shaft of the drive unit extended in both directions. For small drives this type may be too expensive because the gear must have a large center distance in order for one slow-speed shaft to clear the turbine casing. This drive is simple, relatively inexpensive, and it supplies steam necessary for drying paper. When higher speed is contemplated on an existing paper machine, a new turbine is the least expensive drive, takes up little space, and a heavy foundation is not required. Turbine exhaust steam is as free of impurities as boiler steam, and an oil separator is not required.

Partial-load Hand Valves. Since the exhaust steam from the turbine is used for drying paper, the amount required (and also the mechanical load on the turbine) will vary for different grades of paper. The turbine should be equipped with partial-load hand valves, so best efficiency can be obtained for any weight of paper.

Overload Hand Valves. It is advisable to include one or more extra hand-valve-controlled nozzles for overload at starting or low steam pressure.

Control—Variable- and Constant-speed Line-shaft Drive. Variable-speed line-shaft drive requires close speed regulation to prevent paper thickness variation or possible paper break. An oil-relay variable-speed governor must be provided, and on most installations a governor adjustable over a six to one speed range will be satisfactory. The governor-speed setting should be adjustable by hand at the turbine itself, and if desired, the unit can be equipped with a remote push-button-controlled motor-operated speed-changing device.

A push-button-operated solenoid on the emergency overspeed governor can be supplied. This device permits shutting down the turbine in emergency from a remote operating point.

The less expensive oil relay constant-speed governor is sufficient to control the constant-speed line-shaft type of drive.

Centrifugal-compressor Drive. A turbine is efficient at usual centrifugal compressor or exhauster speeds, and these units are direct-connected. Compressor load varies with speed. Therefore, automatic control of a turbine-driven unit is relatively simple. These units require substantial power, and multistage condensing, bleeder, mixed-pressure, and low-pressure turbines, in addition to the back-pressure type, are economical drives in many cases.

Low-pressure condensing turbines can make power for summer air conditioning from low-pressure or exhaust steam that is wasted when there is no building heating load.

When more than one refrigeration compressor is required, as in a brewery or chemical plant, good heat balance may be obtained sometimes with one compressor driven by a single-stage reducing-valve type of turbine and a second compressor driven by a condensing mixed-pressure bleeder turbine. This type of installation permits efficient operation with the relatively steady compressor power demand but takes care of wide fluctuations in process-steam requirements. Both machines exhaust to the process-steam line, and if too much steam is supplied, the valve gear on the mixed-pressure bleeder turbine admits surplus to the low-pressure stages, and power is made with a low condensing water rate.

New centrifugal air-conditioning or refrigeration equipment adds considerably to the power load of any plant. A turbine drive for new equipment of this type should be considered when there is sufficient boiler capacity to take care of the added steam load and if electric drive will load plant-generating capacity so much that there is little margin for expansion. Turbine drives are much less expensive than new generating capacity.

Water is required by the condenser of a centrifugal-refrigeration or air-conditioning unit. Usually, the same water is sufficient to condense all exhaust steam when a condensing turbine is used.

Hand Valves. Partial-load hand valves should be supplied on all but low-pressure, bleeder, and mixed-pressure turbines. Bleeder and mixed-pressure turbines probably should be equipped with automatic nozzle control or sectional inlet valves, although this depends on the character of the process steam and refrigeration loads.

Control. An air-conditioning or refrigeration compressor-drive turbine should have a variable-speed oil governor. The unit can then be arranged so that refrigerant temperature will actuate a small control valve in the turbine-governor oil line to vary the turbine speed automatically and, therefore, the compressor capacity in accordance with refrigeration requirements.

Gas compressors and exhausters can be controlled in a similar manner when the driving turbine is equipped with a variable-speed oil governor.

Dual Drive. Oil-refinery pumps, exciters, etc., that must operate continuously are sometimes dual driven with a turbine at one end and an electric motor at the other. If power fails when the unit is electrically driven, the turbine automatically takes over the drive at slightly reduced speed.

Dual drive may be used for automatic heat balance since the driven load can be divided in any proportion between motor and turbine, depending on the amount of exhaust steam required.

A dual-drive turbine to be installed only as protection against motor failure can sometimes be selected for a capacity less than the rating of the driven machine. For example, an emergency turbine driving an oil-refinery pump could be designed to develop only enough power to maintain sufficient liquid flow to prevent damage to other equipment.

Control. The turbine must be equipped with a constant-speed governor which is adjustable during operation. With the motor driving, the turbine governor is set just below rated speed. On failure of the electric drive, the turbine-governor valve opens, and the turbine drives at slightly reduced speed. To restore speed, the turbine governor must be reset.

If required, the turbine governor may be solenoid-operated so it will take over the load at rated speed on failure of the motor.

It is usually necessary to bleed a small quantity of cooling steam into the turbine casing when the unit is motor-driven.

The exhaust valve of a dual-drive turbine must be open at all times.

INSTALLATION

Foundation. A turbine does not require a massive foundation, and it is usually necessary only to have rigid support for the equipment. Foundation size and weight will depend on the unit being supported and whether it is on the ground or on a floor. If on the ground, the foundation should be isolated from the building floor. Small units are frequently mounted on concrete pads built into strong concrete floors. The floor and pad must be substantial enough to keep the turbine and driven machine in line.

Turbine units are often supported on steelwork. The structure design must not permit alignment of the machines to be disturbed by possible springing. Building a concrete pad under the unit base plate will add stiffness and help to absorb any vibration or noise.

Turbines may be mounted directly on concrete foundations, provided soleplates to go under them are set in the concrete. Steel plates with bolts extending down into the concrete are satisfactory, and sections of channel or I beam are used also. Accurate alignment can be obtained by putting shims between the turbine and the soleplate. Machines should be bolted and, after alignment, doweled to the soleplates. They should never be set directly on the concrete.

Foundation bolts to hold a base plate should be provided with pipe sleeves. These bolts should not be set rigidly in the concrete since some movement of the base plate will be required during alignment.

Alignment. High-speed machinery will not operate satisfactorily if the shafts are not in good alignment. Flexible couplings are not universal joints. Shafts must be lined up as exactly as possible, so the flexible coupling will only be required to compensate for the slight unavoidable misalignment which may occur during operation. Alignment of shafts can be determined by laying a straightedge across the rims of both coupling halves.

Units are lined up cold, and a turbine, when hot, will usually rise more than the driven machine. Therefore, a turbine is usually set lower than the driven machine, with the vertical distance between shafts depending on the drive and size of the unit. A boiler feed pump may be almost as hot as its driving turbine, so the two shafts should be more nearly in line when units are cold. In some cases the steam end of the turbine will rise more than the exhaust end, and it is necessary to allow for this tilting when lining up the units cold. This condition may occur with condensing turbines. The alignment should always be checked when the equipment is hot and any necessary final adjustments made then.

Clearances between coupling faces must be such that the two shafts will not bump. Normally, this clearance will vary between 1/8 and 1/4 in. depending on the machines and type of coupling. More clearance may be required with some equipment such as induced-draft-fan drives, for example, which operate with shafts at high temperature.

Sometimes a turbine and its driven machine are aligned on a common base plate at the manufacturer's plant, and the units are doweled in place. No matter how heavy it is, a base plate will deflect, and it is necessary to line up the units after the whole assembly is placed on the foundation. In such a case the base plate should be supported on the foundation by metal wedges, allowing about an inch for grouting. Foundation bolts should be in place but not tightened. Units should be uncoupled. The wedges can then be adjusted until the machined pads on the base plate are level. The alignment should then be checked at the coupling halves, and if not accurate, the wedges can be adjusted to bring the units into proper alignment. After this has been done, the base plate may be grouted in place. The foundation bolts should not be tightened until the grouting has set. A unit consisting of two or more machines on a common base plate must not be grouted in place until it is accurately lined up as indicated above.

Steam and Exhaust Piping. Turbines are furnished with standard inlet and exhaust connections, and steam pipes are larger or smaller than these depending on turbine steam requirements. Pipe sizes must be figured with allowances for pressure drop due to length of pipe, number of valves, fittings, etc. Turbines are rated with steam pressure at the turbine-inlet flange and back pressure at the exhaust outlet. Low initial steam pressure or high back pressure at the turbine, due to small steam pipes, will reduce power and increase water rate.

Steampipe. A throttle valve should be installed in the steam line near the turbine inlet. If the steam is moist, a steam separator should be placed in the steampipe close to the turbine. Excessive moisture in steam will cause rapid blade wear. All multistage turbines, all low-pressure turbines, and all turbines operating with high-pressure saturated steam (400 lb and above) should have a separator.

Exhaust Pipe. A shutoff valve should be installed in the exhaust line near the turbine, and to protect the turbine casing which will not withstand full inlet pressure, an atmospheric relief valve should be installed between the turbine and the shutoff valve.

Steam- and exhaust pipes should be blown out before they are connected to a turbine.

Pipe Strains. Pipes should be arranged and supported so that strains due to weight or expansion will not be imposed on the turbine. Where piping cannot be so arranged, an expansion joint must be provided next to the turbine. This is almost always required in a large exhaust pipe. Alignment should be checked when hot to make sure that the piping does not distort the turbine casing.

Small Auxiliary Piping. *Drains.* Drains must be provided in the steampipe ahead of the throttle valve and in the turbine casing. All turbine casings are drilled and tapped for a drain, and a short length of pipe with a valve at the end, which allows a free blow, is usually sufficient.

When steam is wet, the turbine casing should be trapped to prevent water accumulation during operation. Blades will wear rapidly and may be broken off if run in water; also, wheel forgings are likely to be damaged.

Water Cooling. If turbine bearings are provided with water cooling, the water inlet pipe must have a valve to control the flow, but the outlet should be free. It is customary to connect water jackets or coils in series.

Glands. Most mechanical drives employ carbon ring glands to prevent steam from leaking along the shaft (Fig. 8-221). When operated with back pressure, no sealing is required, and the leak-off pipe from each gland should be open to atmos-

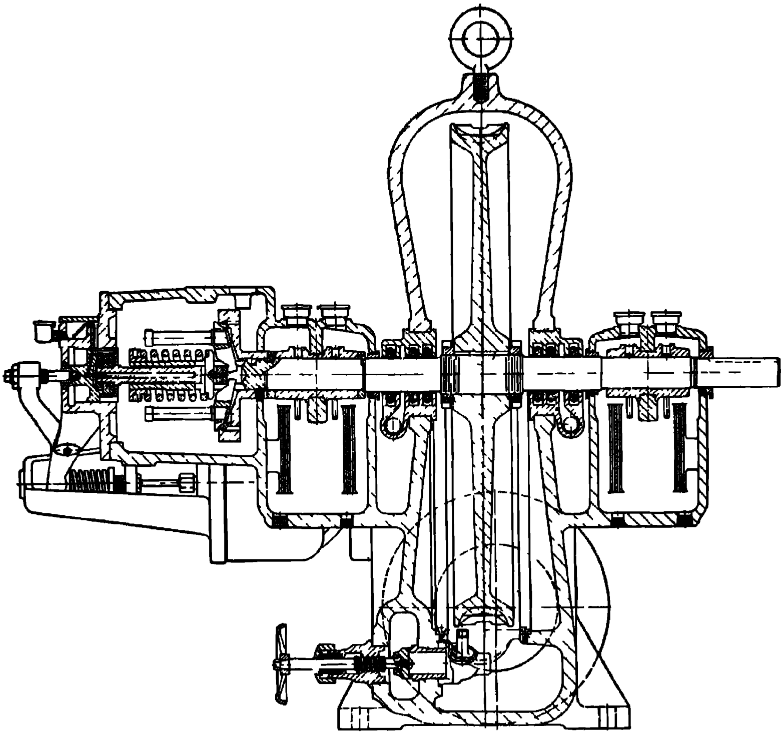


FIG. 8-224. Section through small single-stage turbine showing carbon ring glands and leak-off connections.

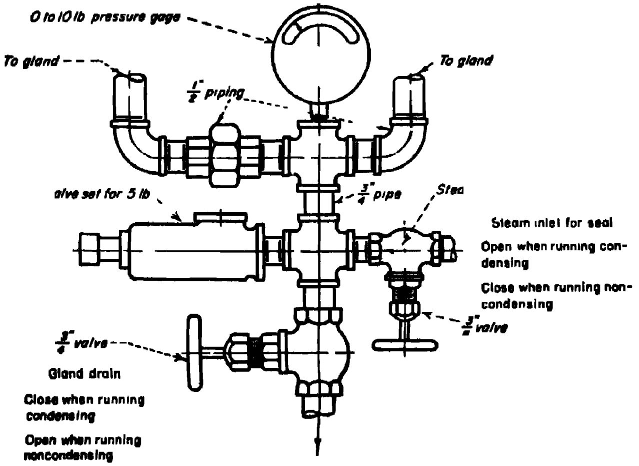


FIG. 8-225. Typical steam-seal piping arrangement for carbon ring glands on condensing turbine.

phere. A valve should not be installed in these lines, nor should they be connected to any other steam line which might cause pressure to build up in the glands and force steam out along the shaft and into the bearings.

Condensing turbine glands require steam sealing. Low pressure steam at not over 5 lb gauge pressure is connected to the leak-off space of each gland. It is desirable to control this pressure with a reducing valve or by a relief valve in the gland seal piping (Fig. 8-225).

OPERATION

Lubrication. A good turbine oil must be used. Turbine bearings run at higher temperatures than those on other types of equipment since they absorb heat from the shaft and the turbine casing in addition to that of the friction load. The manufacturer's recommendations should be taken, but in general, the following tabulation indicates the viscosity range for different bearing temperatures. Viscosities are given for ring-lubricated bearings and for forced-feed or pressure-lubricated bearings.

Table 8-65. Viscosity Ranges for Different Bearing Temperatures

Operating temperature of oil in bearing	*Viscosity SSU*
Ring Lubrication	
Up to 200°F	225–350 at 100°F
Over 200°F	115–150 at 210°F
Forced-feed Lubrication	
Without reduction gear	150–160 at 100°F
With reduction gear	
Up to 180°F	250–350 at 100°F
Over 180°F	100–580 at 100°F

For a turbine and gear unit with a common pressure-lubricating system the lubricant should be selected to suit the gear.

First Run. The manufacturer's instructions should be read carefully, so that the function of all essential parts is understood.

Before starting a new turbine, or one that has not been used for a considerable period of time, the machine must be thoroughly cleaned. All working parts should be wiped clean. The bearings, oil reservoirs, and any oil piping should be flushed with kerosene. Main bearings and other oil reservoirs, including flexible couplings of the lubricated type, should be filled with oil to the proper level. The machine should be rolled over by hand and the governor valve moved to make sure the rotor does not rub and the valve does not stick.

The exhaust valve and all drain valves should be opened. Water should be turned on for any water jackets or cooling coils. The throttle valve should be cracked open to warm the machine and to turn it over slowly. Drains may be closed as soon as water stops running from them. The oil rings should be inspected to make sure that they revolve. If there is a shaft-driven oil pump on the turbine, the pressure should build up quickly as the speed increases. If the unit appears to operate satisfactorily, the throttle valve may be opened gradually and the unit brought up to rated speed. Bearing temperatures should be checked frequently. If bearings run hot, the units may be improperly aligned, or there may be dirt in the bearings.

The speed should be checked frequently, and the governor should begin to operate when near rated speed. With no load, the unit will run from approximately three to six per cent above rated speed, depending on the type of governor. Not until the governor is controlling the speed of the unit should the throttle valve gradually be opened wide.

The emergency overspeed governor should be tested by overspeeding the machine. The trip valve should close at approximately ten per cent overspeed. This governor should immediately slow the machine down below its normal speed or stop it entirely when there is no load and the throttle valve is wide open. This governor should not be reset until the unit is at or below half speed as emergency governor parts may be damaged. Also, resetting this governor too quickly may affect the speed at which the governor will trip on the next run.

The emergency governor should be tested on a regular schedule or when the machine is shut down.

Normally the lubricating oil should be changed after a 2- or 3-day initial run. After this starting-up period, oil should be changed on a regular schedule, as required.

Regular Operation. When the machine is in regular service, the turbine should be warmed up gradually and turned over slowly before bringing it up to full speed, but in an emergency mechanical-drive turbines usually can be started very quickly. This is particularly possible if all parts are drained and if the exhaust valve has been left open to keep the machine warm. Operation of oil rings and governor should be checked each time the turbine is started.

Vibration. Vibration is sometimes caused by misalignment due to foundation settling and pipe strains, or bearings may be worn or loose in the bearing boxes. If the bearings are worn, a heavier oil may allow operation until they can be replaced. Also, the rotor may rust and get out of balance if a turbine is out of service for a considerable time with the casing not properly drained.

Flexible couplings may cause vibration. A lubricated type of coupling must have proper lubrication, or the driving and driven machines, in effect, will be rigidly connected with resulting vibration. A pin and bushing coupling, with worn or sticking parts, will have the same effect. Shafts cannot move freely, and thrust is transmitted from one machine to the other. A mechanical-drive turbine is not designed to take the thrust from its driven machine, so the turbine thrust bearing may run hot and perhaps burn out when couplings are not flexible. Couplings should be inspected on a regular schedule.

Best Efficiency. For best economy a turbine should be operated with as many hand valves closed as possible. However, enough valves should be left open to take care of load swings. Hand valves should be wide open or closed tight. They should never be partly opened, as valve seats will cut rapidly, and there is no improvement in efficiency.

Low Power. When a turbine will not develop enough power, there may be an overload, or the driven-machine efficiency may have decreased because of wear. Turbine initial steam pressure and temperature may be low or exhaust pressure high.

On the turbine itself the governor may be set low, the governor valve improperly set, the steam strainer clogged, one or more nozzles plugged, or the wheel blading worn.

A clogged steam strainer or improperly set governor valve will be indicated by a large differential between steam-line and steam-chest pressure at the maximum load that can be obtained.

The steam used and the power developed by a turbine can be accurately determined. The manufacturer keeps a record of the nozzle diameters, and he can calculate steam flow through the turbine and load being developed by it if he has the following information:

Steam pressure at the turbine inlet
Steam pressure in the turbine steam chest
Superheat or moisture in the steam
Number and position of any hand valves open or closed
Speed of the turbine
Back pressure at the turbine exhaust

Steam-pressure gauges must be calibrated. The manufacturer's figures are based on test data with new machines and do not allow for wear or improper adjustments.

Speed-reducing Gears. Gears furnished as part of a turbogear set are usually of the double-helical or herringbone type.

Installation and Starting. Gears should be installed and lined up in the same manner as a turbine installation. They run at comparatively low temperatures, so when they are lined up cold appropriate allowance must be made for this. They should be started after taking the same precautions as for a turbine. It may be necessary to prime the oil pump when starting up a gear for the first time.

Lubrication. Turbine gears are very carefully made and have rather close clearances. A medium oil with a viscosity at 100°F of about 250 to 350 SSU is usually the most satisfactory. If the gear is close fitting and runs at high speeds, better operation may be obtained with a lighter oil. This is particularly true if the gear tends to overheat. The temperature range of the oil should be from 125 to 175°F. Water cooling is usually supplied so that this range can be maintained. A regulating valve should be installed in the water-supply line, but the outlet from the cooler should be open and without a valve of any kind so that it will be impossible to put excessive water pressure on the cooling equipment.

Usually high-speed gears are not designed to dip in the oil, and they will overheat and foam if this happens. Generally, oil pressure should not exceed 15 lb gauge as higher pressure may cause the gear to heat up.

Bearings. When bearings become worn, they must be renewed in a short time or the gear teeth may be damaged because of internal misalignment. Gear bearings of the babbit-lined cast-iron-shell type should never be rebabbitted since the shells will warp in the process, and it is very difficult to scrape and fit bearings so that the alignment is correct in both horizontal and vertical planes. For the same reason bearing clearance never should be adjusted by planing the bearing horizontal joint as trouble is certain to occur. New bearings should be installed in such cases.

It is very important that shaft alignment be maintained and that the flexible couplings allow the shafts to float. If couplings become rigid through lack of lubrication or wear, load is not properly distributed on the gear teeth, and the teeth will wear and eventually will break. Couplings must never be driven on or off the shafts as the tooth surfaces may be injured.

Gears are operating properly if there is a uniform marking over the entire length of both gear and pinion helixes. If there is heavier marking or pitting on one part of the tooth face, it may have been caused by misalignment of the connected shafts, vibration, worn bearings, or poor lubricant.

ASH HANDLING AND DISPOSAL

BY B. C. BERRY

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INTRODUCTION

An ash- or refuse-handling system is essential to the operation of any furnace burning solid fuel. The system may range in complexity from a hand shovel and a wheelbarrow to a modern hydraulic, pneumatic, or mechanical unit, or a combination of these.

TYPES OF SYSTEMS

The *manual system* utilizing a shovel, hoe, and similar hand-operated tools for the removal of ash from the pit and a barrow for transportation to a disposal point is so fundamentally uneconomical that its use should be considered only in the most unusual cases. It is conceivable that, with a combination of a very small installation, infrequent operation, and a low ash content of fuel, a manual system would best fit the needs and be the only one economically justified. Any such proposal should be critically viewed as to its soundness.

The *mechanical systems* include the screw type that continuously removes ashes from a V-shaped furnace bottom to a container for disposal. This type is usually applicable only to small furnaces of the household-heating-plant range or at the most to small industrial or institutional installations. The maximum size of furnace for which the screw type is ordinarily considered is one burning not over 20,000 lb of coal per day.

The *pneumatic system* should be used only on units burning not over 40,000 lb of coal per hr. If used on furnaces larger than this, the operating costs become excessive.

The pneumatic arrangement for an underfeed stoker-fired boiler should not be used. It is essential that the ashes discharged to ashpit from underfeed stokers be quenched in the pit. The application of the hydraulic system to this type of firing will be discussed.

The *hydraulic system* provides for greater flexibility in type of coal that can be burned. On units burning over 40,000 lb of coal per hr, the hydraulic system or a combination of hydraulic and pneumatic is most applicable.

For purposes of computing volume of ashes to be handled for any given problem, all coal will be assumed to contain 10 per cent ash, and coal consumption will be based on 10 lb of steam generation per pound of fuel burned.

Decision as to type of system to ensure minimum boiler outage and economy in operating costs, both power and labor, requires careful analysis of problem.

Flexibility in design is needed to permit use of varied types of coal which in turn will produce an ash ranging in fusing temperature from 1900 to 2600 to 2700°F.

Modern furnace-bottom and ash-hopper design for powdered-fuel-fired units covers two types of construction: (1) a completely dry bottom where no quenching water is introduced, as shown in Fig. 8-226, and (2) the submerged bottom which has in it a pool of water into which ash and slag are dropped and disintegrated, as shown in Fig. 8-227. The lining for each of these types is different not only in thickness but also in the materials used. In the first, a monolithic lining of thickness dependent upon temperature and type of operation is used, with or without insulation as determined by conditions. In the second, the lining is monolithic. For this type of hopper, a protection at the water line is also indicated in Fig. 8-227, and a method of cooling the protecting elements and keeping the water in the hopper quiescent so that continuous lapping along the sides cannot occur has been found to be the most effective method.

The hopper as shown in Fig. 8-226 should be selected only when furnace is to be a pulverized-fuel-fired unit of the dry-bottom type having large furnace volume with a low heat-release factor, thereby reducing to a minimum any possibility of slagging or fusing of the ash.

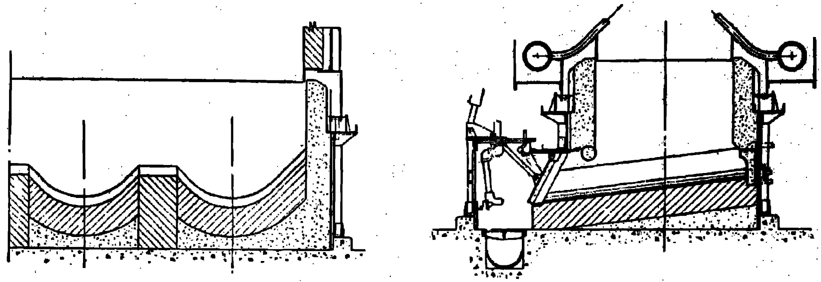

Fig. 8-226. Dry-bottom hopper.

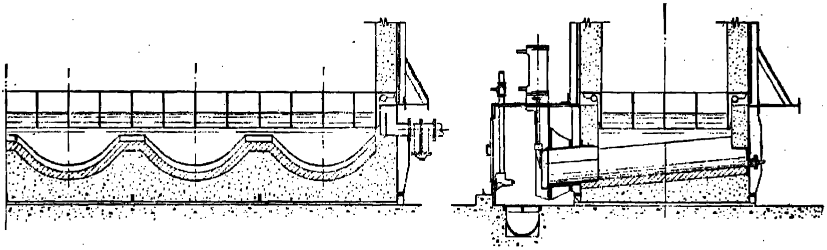

Fig. 8-227. Submerged type of hopper.

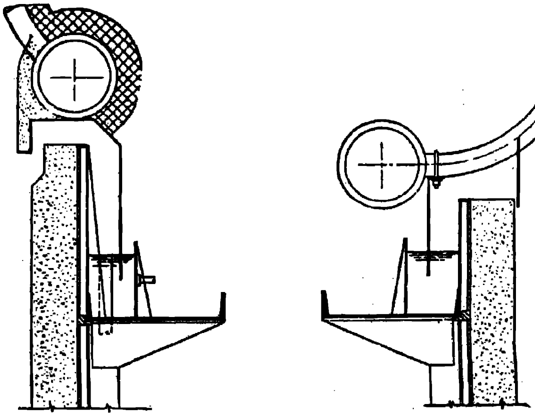

Fig. 8-228. Fluid-expansion joint.

Modern boiler design has also developed to a point where it is essential that all air necessary for combustion shall be introduced at controlled points. Leaks in the setting where excess air can infiltrate are no longer tolerated. It became necessary, therefore, to develop an expansion joint between the setting walls, whether they be refractory or waterwalls, and the furnace bottom or ash hopper so that no air infiltration is possible. This has been accomplished by the fluid seal shown in Fig. 8-228.

It may be a water seal, a sand seal, or of other material which will permit movement of the seal plate. In view of the fact that the four walls of a boiler setting may at times be on different elevations and each one expands both vertically and laterally a different amount, the development of the corners presented a problem, which was solved through the use of flexible material.

If it is a wet-bottom or slag-type steam generator, the wet hopper and hydraulic system must be used. If the ash hopper is to serve a stoker-fired boiler or steam-generating unit, the hopper can be lined with refractory shapes and equipped with quenchers as shown by Fig. 8-229.

In all these hoppers, nozzle arrangements are provided which will remove the ashes at a rate of 1 ton per min.

A powdered-fuel-fired unit of the continuous-slagging type requires a hydraulic system and the pool or submerged-type hopper, as shown in Fig. 8-227.

If the temperature in the furnace throat is high (1800°F or higher) and headroom from basement floor to waterwall headers is cramped, extra precautions must be taken at the water line. This can be done by installing alloy plates at the water line which are sprayed with water at the back of the plate. The plates are hung on the water-cooling header.

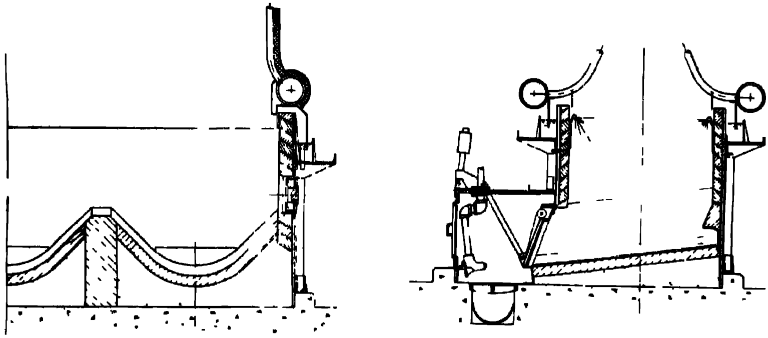

FIG. 8-229. Quenching-type hopper.

Another method of protecting the water line is to use a water-cooled hopper with either a poured- or molded-shape refractory lining.

If it is a cool furnace (throat temperatures below 1800°F), a good poured refractory lining will stand the service at the water line. A good molded shape will also perform well in this service if used on a cool furnace.

The junction of the hopper and pressure parts must provide for both vertical and lateral movement resulting from expansion. This can be done with the fluid seals as previously described, depending upon degree of air tightness desired.

If the system serves one or a number of boilers, the proper arrangement of sluice, sump, pumps, or jet is important. By proper arrangement is meant sump location which permits shortest run of sluice trenches and minimum sluice water requirements. In locating sump, thought should be given to future extension of plant to hold total water requirements down.

On stoker-fired units, the sluice and pump arrangement follows the same general design used for the powdered-fuel-fired units. This is also true for the dry-bottom powdered-fuel-fired units.

In all instances, it is good insurance to build the sump to provide for the installation of a clinker grinder which may be installed later if, owing to an excessive amount of oversize material, it is found necessary.

If it is required to discharge to fill at a distance and against a static head of any reasonable amount, the pump is the best medium for handling the material. If the total dynamic head is low, a jet can be used. The jet is not a very efficient unit, however, and does not permit much flexibility in extending the discharge line.

As the time element is a vital factor in the wear taking place on materials-handling equipment, it is important to handle the refuse in as highly concentrated volume as possible. This factor has a bearing on design arrangement.

In transporting ashes by pumping or jetting through closed pipe lines, the velocity should be maintained at not less than 9 fps on horizontal runs. If the arrangement of the line necessitates long vertical risers, the minimum pipe-line velocity is 11 fps.

The pump used in handling abrasive material should be a closed-impeller centrifugal-type assembly having a reasonably steep head-capacity characteristic. The impeller and volute must be made of abrasive-resisting material having a close grain structure and Brinnelling not less than 300. The design should incorporate sealing features to protect the packing gland and end plates against attack from abrasive-laden solution and also to eliminate short-circuiting of material around the suction shroud of the impeller.

First the ultimate disposal point should be decided upon. Will the refuse be discharged to permanent or temporary fill and at what distance from the plant? Can it be discharged to grab-bucket sump, or must it be stored for truck or railroad-car loading?

If ashes can be discharged to fill, the hydraulic system immediately takes a favorable position. This also holds true for the grab-bucket sump. If it is necessary to store the material on the property for truck or railroad-car loading, the system will incorporate the use of a storage bin into which the material can be pumped and then properly dewatered, leaving what is termed commercially dry ash or slag for loading.

In plants where the water supply is limited, the hydraulic system provides for recirculation of the water used as the transport medium. Recirculation of the water results in a closed system with no loss of water to overflow. This arrangement is practical only where the material is discharged to an overhead tank, grab-bucket sump, or specially arranged dike area located on the plant site.

In small plants requiring the removal of only a small volume of refuse each day, a simplified hydraulic arrangement can be adapted to the service, requiring no high-pressure sluicing water. The handling rate of this system is lower than the arrangements recommended for the larger plants but is suitable and economical for the small installations.

The problem of maintenance costs must be considered in determining the type of system to be used. The wear resulting from wet abrasion is less severe than that encountered from dry abrasives. The time element is also a decisive factor in maintenance costs. Therefore, in a plant where the volume of refuse to be handled in each 24-hr period is large, the materials-handling system selected should incorporate time-saving and wear-resisting features.

A recent development in furnace design on large steam-generating units provides for operating the furnace under positive pressure, thereby eliminating the necessity of using induced-draft fans.

This development requires that the furnace-bottom refuse be removed without disturbing furnace conditions through the use of a sealed hopper and sump arrangement. This requires highly specialized equipment and careful analysis of each project as it develops.

A short analysis of operating comparisons on the hydraulic system versus the pneumatic system follows.

Assume a plant with two boilers of the dry-bottom-furnace type burning pulverized bituminous coal and generating 600,000 lb of steam per hr per boiler, or a total of 1,200,000 lb per hr. The coal consumption will approximate 120,000 lb per hr. Assuming that the coal has an ash content of 10 per cent, the resulting refuse will total 12,000 lb per hr.

Of this total, approximately 40 per cent will be deposited in the furnace-bottom ash hopper. There will therefore be (0.4 × 12,000 × 24) 115,200 lb, or 57.6 tons, of ash to handle each 24-hr period.

The following is a comparison of operating costs of the hydraulic system versus the pneumatic systems:

Hydraulic

With a hydraulic system, material can be sluiced from hopper bottom at rate of 30 to 60 tons per hr. Using lower figure, system will operate 1.92 hr. Allowing for starting and stopping time, total operating time is 2 hr.

Pneumatic

In a pneumatic system, much of the material has to be manually fed into transport system with a hoe. A man can average about 4 to 5 tons per hr with a hoe. Using 5 tons per hr as the average, the system will operate 11.52 hr, plus starting and stopping time, or 11.6 hr per day.

CONCLUSION

If plant location permits depositing material to fill area, a hydraulic system should be selected.

If it is stoker-fired plant, a hydraulic system should be selected.

If boilers are of the slagging type, either intermittent or continuous, a hydraulic system should be selected.

If boilers are to be operated under positive furnace pressure, hydraulic system should be selected.

If water is scarce, the recirculating hydraulic system should be used.

If boilers do not burn in excess of 40,000 lb of coal per hr and heat release does not exceed 15,000 Btu per cu ft of furnace volume, the pneumatic system is applicable.

SECTION 9

REFRIGERATION

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CHARACTERISTICS AND PROPERTIES OF REFRIGERANTS

BY R. J. THOMPSON

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INTRODUCTION

Refrigeration has always been an essential in many industries, business enterprises, and in the home. Ice was the first and only refrigeration medium for many centuries, but there were so many disadvantages connected with its use that other means were sought, and it was found that a number of low-boiling liquids and compressed and liquefied gases could be used to produce refrigeration.

There are two principal methods in which refrigerants are employed: one, generally referred to as the indirect method, is the circulation of cooled water, brine, or chemical solutions; the other, the direct method, is the circulation and expansion of the refrigerant itself. Refrigeration either by the direct or indirect method was first employed almost entirely in industrial applications but was afterward extended to take the place of ice in commercial uses as well, and in the last 20 to 30 years has been adapted to industrial and comfort-cooling air-conditioning and household refrigeration equipment.

Unfortunately, there were certain inherent hazards in the use of the refrigerants employed, and while the hazards could be fairly well controlled in packing houses and ice-manufacturing plants because suitable precautions could be enforced, their control was not practicable in industrial, commercial, and household refrigerating and air-conditioning systems.

A refrigerant is a substance or medium used to produce refrigeration by absorbing heat in its expansion or vaporization. The word "refrigerant" is used broadly to refer to any primary refrigerating medium such as carbon dioxide, ammonia, methyl chloride, sulfur dioxide, or the Freon refrigerants, or, in another sense, to refer to a secondary cooling medium such as calcium chloride brine. As more commonly interpreted, the word "refrigerant" refers to any material used in mechanical refrigeration, by which process the refrigerant is evaporated, recovered, compressed, condensed, or liquefied and permitted to expand or vaporize again as distinguished from those processes in which the spent refrigerant medium is wasted, such as in the melting of ice or the change of state of carbon dioxide from a solid to a gas.

The requirements for the ideal refrigerant may vary in number and order of importance depending upon the particular requirements to be met or the desire of the designer, but it is believed that such requirements may be easily summed up by stating that

1. The refrigerant must be free of panic, health, fire, and property-damage hazard should it escape from the refrigerating or air-conditioning system.
2. The refrigerant must have chemical, physical, and thermodynamic properties which will be readily adaptable to and will permit its use in present-day designs of refrigerating and air-conditioning equipment and, at the same time, produce efficient service at a low initial and ultimate cost over a long period of operation.

CHARACTERISTICS OF REFRIGERANTS

In the following material will be given the desired properties for the ideal refrigerant and the characteristics or properties of all refrigerants that have been used up to the present time. The requirements that are given are all very important and are placed somewhat in an order of sequence which will show their relative importance or value.

The chief hazards were those of fire and explosion of the refrigerants themselves, health hazards due to the toxicity of the refrigerants, panic hazards brought about by the irritating and odoriferous nature of the refrigerant, the hazards due to chemical reactions resulting from accidental introduction of the refrigerant from the cooling system into the chemicals being processed or foodstuffs being refrigerated, and, in some cases, the hazards of extremely high pressures. All the refrigerants in common use prior to 1931 had one or more of these undesirable characteristics. A well-organized search continued for the ideal refrigerant, *i.e.*, one that was neither flammable nor toxic nor capable of entering into chemical reactions and which could be operated at relatively low pressures and which would be efficient and suitable for all types of systems whether for refrigerating or air-conditioning purposes.

Odor. A refrigerant must be odorless and nonirritating in any concentration in order to reduce or eliminate the possibility of a panic hazard occurring in event the refrigerant should accidentally escape from a system or leak out when the system was being repaired or being serviced.

Ammonia is an irritant, and $\frac{1}{30}$ oz in 1,000 cu ft of space imparts to the air a strong smell, and $\frac{1}{5}$ oz per 1,000 cu ft of space is the strongest dose bearable for $\frac{1}{2}$ hr. Concentrations of the odor of $\frac{1}{4}$ lb per 1,000 cu ft of space endured for $\frac{1}{2}$ hr produce blindness. This is an outstanding serious effect produced by ammonia.

Sulfur dioxide is a very irritating gas and is detectable in very small concentrations of $\frac{1}{125}$ oz per 1,000 cu ft of space, while $1\frac{1}{3}$ oz per 1,000 cu ft is suffocating.

Carbon dioxide is odorless, but its physiological effects when breathed in mild or moderate concentrations serve in a measure as a warning of its presence. In concentrations much above 11 lb per 1,000 cu ft, the physiological action may be too rapid in producing stupefaction or loss of consciousness to serve as adequate warning.

Since the odor of methyl chloride is not unpleasant and but faintly noticeable in small quantities, it probably presents a greater hazard than some of the toxic gases possessing distinct warning properties.

Methyl formate is an irritant and gives definite warning of its presence.

Suitable gas masks or oxygen helmets should be readily available when handling irritating or toxic refrigerants.

The vapors of all the Freon refrigerants are odorless at concentrations of less than 20 per cent by volume in air, which is equivalent to the release and vaporization of approximately 45 to 95 lb of liquid into a confined space of 1,000 cu ft. At concentrations higher than 20 per cent by volume in air the odors of the Freon vapors are very mild and somewhat ethereal, being similar to carbon tetrachloride, the commonly used fire-extinguishing fluid. The Freon vapors in all concentrations are nonirritating to the eyes, nose, throat, lungs, or skin. Thus, gas masks or refrigerant-absorbing materials are not required when handling any of these refrigerants.

When a toxic or flammable refrigerant is odorless or practically so, some suitable substance should be added which will not corrode the equipment but which will permanently maintain a distinctive odor. In such a refrigerating system, especially in public places, care should be taken in the addition of odorants as a small leak in the system might cause a panic that would do far more harm than the refrigerant itself. At the present time there are no irritants or odorants or warning agents that will satisfactorily serve such a purpose.

Warning agents are not required and are not even considered desirable for use in the odorless and nonirritating Freon-charged refrigeration equipment.

Toxicity. A refrigerant liquid or vapor must be nontoxic and nonpoisonous to eliminate all possibilities of a health hazard should the refrigerant escape from the system. Toxicity as referred to must not be confused with concentrations of refrigerant-air mixtures high enough to exclude sufficient oxygen to support life.

The Underwriters' Laboratories have found that $\frac{1}{4}$ lb of ammonia per 1,000 cu ft of space will kill or seriously injure with an exposure of $\frac{1}{2}$ hr.

Sulfur dioxide will cause throat irritation, coughing, constriction of the chest, nausea, irritation of the eyes, and is so acutely pungent as to cause a sensation of suffocation even with the first breath. The Underwriters' Laboratories have found that 18 to 19 oz per 1,000 cu ft will kill or seriously injure in 5 min.

Methyl chloride is an anesthetic when concentrated in closed spaces or on long exposure, and there is a definitely delayed toxic action. The Underwriters' Laboratories have found that from 2½ to 3¼ lb per 1,000 cu ft will kill or seriously injure in 2 hr.

It is a well-established fact that the Freon refrigerants as such are the least toxic refrigerants that have ever been discovered, and complete and detailed data on toxicity will be found in the Underwriters' Laboratories reports and Bureau of Mines reports.

The toxic properties of various refrigerants are shown in Table 9-1.

Table 9-1. Toxic Properties of Refrigerants

Underwriters' Laboratories group No.	Refrigerant	Formula	Cu ft per lb of refrigerant, 68°F, 1 atmosphere	Boiling point, deg F	Kills or seriously injures		
					Duration of exposure	Concentration in air	
						Per cent by volume	Lb per 1,000 cu ft
1	Sulfur dioxide	SO_2	6.01	14.0	5 min	0.7	1.165
2	Ammonia	NH_3	22.6	−28.0	½ hr	0.5–0.6	0.221–0.256
3	Methyl formate	$C_2H_4O_2$	6.41	89.2	1 hr	2–2.5	3.12–3.9
4	Methyl chloride	CH_3Cl	7.62	−10.6	2 hr	2–2.5	2.62–3.28
4	Dichloroethylene	$C_2H_2Cl_2$	3.97	118	2 hr	2–2.5	5.04–6.3
a4	Ethyl chloride	C_2H_5Cl	5.96	54.5	1 hr	4.0	6.72
a4	Methylene chloride	CH_2Cl_2	4.53	103.6	½ hr	5.1–5.3	11.25–11.70
b4	Freon 113	$C_2Cl_3F_3$	2.1	117.6	½ hr	4.8–5.2	23.2–25.2
b4	Freon 21	$CHCl_2F$	3.76	48.0	½ hr	10.2	27.1
5	Carbon dioxide	CO_2	8.75	108.4	½–1 hr	29.0–30.0	33.2–34.3
5	Freon 11	CCl_3F	2.8	74.7	2 hr	10	35.7
5	Freon 22	$CHClF_2$	4.37	−41.4	2 hr	18–22.6	40.2–50.5
6	Freon 114	$C_2Cl_2F_4$	2.25	38.4	2 hr	20.1–21.5*	89.6–95.7
6	Freon 12	CCl_2F_2	3.18	−21.7	2 hr	28.5–30.4*	89.6–95.7

* Oxygen deficiency.

Flammability. A refrigerant must be nonflammable and nonexplosive in any concentration in order to eliminate all possibility of a fire or explosion hazard. The fact has been recognized that even a small amount of flammable or explosive vapor may be concentrated on the floor or in a confined space around a leak which may produce an explosive mixture, even though the small amount of refrigerant when evenly diffused to all parts of the room may not produce an explosive mixture. Thus all flammable and combustible refrigerants should be considered fire hazards regardless of the concentrations required or the amount of refrigerant permitted to escape from the system.

In the Underwriters' Laboratories reports and also the Bureau of Mines reports, it will be found that all the Freon refrigerants, sulfur dioxide, carbon dioxide, and methylene chloride have been classified as noncombustible and nonflammable. Formation of combustible mixtures by Freon 21 and Freon 113 under practical conditions, even at the higher temperatures, is extremely unlikely, and their fire hazards are, therefore, very small, as is the case with methylene chloride.

Explosive properties of the commonly used refrigerants are shown in Table 9-2.

Corrosion. A refrigerant must be noncorrosive to all metals or materials used in refrigeration construction in order to permit the refrigeration manufacturer a wide selection of materials with which to design efficient condensers, evaporators, bearings, pipe lines, control apparatus, etc.

Certain refrigerants are mildly corrosive to certain metals but actively corrosive to other metals. This calls for extreme care in the selection of iron, steel, brass, galvanized metal, acid-resisting bronze, monel metal, nickel, high-nickel alloys, and aluminum or for the trimming of iron valves with brass alloy, steel, or aluminum.

The Freon refrigerants are noncorrosive to all metals used in refrigeration apparatus, including steel, cast iron, brass, copper, tin, lead, zinc, silver, aluminum, etc.

Any type of system, regardless of the refrigerant used, should be dry, especially in the case of those refrigerants in which water is not soluble in order to eliminate the possibility of the moisture freezing out at the regulating valve, intercrystalline embrittlement of brass bellows or diaphragms under cyclic stress, shorting of field coils in hermetically sealed units, and the emulsification of the lubricating oil, any one of which might cause faulty operation of the system. It is to be pointed out that water which may be in a system together with air will cause oxidation of metal parts which must be distinguished as oxidation or rusting and is not due to corrosion by the refrigerant.

Table 9-2. Explosive Properties of Refrigerants

Refrigerant	Explosive range, concentration in air		Maximum explosion pressure, psig	Time of development of pressure, sec
	Per cent by volume	Lb per 1,000 cu ft		
Ammonia	16–25	7.1–11.05	50	0.175
Methyl formate	4.5–20.0	7.02–31.2	96	0.026
Methyl chloride	8.1–17.2	10.6–22.6	69	0.110
Dichlorethylene	5.6–11.4	14.1–28.7	76	0.095
Ethyl chloride	3.7–12.0	6.12–20.1	87	0.028
Sulfur dioxide		Nonflammable		
Carbon dioxide		Nonflammable		
Methylene chloride		Nonflammable		
Freon 11		Nonflammable		
Freon 12		Nonflammable		
Freon 21		Nonflammable		
Freon 22		Nonflammable		
Freon 113		Nonflammable		
Freon 114		Nonflammable		

Stability. A refrigerant must be stable, and over long years of service it has been found that the Freon refrigerants will withstand repeated evaporations, compressions, and condensations without dissociation, and these refrigerants do not form high-boiling or solid compounds.

Certain refrigerants such as ammonia show evidence of some disintegration in both compression and absorption systems due to reaction of moisture with metals, due to disassociation, and due to unsaturated hydrocarbons in the mineral lubricating oil which necessitates constant purging of the foul gases, which are constantly recreated, in order to obtain economic operation. Foul gases are dangerous in many systems when building up test pressures with air in case some refrigerant is present because certain of the disassociation products can be readily detonated when present in the proper mixtures. Disassociation of a refrigerant in a system will also cause high condensing temperature, high current consumption or high cost of operation, and noncondensable vapor lock in h-p float-valve chambers.

Inertness. A refrigerant must be inert with respect to compounded materials such as stuffing-box and valve-stem packing, gasket materials, cotton-covered windings, insulating varnishes, etc.

Most of the refrigerants have solvent action on many materials and composition gaskets with the result that such materials must be carefully tested and selected before use. In general, such gasket materials should not contain natural rubber, but instead neoprene should be obtained from the rubber companies which are in position to supply materials of the proper characteristics. Pressed fibers, including asbestos, with a number of insoluble binders may be used as may also metallic gaskets. Cotton on motor windings must be properly cleaned and degummed, and insulators and varnishes must be selected with proper regard to the solvent properties of the particular refrigerant.

Effect on Materials. A refrigerant must have no effect on any materials being refrigerated, such as food, flowers, furs, or fabrics, or any materials that may be exposed to the refrigerant vapor or liquid.

Certain refrigerants such as ammonia and sulfur dioxide cause mild to serious contamination to plant life, food products, textiles, or other materials with which they may come in contact, and excessive exposure of fruits, vegetables, meats, and liquids will cause rot, scald, and burns or make them so unpalatable as to prevent their consumption.

The Freon refrigerants, either liquid or vapor, are not absorbed by and have no effect on any materials being refrigerated, and the vapor has no effect on the odor, taste, color, or structure of dairy products, meats, or vegetables. The Freon vapors have no effect upon the odor, color, continued blooming, or structure of flowers or plant life, and the liquid or vapor has no effect upon the color or structure of furs or fabrics.

Leak Detection. A leak of refrigerant must be capable of being easily detected and located.

Leaks of ammonia, aside from odor, are detected and located by ammonium sulfite (white fog or fumes) resulting from burning sulfur sticks in the presence of the refrigerant or determined by the alkaline reaction on moistened litmus paper or phenolthalein.

Leaks of sulfur dioxide, aside from odor, are detected and located by ammonium sulfite resulting from exposure to 28 per cent ammonia in water in the presence of the refrigerant.

Many odorless and/or flammable refrigerants, such as methyl chloride, methylene chloride, carbon dioxide, isobutane, ammonia, and methyl formate, are not easily detected or located, owing to their relatively high chemical stability, and other methods of leak detection must be resorted to, such as the application of soap and water solution at all suspected points or observation of oil spots or leaks. It is quite obvious that such methods are not entirely satisfactory in dark, confined, or inaccessible spaces or locations.

Owing to the fact that the Freon refrigerants are stable and inert, the usual visual method called the "smoke" test cannot be used to determine leaks as there is no known chemical compound that will react with the refrigerants to produce a visible vapor.

The best known method of detecting and locating a leak of Freon vapor is by means of an alcohol or gas-burning halide torch. Owing to the combustion of the fuel in the burner, a draft of from 2 to 3 fps is created through a steel tube which is inserted in the burner at the base of the flame. A flexible rubber hose is placed on the end of the steel tube, and the open end of the rubber exploring hose is then passed around all possible sources of leak. Any air suspected of containing Freon vapor is conducted through the hose and tube into the burner and over metallic copper. Should there be any Freon vapor in the air being tested, the high-temperature flame will cause the refrigerant vapor to break down and form a volatile copper halide. This will cause the flame color to change from the normal colorless to a bright green if the air contains Freon vapor.

A leak of Freon vapor may be definitely located by means of a halide torch or lamp even when the leak occurs in a dark or confined space. Leaks at the rate of 1 oz of Freon per month (0.06 per cent) may be readily detected, although these detector torches will detect leaks as low as 0.01 per cent, which is equivalent to the escape of Freon at the rate of 1 oz in 6 months. Leak-detector torches of nonferrous metal burners or of a pump-up type are not recommended, as the color change of the flame will not be sufficiently rapid and fully serve the purpose of the lamp. The use of odorants or chemicals to produce reactions for a smoke test are not recommended. Leak-detector lamps of the type now on the market should not be used for locating leaks of flammable or combustible gases used as refrigerants.

The rate of leakage of a refrigerant gas is dependent on such factors as pressure, capillarity, viscosity, density, rate of diffusion, and molecular weight, but a very good measure of this rate of leak may be expressed as being inversely proportional to the square root of molecular weight, assuming the pressure to be the same in all cases.

Thus for the same pressure it will be found in Table 9-3 that the Freon vapor or gas leakage will be less than the leakage of any of the other compressed gases used as refrigerants.

Table 9-3. Rate of Leak

Refrigerant	Molecular weight	Square root of molecular weight	Rate of leak (same pressure)
Freon 113	187.39	13.69	0.68
Freon 114	170.93	13.00	0.71
Freon 11	137.4	11.63	0.80
Freon 12	120.9	11.0	0.85
Freon 21	102.93	10.16	0.92
Freon 22	86.48	9.3	1.00
Methylene chloride	84.92	9.21	1.01
Sulfur dioxide	64.07	8.0	1.16
Methyl formate	60.03	7.75	1.20
Isobutane	58.8	7.63	1.22
Methyl chloride	50.48	7.11	1.31
Carbon dioxide	44.0	6.64	1.40
Air	29.0	5.38	1.73
Ammonia	17.03	4.13	2.25

Surface-wetting Ability. A refrigerant must have good surface-wetting characteristics so as to release refrigerant vapor or gas bubbles or vapor film from the heat-transfer surfaces of the evaporator and condenser to make such surfaces more effective and efficient.

The various refrigerants have different characteristics or properties for wetting heat-transfer surface as the gas bubbles of one type spread out on the unwetted surface, thereby drawing the free edge of the bubble into a fine wedge between the refrigerant liquid and the heat-absorbing surface and holding it fast against the action of buoyancy. In such a case the bubble becomes rather large before being detached and replaced by a new bubble.

In the case of gas bubbles in an evaporator using a surface-wetting refrigerant, the liquid refrigerant tends to push away and to sheer off the bubble, which rests at only one point and becomes globular or oval and leaves the surface while still very small. Very complete information on the subject appears in various engineering journals and technical papers.

The chlorinated hydrocarbon refrigerants and the halo-fluoro Freon refrigerants are excellent in this respect. They have very good surface-wetting characteristics, release gas bubbles immediately and uniformly when formed, and eliminate the use of an ebullating surface, which places the liquid refrigerant directly in contact with heat-transfer surface which increases the heat-transfer efficiency and prevents superheating and violent blasting out of the gas through the liquid refrigerant in the evaporator.

Vapor Density. Low vapor density is at times considered desirable by certain engineers in order to obtain high gas velocity through valves, fittings, and small suction and discharge lines without causing excessive pressure drop.

Regardless of the refrigerant used, consideration must be given to the elimination or reduction of restrictions in suction and discharge lines, compressor valves, and fittings in order to permit free passage of refrigerant vapor. This consideration has made necessary the redesign of compressor intake and discharge valves with close study of pressure-volume diagrams, with the result that the present-day systems have shown greatly increased operating efficiency and reduced valve noise without excessively large or costly installation and without engineering difficulties.

Densities of refrigerant vapor in pounds per cubic feet at saturated pressures and standard ton temperature conditions are shown in Table 9-4.

Table 9-4. Densities of Refrigerant Vapors

Refrigerant	Vapor density lb per cu ft	
	5°F	86°F
Carbon dioxide	3.741	21.09
Freon 22	0.8034	3.213
Freon 12	0.6735	2.569
Freon 114	0.2369	1.159
Methyl chloride	0.2209	0.930
Isobutane	0.156	0.658
Sulfur dioxide	0.1558	0.844
Ammonia	0.1227	0.564
Freon 21	0.1095	0.577
Freon 11	0.0815	0.446
Freon 113	0.0370	0.257
Methyl formate	0.0212	0.141
Methylene chloride	0.202	0.15

Table 9-5. Density of Liquid Refrigerants

Refrigerant	Density lb per cu ft 86°F	Specific gravity (water = 1)
Freon 113	96.96	1.559
Freon 11	91.38	1.468
Freon 114	89.91	1.443
Freon 21	84.52	1.360
Sulfur dioxide	84.4	1.358
Methylene chloride	83.3	1.340
Freon 12	80.63	1.297
Freon 22	73.36	1.177
Methyl formate	61.0	0.982
Methyl chloride	55.8	0.898
Carbon dioxide	37.4	0.602
Ammonia	37.2	0.598
Isobutane	34.1	0.549

Liquid Density. Density or gravity of liquid refrigerant concerns only the calculations for buoyancy of float valve, but when calculations are made consideration must be given to the density or gravity of the refrigerant-oil mixtures rather than only the liquid refrigerant.

Density or gravity is of minor importance when calculating pressure drop in liquid lines as the liquid velocities are of a very low order. Density of liquid has influence on the volume of liquid refrigerant that may be required in the system but density alone is of secondary importance when selecting the ideal refrigerant. Refrigerants of high density or gravity are now being used quite widely in preference to refrigerants of low density and gravity.

The density of liquid refrigerants in pounds per cubic foot is shown in Table 9-5.

Critical Temperature and Pressure. The refrigerant must have a high critical temperature to permit high operating efficiency when using either high-temperature water or air as a condensing medium. The critical temperature is the temperature above which the refrigerant can exist only as a gas, and the pressures at those tem-

Table 9-6. Critical Temperatures and Pressures

Refrigerant	Critical temperature deg F	Critical pressure psia
Methylene chloride	421	640
Methyl formate	418	607
Freon 113	417.4	495
Freon 11	388.4	635
Freon 21	353.3	750
Sulfur dioxide	314.8	1,141.5
Freon 114	294.3	474
Methyl chloride	289.6	969.2
Isobutane	272.7	537.1
Ammonia	271.2	1,651
Freon 12	232.7	582
Freon 22	204.8	716
Carbon dioxide	87.8	1,071

Table 9-7. Freezing Points

Refrigerant	*Freezing point deg F*
Freon 22	256
Freon 12	252.4
Isobutane	229
Freon 21	211
Freon 11	168
Methyl formate	148
Methyl chloride	143.7
Methylene chloride	143
Freon 114	137
Ammonia	107.9
Sulfur dioxide	98.9
Carbon dioxide	-69.9
Freon 113	-31.0

peratures are shown in Table 9-6. It will be observed that, with the exception of carbon dioxide, all refrigerants have critical temperatures well above condensing-medium temperatures that will be experienced in refrigeration practice.

Freezing Point. The refrigerant must have a low freezing point which should be well below any temperature that may be encountered in machine operation. Table 9-7 shows the freezing points of the various refrigerants, all of which are lower than any temperature upon which they are called upon to produce under normal operating conditions.

REFRIGERANT SYSTEM LUBRICATION

A refrigerant must be miscible with mineral lubricating oils in order to simplify the lubrication problem and permit the most efficient lubrication of the compressor under all operating conditions.

Practically all refrigerants are more or less miscible with mineral lubricating oil, which reduces the viscosity of the oil, but the miscibility is dependent on temperature and pressure of the solution, base of the oil, viscosity of the oil, and design of the compressor and lubricating system. Certain refrigerants are miscible with oils to a very slight degree, while other refrigerants are completely miscible when under pressure and low temperature conditions. This question of miscibility should not be considered a serious determining factor as refrigerants that are highly miscible, such as methyl chloride, ethyl chloride, isobutane, methylene chloride, Freon 12, Freon 11, Freon 21, Freon 22, Freon 113, and Freon 114, have been used with complete satisfaction. Refrigerants that are considered relatively immiscible with mineral lubricating oil are ammonia, carbon dioxide, sulfur dioxide, and methyl formate, although the question of immiscibility was not the reason that prompted their use.

Many refrigerants are not soluble or are soluble only to a limited degree in mineral lubricating oil which will permit the heavier fractions of the oil, wax, or gum to separate or deposit on the cold heat-transfer surface of the evaporator or condenser, with the result that they will greatly reduce heat transfer or prevent return of all the oil to the compressor where it is needed.

The chlorinated hydrocarbon refrigerants and the halo-fluoro Freon refrigerants which dissolve in the mineral lubricating oil will automatically remove the heavy oil, wax, or gum with the result that heat-transfer efficiency is greatly improved.

All chlorine-bearing compounds greatly assist in the lubrication of refrigeration compressors because they act as a carrier for the oil causing wetting of the bearing surface. The Freon refrigerants do not react with any of the mineral lubricating oils used in refrigeration apparatus.

The advantages of using a refrigerant which is miscible with lubricating oil are as follows:

1. The oil foam or fog will permit return of the oil to the compressor from the evaporator.
2. Positive return of oil from evaporator.
3. Boiling point of refrigerant-oil mixture changed but very little even with large quantities of oil present.
4. Refrigerant a good carrier of oil to moving parts ordinarily difficult to reach.
5. Simplified construction of oil return.
6. No problem of oil separation and return from liquid receivers, evaporators, or high-side float chambers.
7. Good heat-transfer characteristics due to eliminating insulating film of oil.
8. Solubility lowers pour point of the oil in low-temperature evaporators

Considerable data have been developed which show the solubility of the Freon refrigerants in mineral lubricating oil at various pressures and temperatures. Other data have been developed which show the boiling point of the Freon refrigerants and oil solutions and also the viscosity of Freon oil solutions at various temperatures.

The various curves show a reduction in the viscosity of the oil when diluted with the Freon refrigerants, especially at low oil temperature, but relatively small change in viscosity at the higher oil temperatures which will exist in the bearing under operating conditions. It is not believed that one should be directly concerned with the solubility of Freon in oil, as the curves show that the addition of 5, 10, or possibly 15 per cent of Freon in oil under actual operating conditions does not greatly reduce the viscosity. In other words, the reduction in oil viscosity when using Freon at temperatures of from 160 to 180°F is not greatly changed from that of the oil itself.

The experience of the refrigeration and air-conditioning industries has shown that Freon refrigerants do not cause injury to the lubricating properties. It is true that bearings may be scored at times, but invariably it will be found that the damage to the bearings had been caused by the presence of factory dirt, oxide, or scale, foundry sand, or other foreign matter which had been removed from the casting or pipe owing to the cleansing properties of Freon refrigerants.

TYPE OF COMPRESSOR

The evaporator temperature to be obtained and the type and size of compressor to be used will determine the selection of the refrigerant to be employed.

Under normal temperature conditions, it will be found that refrigerants with boiling points of less than approximately 15°F are adaptable to reciprocating-type compressors. Refrigerants with boiling points of between approximately 10 and 90°F are adaptable to rotary-type compressors of fractional-horsepower size and of hermetically sealed design. Refrigerants with boiling points higher than 35°F are adaptable to centrifugal-type compressors of larger than 50 tons' capacity.

Low-boiling-temperature refrigerants have higher pressures and can best be used in low-piston-displacement reciprocating-type compressors operating under positive pressures on the low side of the system and with pressure differentials of approximately $3\frac{1}{2}$ atmospheres (50 psi) and higher.

Medium-boiling-temperature refrigerants are used in medium-vapor-displacement rotary-type compressors operating at or near atmospheric pressure, such as 2 lb abs on the low side of the system and with pressure differentials of $1\frac{1}{2}$ to $3\frac{1}{2}$ atmospheres (20 to 50 psi).

High-boiling-temperature refrigerants are used only in large-vapor-displacement high-speed centrifugal-type compressors operated under negative pressures of as low as 1 lb abs on the low side of the system and with pressure differentials of up to $1\frac{1}{2}$ atmospheres (22 psi).

Thus under normal temperature conditions found in the average refrigerating or air-conditioning system, it will be found that reciprocating-type compressors require the use of carbon dioxide, ammonia, Freon 22, Freon 12, methyl chloride, isobutane, and sulfur dioxide. Fractional-horsepower rotary-type compressors require the use of isobutane, sulfur dioxide, Freon 114, Freon 21, Freon 11, and methyl formate. Centrifugal-type compressors require the use of Freon 11, methylene chloride, and Freon 113.

Any refrigerant can be used in any type of compressor provided that proper cognizance is given to evaporator and condenser temperatures, pressures and pressure differentials, density and volume of refrigerant vapor, and other factors. For example, Freon 12 or Freon 22, both of which have high vapor density, may be used in a centrifugal-type compressor for producing extremely low temperatures (−150°F) provided that the difference in pressure between the high and the low sides of the system is held within $1\frac{1}{2}$ atmospheres, or approximately twenty-two pounds per square inch. Condensing of the Freon 12 or Freon 22 will need to be taken care of by means of a second centrifugal-type compressor using Freon 11 or Freon 114. This would be known as a cascade type of system. Freon 12 or Freon 22 may be used in reciprocating-type compressors for producing very low temperatures (−100 to −150°F) provided a two- or three-stage compressor is employed to hold the compression ratios between stages as low as possible.

Isobutane, sulfur dioxide, and methyl formate are limited to fractional-horsepower reciprocating- or rotary-type compressors only. Such systems were used only in household units but have long since been discontinued

Refrigerant Heat Values. The heat values of refrigerant liquid and vapor in refrigerating effect per pound must be considered with the liquid refrigerant volume in order to establish a value for the refrigerant itself. A refrigerant must not be judged only by the fact that it has a high or low refrigerating effect per pound, but the volume per pound of the liquid refrigerant must also be taken into account in order to arrive at the volume of liquid refrigerant to be vaporized per unit of time and a given load.

The Freon refrigerants have relatively low refrigerating effects per pound, but this must not be considered a disadvantage as it merely indicates that more liquid refrigerant must be metered to the evaporator in order that the desired amount of refrigeration be produced. Actually it is a decided advantage to circulate a large volume of liquid refrigerant through the expansion or float valve as the large volume of liquid refrigerant required will simply call for a large orifice diameter, which will permit the use of less sensitive, more accurate, and more positive operating and regulating devices with less critical adjustment.

Refrigerants with high heat values or refrigerating effects are not desirable, especially for use in the smaller size of systems owing to the small amount of liquid refrigerant in the system and the difficulty encountered in accurately controlling the flow of such small volumes of liquid refrigerant through the valve orifice that is required. For small systems, the adjustment of the regulating-valve orifice is most delicate when using refrigerants with high heat values.

The heat values, weight, and volume of liquid refrigerant to be metered through the regulating valve under standard ton conditions are shown in Table 9-8.

Table 9-8

	Heat content, Btu per lb		Btu refrigerating effect per lb	Lb liquid refrigerant evaporated per min	Cu ft liquid refrigerant per lb, 86°F	Cu in. liquid refrigerant to be evaporated per min
	Vapor, 5°F	Liquid, 86°F				
Ammonia	613.35	138.9	474.45	0.4215	0.02691	19.6
Sulfur dioxide	183.49	42.12	141.37	1.414	0.01184	28.9
Methyl formate	233.55	44.32	189.23	1.056	0.0164	29.9
Methylene chloride	163.8	29.75	134.05	1.492	0.01198	30.9
Methyl chloride	196.92	46.67	150.25	1.331	0.01778	40.9
Freon 21	119.97	30.56	89.41	2.237	0.01183	45.73
Freon 11	92.88	25.34	67.54	2.961	0.01094	55.976
Freon 113	79.60	25.93	53.67	3.726	0.01031	66.48
Freon 22	105.56	36.28	69.28	2.887	0.01363	67.97
Freon 12	78.79	27.72	51.07	3.916	0.0124	83.9
Freon 114	72.21	29.11	43.10	4.640	0.01112	89.16
Isobutane	162.0	50.5	111.5	1.794	0.0294	91.1
Carbon dioxide	102.14	45.45	56.69	3.528	0.0267	162.8

Compressor Vapor Displacement. A refrigerant should have such vapor volume per pound that it will permit low compressor displacement and make possible small, compact compressors of relatively low-speed reciprocating type or of high-speed centrifugal type.

Under standard ton conditions, the displacement of the compressor will be dependent upon the refrigerating effect per pound, the weight of liquid refrigerant to be evaporated, as given in Table 9-9, and the volume per pound of refrigerant vapor.

The vapor displacement required by the compressor is almost directly related to the boiling point of the refrigerant at atmospheric pressure as shown in Table 9-9. In other words, the lower the boiling point of the refrigerant, the lower will be the displacement required, or rather the higher the boiling point of the refrigerant, the greater will be the vapor displacement required.

Pressure. The condensing and evaporator pressures should be positive and as close as possible to atmospheric pressure to permit lightweight construction, low initial cost, lower power consumption, and compactness of the compressor unit, as well as to reduce the leakage of the refrigerant to a minimum, to assist in detecting and locating leaks, and to avoid the leakage of moisture-laden air into the system.

Table 9-9

Refrigerant	Boiling point, deg F	Lb liquid refrigerant evaporated per min	Vapor volume per lb, 5°F	Cu ft displacement per min
Carbon dioxide	−108.4	3.528	0.2673	0.943
Freon 22	−41.4	2.887	1.246	3.596
Ammonia	−28.0	0.4215	8.150	3.436
Freon 12	−21.6	3.916	1.485	5.815
Methyl chloride	−10.76	1.331	4.471	5.95
Isobutane	13.6	1.794	6.41	11.497
Sulfur dioxide	14.0	1.414	6.421	9.084
Freon 114	38.4	4.640	4.221	19.587
Freon 21	48.0	2.237	9.132	20.427
Freon 11	74.7	2.961	12.27	36.33
Methyl formate	89.2	1.056	47.25	49.9
Methylene chloride	103.7	1.492	49.9	74.45
Freon 113	117.6	3.726	27.04	100.76

In Table 9-10 will be found the pressures of the various refrigerants and compression ratios when under standard ton conditions

Operating head pressures will vary with the temperature of condensing medium, type of condenser used, whether air- or water-cooled, design and condition of the condenser, operating back pressure, superheat from the compressor, and other factors Such pressures that may be observed will correspond to a temperature of approximately 10 to 15°F higher than the corresponding temperature of condensing medium for a good design of water-cooled condenser and as much as possibly 25° higher than the condensing air temperature for small air-cooled condensers. The temperature will also depend on the degree of air circulation

Table 9-10

Refrigerant	Gauge pressure psi		Compression ratio
	86°F	5°F	
Carbon dioxide	1,024.3	319.7	3.11
Freon 22	159.8	28.33	4.05
Ammonia	154.5	19.57	4.94
Freon 12	93.2	11.81	4.07
Methyl chloride	80.83	6.19	4.57
Sulfur dioxide	51.75	5.87*	5.63
Isobutane	44.8	3.3*	4.55
Freon 114	21.99	16.14*	5.42
Freon 21	16.53	19.25*	5.98
Freon 11	3.58	23.95*	6.20
Methyl formate	1.84*	26.34*	7.84
Methylene chloride	9.44*	27.53*	8.57
Freon 113	13.93*	27.92*	8.01

* Inches of mercury below atmospheric pressure.

REFRIGERANT DATA

Superheat. Complete superheat data are not available for many of the refrigerants as superheat values has never been developed for them Thus all comparisons cannot be made for performance in high-temperature locations High superheat values mean high temperatures, carbonization of oil, damage to discharge values, and large condenser surface for the removal of superheat. It is interesting to note in Table 9-11 that the temperatures of compression for the Freon refrigerants are quite low under standard ton conditions of 5°F evaporator, 86°F condenser, 9°F superheat of vapor, and 9°F subcooling of liquid.

Horsepower. The theoretical indicated horsepower per ton of refrigeration is substantially the same for all refrigerants with the exception of carbon dioxide under a given set of temperature operating conditions.

Engineering design and many times sales, installations, service, and operation are responsible for the wide variation in actual horsepower per ton required. It has been found from practice that, regardless of the C_p, C_v, C_p/C_v ratio, ratio of differential heat of liquid to latent heat, specific heat of liquid, coefficient of performance, power loss due to liquid expansion and compression, superheat, etc , Freon-charged systems operate with less brake horsepower than any other of the commonly used refrigerants when using compressors of a type designed to handle the particular refrigerant. The use of heat exchangers for the Freon refrigerants accounts for quite noticeable gains in efficiency.

Cost. The selection of a refrigerant for all types of installations must be based upon its over-all utility rather than influenced by its cost alone. The cost of a refrigerant loses its so-called importance when the particular refrigerant is nonhazardous under all habitable conditions, where it permits simplicity of construction, operates quietly, is lightweight and compact, and allows use of a wide variety of metals and materials of only moderate construction, and in addition, eliminates secondary brine systems, makes possible high operating efficiency, makes obtainable greater flexibility of application and automatic operation, and lends itself to perfect lubrication and greater reliability of automatic controls and expansion valves, and, most important of all, permits low initial and ultimate cost of complete installation over a long period of satisfactory and adequate refrigeration service, whether it be for household, commercial, industrial, or comfort-cooling air-conditioning and refrigeration purposes.

Not only do the Freon refrigerants make application of refrigerating systems safe and efficient, but in the smaller sizes they have permitted a production and installation cost of practically half that of systems of similar capacity charged with refrigerants that have been used in the past.

Table 9-11

Refrigerant	*Temperature of compression*
Freon 114	90°F*
Freon 113	93
Freon 12	109.5
Freon 11	121
Freon 22	139
Freon 21	150
Methyl chloride	151
Sulfur dioxide	204
Ammonia	222.6
Carbon dioxide	—†
Methylene chloride	—
Methyl formate	—
Isobutane	—

* 15°F superheat of vapor.

† Not operated under standard ton conditions.

Use of Warning Agents. The unanimous opinion of chemists and manufacturers who have worked on this problem is that, although several substances are very good warning agents in some respects, each one has some very objectionable properties. There appears to be no warning agent that will satisfactorily serve such a purpose. Therefore, use of irritants or odorants is not advised in connection with Freon-charged apparatus. Various reasons such as insufficient warning power, toxicity, instability, corrosiveness, and low volatility have served to eliminate from consideration the use of warning agents with the Freon refrigerants.

Use of Antifreeze. We do not believe it advisable to add any liquid or solid compound to any refrigerating system with the hope of eliminating moisture troubles or destroying any water that might be present because in many cases very objectionable and corrosive compounds will be formed, especially when such compounds are improperly used and their properties not fully understood. Instead of adding such a compound to a system we believe it to be much more advisable to remove the water from the system and eliminate the cause of the difficulty rather than to add such a material as a "crutch" or "cure-all," which certainly will not improve the refrigerating effect of the system.

It should be the proper practice to remove the moisture and other impurities from the refrigerating system and the lubricating oil rather than to add any new compounds which of themselves may contain water or other impurities. In other words, it is believed that a cure should be effected by removing the moisture from the system rather than merely to seek temporary relief by adding impurities to the system.

It is an indisputable fact that a system should be thoroughly dried to prevent all possibilities of oxidation and corrosion ever occurring or the freezing out of the moisture at the regulating valve. The recommended drying agents to be used for the removal of water from a Freon-charged system are silica gel, activated alumina, or calcium sulfate.

HANDLING REFRIGERANTS

One of the important requirements of a refrigerant is that it be handled with complete safety, not only by the owner or operator of the refrigerating system but also by the installation and service men.

It is not required that the Freon and other of the nontoxic and inert refrigerants be discharged into a water, acid, or an alkali solution when disassembling or servicing an installation. Such refrigerants may be discharged into the atmosphere as no damage will be caused to any materials being stored in the refrigerator or any injury or damage to flowers, plants, shrubbery, or family pets. Ammonia should be discharged into water, and sulfur dioxide should be discharged into an alkali solution in order that such refrigerants be absorbed or neutralized.

Freon 12, Freon 22, methyl chloride, carbon dioxide, and isobutane should be discharged slowly so that the liquid refrigerant is permitted to vaporize before it comes in contact with any materials so as to eliminate the possibility of freezing. Of course, any of these refrigerants may be pumped down or discharged into clean cylinders for reuse.

Inasmuch as the Freon refrigerants are nonirritating and nontoxic, it is not necessary to wear a gas mask when servicing such equipment. However, it is essential that proper protection be afforded the eyes by the use of goggles or large-lensed spectacles to eliminate the possibility of low-boiling-point liquid refrigerant coming in contact with the eyes and possible damage due to the freezing of the moisture in the eye. Should the low-boiling-point refrigerant come in contact with the eyes, the person suffering the injury should be taken at once to a good eye specialist. Rubbing or irritating the eyes should be avoided, and the following first-aid treatment should be given immediately:

1. Drops of sterile mineral oil should be introduced into the eyes as an irrigant.

2. The eyes should then be washed with one of the following if the irritation continues at all:

   a. A weak boric acid solution.

   b A sterile salt solution not to exceed 2 per cent sodium chloride.

Should the liquid Freon gas come in contact with the skin, the injury should be treated the same as though the skin had been frostbitten or frozen.

First aid for persons exposed to sulfur dioxide consists of taking the patient into fresh air and, if the case appears serious, calling a physician

The first aid for persons exposed to ammonia fumes is as follows:

Summon a doctor promptly. If patient is unconscious, keep up artificial respiration. If eyes are affected, hold the lids open and pour water or 2 per cent boric solution over the eyeballs and lids washing thoroughly for 5 min then allow two drops of liquid petrolatum to fall on the eyeball

If the skin is affected, strip ammonia-saturated clothing from the body. Wash with water, then paint burned surfaces with a saturated solution of picric acid (use boric acid near the eyes). Do not cover burns with clothing or dressings.

If the nose and throat are affected, have patient snuff boric acid solution up nose and rinse mouth thoroughly.

Encourage patient to drink large amounts of water. Ammonia gas is lighter than air, therefore, keep head as low as possible in presence of escaping ammonia. Keep near the floor in going to rescue of one overcome by ammonia, and keep a wet sponge or cloth over the mouth and nostrils Keep the following on hand in clean, easily accessible containers for first-aid purposes:

1. One-gallon bottle 2 per cent boric solution
2. One-quart bottle saturated aqueous solution of picric acid
3. One-pound sealed package of absorbent cotton
4. One ounce liquid petrolatum and medicine dropper

In cases of exposure to methyl chloride, the patient should be removed to fresh warm air and a doctor called. The following first-aid treatments may be used by properly trained persons:

1. Remove patient immediately from gas-contaminated air.
2. Administer artificial respiration if necessary.
3. Administer a mixture of 93 per cent oxygen and 7 per cent carbon dioxide. Such a mixture is commercially available in cylinders. An H. H. inhalator may be used.
4. Give by mouth sodium bicarbonate in water.
5. Administer aromatic spirits of ammonia as a stimulant.
6. Subsequent treatment to be prescribed by attending physician.

PROPERTIES OF REFRIGERANTS

The physical and thermodynamic properties of saturated vapor and liquid refrigerants will be found in Tables 9-12 to 9-24.

Table 9-12. Properties of Freon 11

Temperature, deg F	Pressure, psig	Volume vapor, cu ft/lb	Density liquid, lb/cu ft	Heat content, Btu/lb		Temperature, deg F
				Liquid	Vapor	
−100	29 79*	453 1	105 5	−11 86	80 50	−100
− 95	29 76*	360 2	105 1	−10 87	81 07	− 95
− 90	29 72*	288 6	104 8	− 9 87	81 64	− 90
− 85	20 67*	232 8	104 4	− 8 88	82 21	− 85
− 80	29 60*	189 0	104.1	− 7 89	82 79	− 80
− 75	29 52*	154 5	103 7	− 6 90	83 37	− 75
− 70	29 43*	127 1	103 4	− 5 91	83 95	− 70
− 65	29 32*	105 1	103 0	− 4 92	84 54	− 65
− 60	29 19*	87 5	102 7	− 3 94	85 12	− 60
55	29 04*	73 23	102 3	− 2 95	85 71	− 55
50	28 86*	61 58	102 0	− 1 97	86 30	− 50
− 45	28 66*	52 06	101 6	− 1 00	86 89	− 45
− 40	28 42*	44 21	101 25	0 00	87 48	− 40
− 35	28 14*	37 75	100 88	0 99	88 08	− 35
30	27 81*	32 33	100 52	1 97	88 67	− 30
25	27 45*	27 87	100 14	2 95	89 27	− 25
− 20	27 03*	24 06	99 77	3 94	89 87	− 20
− 15	26 55*	20 89	99 40	4 93	90 47	− 15
− 10	26 01*	18 17	99 03	5 91	91 07	− 10
− 5	25 40*	15 90	98 64	6 90	91 67	− 5
0	24 72*	13 94	98 27	7 89	92 27	0
5†	23 95*	12 27	97 88	8 88	92 88	5†
10	23 10*	10 83	97 50	9 88	93 48	10
15	22 14*	9 60	97 11	10 88	94 09	15
20	21 08*	8 52	96 72	11 87	94 69	20
25	19 91*	7 58	96 33	12 88	95 30	25
30	18 61*	6 78	95 94	13 88	95 91	30
35	17 07*	6 06	95 54	14 88	96 51	35
40	15 61*	5 45	95 14	15 89	97 11	40
50	12 00*	4 42	94 34	17 92	98 32	50
60	7 73*	3 63	93 53	19 96	99 53	60
70	2 64*	2 99	92 71	22 02	100 73	70
75	0 10	2 73	92 30	23 45	101 33	75
80	1 61	2 49	91 88	24 09	101 93	80
86†	3 58	2 24	91 38	25 34	102 65	86†
90	4 99	2 09	91 04	26 18	103 12	90
95	6 88	1 92	90 61	27 22	103 71	95
100	8 90	1 77	90 19	28 27	104 30	100
105	11 08	1 63	89 76	29 33	104 89	105
110	13 39	1 50	89 34	30 40	105 47	110
115	15 86	1 38	88 90	31 46	106 06	115
120	18 50	1 28	88 47	32 53	106 63	120
125	21 49	1 18	88 03	33 60	107 21	125
130	24 29	1 10	87 60	34 67	107 78	130
135	27 45	1 02	87 16	35 76	108 35	135
140	30 80	0 951	86 69	36 84	108 91	140
150	38 15	0 824	85 76	39 02	110 02	150
160	46 34	0 718	84 85	41 23	111 12	160

* Inches of mercury below 1 atmosphere.

† Standard ton temperatures.

Table 9-13. Properties of Freon 12

Temperature, deg F	Pressure, psig	Volume vapor, cu ft/lb	Density liquid, lb/cu ft	Heat content, Btu/lb		Temperature, deg F
				Liquid	Vapor	
−155	29 68*	232 29	104 86	− 24 61	60 00	−155
−150	29 61*	179 79	104 46	−23 50	60 57	−150
−145	29 52*	140 52	104 05	−22 39	61 14	−145
−140	29 40*	110 92	103 64	−21 29	61 72	−140
−135	29 26*	88 34	103 22	−20 19	62 30	−135
−130	29 08*	70 94	102 80	−19 10	62 88	−130
−125	28 87*	57 42	102 38	− 18 02	63 46	−125
−120	28 61*	46 84	101 95	−16 94	64 04	−120
−115	28 31*	38 49	101 52	−15 85	64 63	−115
−110	27 94*	31 84	101 08	−14 78	65 22	−110
−105	27 51*	26 51	100 64	−13 71	65 81	−105
−100	27 01*	22 20	100 20	−12 64	66 40	−100
− 95	26 42*	18 71	99 75	−11 58	66 99	− 95
− 90	25 74*	15 86	99 30	−10 51	67 59	− 90
− 85	24 95*	13 51	98 85	− 9 46	68 18	− 85
− 80	24 05*	11 57	98 39	− 8 40	68 77	− 80
− 75	23 01*	9 96	97 92	− 7 35	69 36	− 75
− 70	21 84*	8 61	97 46	− 6 30	69 95	− 70
− 65	20 50*	7 47	96 99	− 5 25	70 54	− 65
− 60	19 00*	6 52	96 51	− 4 20	71 13	− 60
− 55	17 31*	5 70	96 04	− 3 15	71 72	− 55
− 50	15 42*	5 01	95 55	− 2 11	72 31	− 50
45	13 31*	4 42	95 07	− 1 06	72 91	− 45
− 40	10 92*	3 91	94 58	0 00	73 50	− 40
− 35	8 34*	3 47	94 10	1 01	74 10	− 35
− 30	5 45*	3 09	93 59	2 03	74 70	− 30
− 25	2 28*	2 76	93 08	3 05	75 30	− 25
− 20	0 58	2 47	92 58	4 07	75 87	− 20
− 15	2 46	2 22	92 07	5 10	76 46	− 15
− 10	4 50	2 00	91 57	6 14	77 05	− 10
− 5	6 74	1 81	91 03	7 20	77 64	− 5
0	9 17	1 64	90 52	8 25	78 21	0
5†	11 81	1 485	90 00	9 32	78 79	5†
10	14 65	1 35	89 45	10 39	79 36	10
15	17 74	1 26	88 92	11 48	79 94	15
20	21 05	1 21	88 37	12 55	80 49	20
25	25 63	1 03	87.80	13 66	81 06	25
30	28 16	0 939	87 24	14 76	81 61	30
35	32 57	0 863	86 67	15 87	82 16	35
40	36 98	0 792	86 10	17 00	82 71	40
50	46 69	0 673	84 94	19 27	83 78	50
60	57 71	0 575	83 78	21 57	84 82	60
70	70 12	0 493	82 60	23 90	85 82	70
75	76 90	0 457	82 00	25 08	86 32	75
80	84 06	0 425	81 39	26 28	86 80	80
86†	93 2	0 389	80 63	27 72	87 37	86†
90	99 6	0 368	80 11	28 70	87 74	90
95	108 0	0 342	79 46	29 93	88 19	95
100	116 9	0 319	78 80	31 16	88 62	100
105	126 2	0 293	78 14	32 40	89 03	105
110	136 0	0 277	77 46	33 65	89 43	110
115	146 3	0 258	76.70	34.90	89 00	115
120	157 1	0 240	76 02	36 16	90 15	120
125	168 4	0 223	75 25	37 41	90 46	125
130	180 2	0 208	74 46	38 69	90 76	130
135	192 7	0 194	73 64	39 95	91 01	135
140	205 5	0 180	72 73	41 24	91 24	140

* Inches of mercury below 1 atmosphere.
† Standard ton temperatures.

Table 9-14. ·Properties of Freon 21

Temperature, deg F	Pressure, psig	Volume vapor, cu ft/lb	Density liquid, lb/cu ft	Heat content, Btu/lb		Temperature, deg F
				Liquid	Vapor	
−40	27.16*	32.09	94.52	0.00	114.56	−40
−35	26.65*	27.87	94.15	1.18	115.16	−35
−30	26.08*	23.61	93.79	2.36	115.76	−30
−25	25.42*	20.39	93.42	3.54	116.36	−25
−20	24.67*	17.66	93.04	4.71	116.96	−20
−15	23.82*	15.38	92.66	5.90	117.57	−15
−10	22.87*	13.43	92.28	7.07	118.17	−10
− 5	21.79*	11.77	91.90	8.26	118.77	− 5
0	20.59*	10.35	91.52	9.44	119.37	0
5†	19.25*	9.132	91.13	10.63	119.97	5†
10	17.75*	8.09	90.74	11.81	120.57	10
15	16.08*	7.18	90.35	13.01	121.17	15
20	14.25*	6.39	89.96	14.21	121.78	20
25	12.22*	5.71	89.56	15.40	122.38	25
30	9.98*	5.11	89.16	16.61	122.98	30
35	7.51*	4.59	88.76	17.82	123.58	35
40	4.84	4.13	88.35	19.04	124.19	40
50	0.63	3.37	87.54	21.49	125.39	50
60	4.20	2.77	86.71	23.98	126.60	60
70	8.38	2.30	85.87	26.49	127.79	70
75	10.73	2.10	85.45	27.76	128.39	75
80	13.20	1.92	85.03	29.03	128.98	80
86†	16.53	1.73	84.52	30.56	129.88	86†
90	18.88	1.62	84.17	31.59	130.14	90
95	22.00	1.49	83.74	32.89	130.72	95
100	25.34	1.37	83.31	34.18	131.29	100
105	28.91	1.26	82.87	35.49	131.86	105
110	32.70	1.17	82.43	36.79	132.42	110
115	36.74	1.07	81.99	38.07	132.98	115
120	41.05	1.00	81.51	39.46	133.53	120
125	45.62	0.93	81.10	40.79	134.08	125
130	50.45	0.86	80.65	42.13	134.61	130
135	55.58	0.80	80.19	43.49	135.13	135
140	61.02	0.75	79.74	44.86	135.66	140
150	72.81	0.65	78.82	47.62	136.68	150
160	85.91	0.56	77.90	50.43	137.69	160

* Inches of mercury below 1 atmosphere.
† Standard ton temperatures.

Table 9-15. Properties of Freon 22

Temperature, deg F	Pressure, psig	Volume vapor, cu ft/lb	Density liquid, lb/cu ft	Heat content, Btu/lb		Temperature, deg F
				Liquid	Vapor	
−155	29 51*	188 1	97 67	−29 07	86 78	−155
−150	29 39*	146 1	97 33	−27 79	87 36	−150
−145	29 23*	114 5	96 99	−26 52	87 94	−145
−140	29 04*	90 61	96 63	−25 25	88 53	−140
−135	28 80*	72 33	96 27	−23 99	89 11	−135
−130	28 51*	58 21	95 91	−22 73	89 70	−130
−125	28 15*	47 23	95 53	−21 47	90 29	−125
−120	27 72*	38 60	95 15	−20 22	90 88	−120
−115	27 21*	31 77	94 76	−18 98	91 47	−115
−110	26 61*	26 33	94 37	−17 73	92 07	−110
−105	25 90*	21 96	93 97	−16 48	92 67	−105
−100	25 06*	18 43	93 56	−15 23	93 27	−100
− 95	24 09*	15 51	93 14	−13 98	93 87	− 95
− 90	22 96*	13 20	92 72	−12 73	94 47	− 90
− 85	21 67*	11 26	92 29	−11 47	95 08	− 85
− 80	20 18*	9 65	91 85	−10 22	95 68	− 80
− 75	18 51*	8 32	91 40	− 8 95	96 28	− 75
− 70	16 55*	7 19	90 95	− 7 69	96 88	− 70
− 65	14 40*	6 25	90 49	− 6 42	97 48	− 65
− 60	11 89*	5 45	90 03	− 5 16	98 08	− 60
− 55	9 01*	4 78	89 56	− 3 87	98 68	− 55
− 50	6 03*	4 19	89 08	− 2 58	99 28	− 50
− 45	2 61*	3 70	88 59	− 1 28	99 88	− 45
− 40	0 61	3 28	88 10	0 00	100 46	− 40
− 35	2 71	2 90	87 60	1 31	101 05	− 35
− 30	5 02	2 59	87 09	2 62	101 63	− 30
− 25	7 56	2 31	86 58	3 95	102 22	− 25
− 20	10 31	2 07	86 06	5 28	102 79	20
− 15	13 31	1 87	85 53	6 65	103 36	− 15
− 10	16 59	1 68	84 99	7 96	103 92	− 10
− 5	19 69	1 51	84 45	9 28	104 47	− 5
− 0	24 09	1 37	83 90	10 63	105 02	− 0
5†	28 33	1 25	83 34	11 97	105 56	5†
10	32 93	1 13	82 78	13 29	106 08	10
20	43 28	0 937	81 63	15 98	107 13	20
30	55 23	0 782	80 45	18 74	108 13	30
40	69 02	0 656	79 25	21 70	109 09	40
50	84 70	0 554	78 02	24 73	109 98	50
60	102 5	0 470	76 75	27 83	110 78	60
70	122 5	0 400	75 46	30 99	111 49	70
75	133 4	0 370	74 81	32 61	111 81	75
80	145 0	0 342	74 15	34 27	112 13	80
86†	159 8	0 311	73 36	36 28	112 47	86†
90	170 1	0 293	72 81	37 61	112 67	90
95	183 7	0 271	71 10	39 41	112 89	95
100	197 9	0 252	71 35	40 98	113 06	100
105	213 0	0.234	70.58	42 65	113 18	105
110	228 7	0 217	69 78	44 35	113 29	110
115	245 3	0 201	68 95	46 09	113 49	115
120	262 6	0 187	68 10	47 85	113 52	120

* Inches of mercury below 1 atmosphere.

† Standard ton temperatures

Table 9-16. Properties of Freon 113

Temperature, deg F	Pressure, psig	Volume vapor, cu ft/lb	Density liquid, lb/cu ft	Heat content, Btu/lb		Temperature, deg F
				Liquid	Vapor	
-30	29 31*	82 26	105 64	1 97	74 65	-30
-25	29 19*	60 35	105 30	2 96	75 35	-25
-20	29 05*	58 61	104 96	3 96	76 05	-20
-15	28 89*	49 82	104 61	4 96	76 76	-15
-10	28 69*	42 48	104 26	5 96	77 47	-10
- 5	28 47*	36 43	103 91	6 97	78 17	- 5
0	28 21*	31 31	103 56	7 98	78 89	0
5†	27 92*	27 04	103 20	8 98	79 60	5†
10	27 60*	23 45	102 84	10 00	80 32	10
15	27 22*	20 41	102 47	11 02	81 03	15
20	26 80*	17 81	102 10	12 03	81 75	20
25	26 32*	15 61	101 73	13 05	82 47	25
30	25 79*	13 71	101 36	14 08	83 20	30
35	25 19*	12 09	100 98	15 11	83 92	35
40	24 52*	10 68	100 60	16 16	84 65	40
50	22 94*	8 43	99 83	18 24	86 11	50
60	21 02*	6 71	99 05	20 35	87 57	60
70	18 68*	5 40	98 26	22 48	89 04	70
75	17 31*	4 87	97 85	23 54	89 77	75
80	15 87*	4 39	97 45	24 63	90 51	80
86†	13 93*	3 89	96 96	25 93	91 39	86†
90	12 53*	3 60	96 63	26 80	91 98	90
95	10 64*	3 27	96 21	27 89	92 72	95
100	8 59*	2 98	95 79	28 99	93 45	100
105	6 35*	2 71	95 37	30 11	94 20	105
110	3 95*	2 48	91 95	31 22	94 93	110
115	1 37*	2 27	94 52	32 34	95 67	115
120	0 70	2 08	94 09	33 48	96 41	120
125	2 16	1 91	93 65	34 60	97 15	125
130	3 74	1 75	93 22	35 75	97 89	130
135	5 44	1 62	92 77	36 90	98 62	135
140	7 23	1 49	92 33	38 05	99 36	140
150	11 23	1 27	91 41	40 38	100 82	150
160	15 74	1 09	90 53	12 74	102 29	160
170	20 83	0 94	89 60	45 12	103 74	170
180	26 52	0 82	88 67	47 53	105 19	180
190	32 90	0 71	87 72	49 97	106 63	190
200	39 96	0 62	86 76	52 45	108 07	200
210	47 80	0 65	85 79	54 96	109 50	210
220	56 37	0 48	84 80	57 49	110 92	220

* Inches of mercury below 1 atmosphere.
† Standard ton temperatures

Table 9-17. Properties of Freon 114

Temperature, deg F	Pressure, psig	Volume vapor, cu ft/lb	Density liquid, lb/cu ft	Heat content, Btu/lb		Temperature, deg F
				Liquid	Vapor	
−80	28.97*	51.26	105.60	−8.73	60.44	−80
−75	28.79*	43.28	105.12	−7.65	61.12	−75
−70	28.59*	36.40	104.79	−6.57	61.79	−70
−65	28.28*	30.75	104.37	−5.50	62.46	−65
−60	27.99*	26.06	103.96	−4.40	63.16	60
−55	27.60*	22.22	103.53	−3.30	63.88	−55
−50	27.20*	18.96	103.11	−2.20	64.53	−50
−45	26.70*	16.28	102.67	−1.10	65.21	−45
−40	26.12*	14.02	102.25	0.00	65.91	−40
−35	25.46*	12.08	101.81	1.13	66.60	−35
−30	24.72*	10.45	101.37	2.27	67.30	−30
−25	23.86*	9.09	100.92	3.40	68.00	−25
−20	22.91*	7.92	100.47	4.54	68.70	−20
−15	21.83*	6.93	100.01	5.67	69.40	−15
−10	20.63*	6.10	99.56	6.81	70.10	−10
−5	19.28*	5.38	99.09	7.95	70.80	5
0	17.79*	4.76	98.62	9.09	71.50	0
5†	16.14*	4.22	98.15	10.23	72.21	5†
10	14.31*	3.76	97.68	11.37	72.91	10
15	12.28*	3.35	97.19	12.52	73.61	15
20	10.07*	3.00	96.71	13.66	74.32	20
25	7.63*	2.70	96.22	14.81	74.98	25
30	4.99*	2.43	95.73	15.97	75.73	30
35	2.08*	2.18	95.23	17.12	76.44	35
40	0.52	1.98	94.73	18.28	77.14	40
50	4.03	1.63	93.71	20.61	78.55	50
60	8.13	1.35	92.68	22.95	79.95	60
70	12.87	1.13	91.63	25.30	81.35	70
75	15.41	1.03	91.09	26.49	82.04	75
80	18.34	0.95	90.56	27.68	82.73	80
86†	21.99	0.86	89.91	29.11	83.56	86†
90	24.59	0.81	89.47	30.06	84.11	90
95	28.03	0.74	88.92	31.26	84.80	95
100	31.69	0.69	88.37	32.47	85.48	100
105	35.58	0.64	87.81	33.67	86.15	105
110	39.71	0.59	87.25	34.89	86.83	110
115	44.10	0.55	86.66	36.10	87.50	115
120	48.74	0.51	86.08	37.33	88.16	120
125	53.66	0.47	85.49	38.57	88.82	125
130	58.84	0.44	84.89	39.80	89.47	130
135	64.32	0.41	84.28	41.05	90.12	135
140	70.09	0.38	83.66	42.29	90.76	140

* Inches of mercury below 1 atmosphere.
† Standard ton temperatures.

Table 9-18. Properties of CO_2

Temperature deg F	Pressure psig	Volume vapor cu ft/lb	Density liquid lb/cu ft	Heat content Btu/lb		Temperature deg F
				Liquid	Vapor	
−69.9	60.40*	1.157	74.6	−13.7	135.9	−69.9
−60	80.0	0.927	72.2	−10.1	136.6	−60
−50	103.5	0.749	71.0	−4.7	137.2	−50
−40	131.1	0.611	69.6	0.00	137.8	−40
−30	163.1	0.503	68.2	4.5	138.2	−30
−20	200.2	0.417	66.8	9.1	138.5	−20
−10	242.6	0.347	65.2	13.9	138.7	−10
0	290.8	0.290	63.7	18.8	138.9	0
5†	319.7	0.267	62.8	21.3	138.8	5†
10	345.5	0.241	62.0	24.0	138.7	10
20	407.1	0.205	60.1	29.4	138.3	20
30	476.1	0.172	58.2	35.4	137.8	30
40	553.1	0.141	56.0	41.7	136.7	40
50	638.9	0.121	53.5	48.4	135.0	50
60	733.9	0.099	50.8	55.5	132.1	60
70	838.7	0.080	47.3	63.7	127.5	70
80	954.0	0.061	42.2	73.9	118.7	80
86†	1,028.3	0.048	37.2	83.3	110.4	86
87.8‡	1,054.7‡	0.035	28.9	97.0	97.0	87.8‡

* Freezing point

† Standard ton temperatures

‡ Critical points

Table 9-19. Properties of Ammonia

Temperature, deg F	Pressure, psig	Volume vapor, cu ft/lb	Density liquid, lb/cu ft	Heat content, Btu/lb		Temperature, deg F
				Liquid	Vapor	
−60	18 6*	44 73	43 91	−21 2	589 6	−60
−55	16 6*	38 38	43 70	−15 9	591 6	−55
−50	14 3*	33 08	43 49	−10 6	593 7	−50
−45	11 7*	28 62	43 28	− 5 3	595 6	−45
−40	8 7*	24 86	43 08	0 0	597 6	−40
−35	5 4*	21 86	42 86	5 3	599 5	−35
−30	1 6*	18 97	42 65	10 7	601 4	−30
−25	1 3	16 66	42 44	16 0	603 2	−25
−20	3 6	14 68	42 22	21 4	605 0	−20
−15	6 2	12 97	42 00	26 7	606 7	−15
−10	9 0	11 50	41 78	32 1	608 5	−10
− 5	12 2	10 23	41 56	37 5	610 1	− 5
0	15 7	9 12	41 34	42 9	611 8	0
5†	19 6	8 15	41 11	48 3	613 3	5†
10	23 8	7 30	40 89	53 8	614 9	10
15	28 4	6 56	40 66	59 2	616 3	15
20	33 5	5 91	40 43	64 7	617 8	20
25	39 0	5 33	40 20	70 2	619 1	25
30	45 0	4 83	39 96	75 7	620 5	30
35	51 6	4 37	39 72	81 2	621 7	35
40	58 6	3 97	39 49	86 8	623 0	40
50	74 5	3 29	39 00	97 9	625 2	50
60	92 9	2 75	38 50	109 2	627 3	60
70	114 1	2 31	38 00	120 5	629 1	70
75	125 8	2 13	37 74	126 2	629 9	75
80	138 3	1 96	37 48	132 0	630 7	80
86†	154 5	1 772	37 16	138 9	631 5	86†
90	165 9	1 66	36 95	143 5	632 0	90
95	181 1	1 53	36 67	149 4	632 6	95
100	197 2	1 42	36 40	155 2	633 0	100
105	214 2	1 31	36 12	161 1	633 4	105
110	232 3	1 22	35 84	167 0	633 7	110
115	251 5	1 13	35 55	173 0	634 0	115
120	271 7	1 05	35 26	179 0	634 0	120
125	293 1	0 97	34 96	185 1	634 0	125

* Inches of mercury below 1 atmosphere.
† Standard ton temperatures.

Table 9-20. Properties of Methyl Chloride

Temperature, deg F	Pressure, psig	Volume vapor, cu ft/lb	Density liquid, lb/cu ft	Heat content, Btu/lb		Temperature, deg F
				Liquid	Vapor	
−80	25 94*	41 08	66 98	−13 80	184 75	−80
−70	24 32*	29 84	66 31	−10 52	186 25	−70
−60	22 19*	22 09	65 66	− 7 04	187 74	−60
−50	19 43*	16 61	65 02	− 3 53	189 19	−50
−40	15 92*	12 72	64 39	0 00	190.66	−40
−35	13 84*	11 20	64 08	1 78	191 37	−35
−30	11 52*	9 87	63 78	3 56	192 08	−30
−25	8 45*	8 74	63 47	5 35	192 79	−25
−20	6 09*	7 76	63 17	7 15	193 49	−20
−15	2 93*	6 91	62 87	8 94	194 18	−15
−10	0 27	6 18	62 58	10 75	194 87	−10
− 5	2 15	5 53	62 28	12 56	195 55	− 5
0	4 20	4 97	62 00	14 39	196 23	0
5†	6 46	4 47	61 65	16 21	196 92	5†
10	8 90	4 04	61 31	18 04	197 58	10
15	11 58	3 65	61 04	19 88	198 21	15
20	14 46	3 31	60 72	21 73	198 84	20
25	17 60	3 01	60 39	23 58	199 44	25
30	20 98	2 74	60 06	25 44	200 03	30
35	24 64	2 50	59 70	27 30	200 60	35
40	28 56	2 29	59 38	29 17	201 17	40
50	37 29	1 92	58 69	32 93	202 28	50
60	47 30	1 62	58 00	26 71	203 33	60
70	58 71	1 38	57 34	40 52	204 34	70
75	64 97	1 28	57 02	42 44	204 81	75
80	71 56	1 18	56 69	44 36	205 27	80
86†	80 00	1 08	56 24	46 67	205 80	86†
90	85 95	1 02	55 99	48 21	206 13	90
95	93 76	0 94	55 62	50 15	206 54	95
100	102 0	0 88	55 31	52 09	206 94	100
105	110 7	0 82	54 93	54 04	207 32	105
110	119 8	0 77	54 55	56 00	207 70	110
115	129 5	0 72	54 18	57 97	208 05	115
120	139 5	0 67	53 79	59 93	208 39	120
125	150 6	0 63	53 40	61 91	208 71	125
130	161 1	0 59	52 99	63 89	209 02	130
135	172 8	0 55	52 60	65 88	209 31	135
140	184 9	0 52	52 22	67 87	209 58	140
150	210 7	0 46	51 41	71 87	210 10	150
160	238 8	0 41	50 56	75 90	210 56	160
170	269 2	0 36	49 63	79 97	210 93	170

* Inches of mercury below 1 atmosphere

† Standard ton temperatures.

Table 9-21. Properties of Sulfur Dioxide

Temperature, deg F	Pressure, psig	Volume vapor, cu ft/lb	Density liquid, lb/cu ft	Heat content, Btu/lb		Temperature, deg F
				Liquid	Vapor	
−40	23.53*	22.42	95.79	0.00	178.61	−40
−30	21.10*	16.56	94.94	2.93	179.90	−30
−20	17.94*	12.42	94.10	5.98	181.07	−20
−10	13.91*	9.44	93.27	9.16	182.13	−10
0	8.85*	7.28	92.42	12.44	183.07	0
5†	5.88*	6.42	92.00	14.11	183.49	5†
10	2.61*	5.68	91.58	15.80	183.87	10
15	0.51	5.04	91.16	17.49	184.21	15
20	2.48	4.49	90.72	19.20	184.52	20
25	4.64	3.99	90.24	20.92	184.79	25
30	7.00	3.58	89.76	22.64	185.02	30
35	9.58	3.21	89.29	24.38	185.22	35
40	12.40	2.89	88.81	26.12	185.37	40
50	18.75	2.35	87.87	29.61	185.56	50
60	26.23	1.93	86.95	33.10	185.59	60
70	34.92	1.59	86.02	36.58	185.46	70
75	39.77	1.45	85.52	38.32	185.34	75
80	44.96	1.32	85.03	40.05	185.17	80
86†	51.75	1.19	84.44	42.12	184.92	86†
90	56.55	1.10	84.05	43.50	184.72	90
95	62.90	1.01	83.57	45.20	184.43	95
100	69.82	0.93	83.07	46.90	184.10	100
110	85.06	0.78	82.03	50.26	183.31	110
120	106.23	0.66	80.90	53.58	182.36	120
130	121.78	0.56	79.81	56.85	181.24	130
140	143.91	0.48	78.61	60.04	179.91	140

* Inches of mercury below 1 atmosphere

† Standard ton temperatures

Table 9-22. Properties of Isobutane

Temperature, deg F	Pressure, psig	Volume vapor, cu ft/lb	Density liquid, lb/cu ft	Heat content, Btu/lb		Temperature, deg F
				Liquid	Vapor	
−20	14.6*	11.00	38.25	−9.0	156.5	−20
−10	11.0*	8.91	37.95	−4.5	158.5	−10
0	6.3*	7.17	37.60	0.0	160.5	0
5†	3.3*	6.41	37.40	2.5	162.0	5†
10	0.2*	5.75	37.20	5.0	163.5	10
20	3.5	4.68	36.80	10.0	166.0	20
30	7.6	3.86	36.40	15.5	169.0	30
40	12.2	3.22	36.00	21.0	172.0	40
50	17.8	2.71	35.60	27.0	175.5	50
60	24.0	2.28	35.20	33.0	179.0	60
70	31.1	1.94	34.80	39.5	183.0	70
80	39.2	1.66	34.35	46.5	187.0	80
86†	44.8	1.52	34.10	50.5	189.5	86†
90	48.6	1.42	33.90	53.5	191.0	90
100	59.0	1.23	33.45	61.0	195.5	100
110	70.4	1.07	33.00	69.0	200.0	110
120	83.3	0.93	32.50	77.0	201.5	120
130	97.3	0.81	32.00	86.0	210.0	130
140	112.1	0.72	31.50	95.0	215.5	140

* Inches of mercury below 1 atmosphere.

† Standard ton temperatures

Table 9-23. Properties of Methyl Formate

Temperature, deg F	Pressure psig	Volume vapor, cu ft/lb	Density liquid lb/cu ft	Heat content, Btu/lb		Temperature, deg F
				Liquid	Vapor	
0	26.87*	54.0		0.00	232.5	0
5†	26.3*	47.3		2.6	233.6	5†
10	25.80*	40.5		5.2	234.0	10
20	24.43*	31.0		10.3	236.6	20
30	22.7*	24.0		15.5	238.7	30
40	20.45*	18.9		20.6	240.7	40
50	17.7*	15.1		25.8	242.8	50
60	14.41*	12.0		30.9	244.8	60
70	10.2*	9.80		36.1	246.8	70
80	5.35*	7.98		41.2	248.8	80
86†	1.84*	7.10	61.0	44.3	250.1	86†
90	0.23	6.51		46.4	250.9	90
100	3.56	5.38		51.5	252.9	100
110	7.5	4.46		56.7	255.0	110
120	12.54	3.74		61.8	257.0	120
130	17.95	3.17		67.0	259.0	130
140	23.71	2.65		72.1	261.0	140

* Inches of mercury below 1 atmosphere
† Standard ton temperatures

Table 9-24. Properties of Methylene Chloride

Temperature, deg F	Pressure psig	Volume vapor, cu ft/lb	Density liquid lb/cu ft	Heat content, Btu/lb		Temperature, deg F
				Liquid	Vapor	
5†	27.53*	49.90		1.7	163.8	5†
10	27.12*	42.55		3.4	164.4	10
20	26.01*	31.40		6.8	165.6	20
30	24.71*	23.90		10.2	166.9	30
40	23.04*	18.60		13.6	168.0	40
60	18.72*	11.68		20.4	170.1	60
80	12.02*	7.50		27.2	172.0	80
86†	9.41*	6.68	83.5	29.75	173.1	86†
100	2.97*	5.11		34.0	173.7	100
120	1.50	3.65		40.8	175.0	120
140	12.09	2.69		47.6	176.0	140

* Inches of mercury below 1 atmosphere
† Standard ton temperatures

CENTRIFUGAL REFRIGERATION MACHINES

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HISTORY

The centrifugal compressor as a means of pumping and compressing air and gases was originated in Europe in 1902. In this year, a crude air blower was built to the designs of the famous French engineer, Rateau. The first machine resembling to any great extent modern design was an air compressor built by Brown, Boveri & Co., Ltd., Baden, Switzerland, in 1906. The German firm of C. H. Jaeger & Co., Leipzig, entered the field in 1908 and was selected by W. H. Carrier in 1920 to build the first centrifugal refrigeration compressor.

At that time all the common refrigerants except carbon dioxide were irritating or toxic and therefore constituted a hazard if used in public places in conjunction with air-conditioning systems. As a solution to this problem Carrier selected the low vapor pressure, relatively high molecular weight substance, dichloroethylene, as a suitable refrigerant and at the same time recognized that this gas was ideally suited to centrifugal compression. It was in this way that the centrifugal refrigeration machine was originated.

In the 27 years which have elapsed since the first machine was built and sold commercially, several other manufacturers have entered the field, and about 2,400 machines have been installed in the United States. Because of the relatively large size of this equipment the total installed capacity of the above machines, expressed in horsepower, is probably close to 750,000 hp.

ELEMENTS AND PRINCIPLES

Compressor. The manner in which a centrifugal compressor develops pressure is usually not entirely clear to those unfamiliar with this type of equipment. Obviously, if any material, liquid or gas, is rotated at high speed, it will tend to be thrown outward from the center of rotation by centrifugal force. If this flow is in any way restrained, the centrifugal force will produce pressure within the material which increases with increasing distance from the center of the rotation.

This describes, in simple terms, the way in which a pressure rise is produced in the impeller of a centrifugal compressor. But additional pressure rise is developed in stationary parts of the compressor.

When the gas leaves the impeller of a centrifugal compressor, it has a high velocity V (see Fig. 9-1) relative to the compressor casing. By directing this high-velocity gas into suitably designed stationary passages called diffusers, a considerable portion of the energy equivalent of this high velocity can be converted to pressure.

There are three basic types of diffusers:

1. The open or vortex type
2. The volute type
3. The vaned type

All three types are in current use in refrigeration compressors.

It may be said, then, that pressure is developed by a centrifugal compressor through the action of dynamic forces on the gas particles in two ways:

1. In the impeller, by the action of centrifugal force
2. In the diffuser, by the conversion of kinetic energy of the gas leaving the impeller to energy of compression

In most cases the greater portion of the total compression takes place in the impeller.

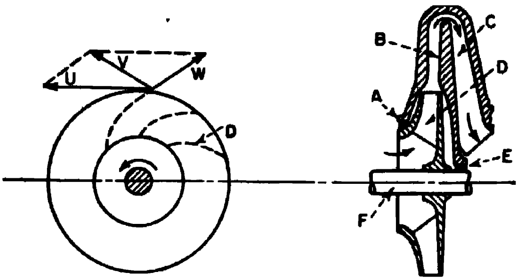

FIG. 9-1. Major components and velocity diagram of centrifugal compressor. (*A*) Inlet ring labyrinth; (*B*) diffuser passage; (*C*) return channel to following stage; (*D*) impeller blading; (*E*) shaft labyrinth; (*F*) shaft; (*U*) impeller-tip speed; (*V*) velocity entering diffuser, (*W*) relative velocity leaving impeller.

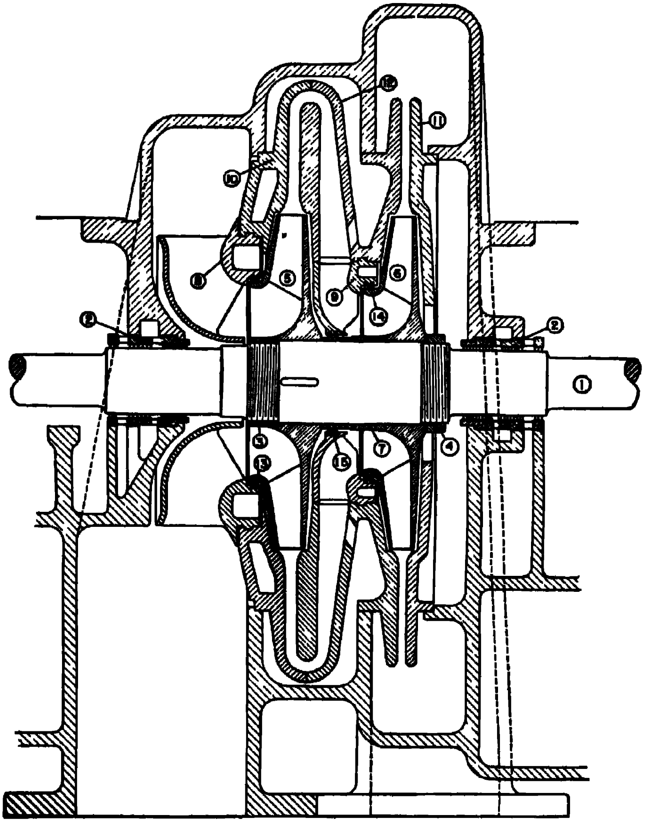

FIG. 9-2. Centrifugal compressor details. (1) Shaft; (2) shaft labyrinth, either end; (3) impeller lock nut, suction end, (4) impeller lock nut, discharge end, (5) impeller, first stage; (6) impeller, second stage; (7) impeller spacer; (8) inlet guide vane, first stage; (9) inlet guide vane, second stage; (10) intake wall; (11) discharge wall; (12) diaphragm, first stage; (13) inlet labyrinth, first stage; (14) inlet labyrinth, second stage; (15) spacer labyrinth.

In a centrifugal compressor the gas is in no way completely enclosed within definable walls and squeezed into a smaller volume as in the case of so-called "positive-displacement" compressors, whether of the reciprocating or rotary type. There are no major rubbing surfaces except shaft and thrust bearings and no close clearances except the labyrinth packings used to reduce interstage leakage in multistage machines (see Fig. 9-2). This fact makes it possible to segregate the lubrication system from the stream of gas passing through the compressor and practically eliminates contamination of the refrigerant by oil.

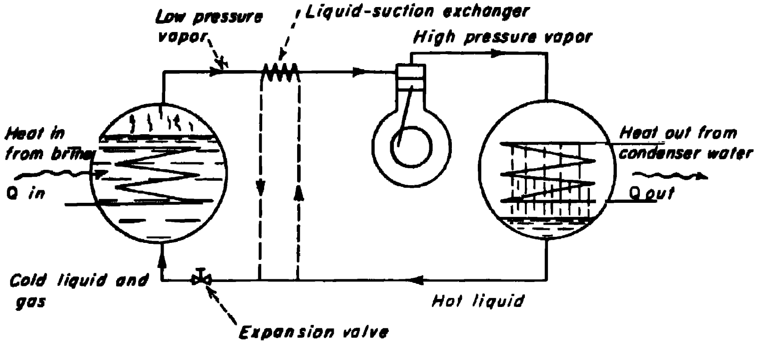

FIG. 9-3a. Simple refrigeration cycle.

Cycle. The simple single-stage refrigeration cycle shown in Fig. 9-3a is undoubtedly familiar. The only important variation is the optional use of a liquid-suction heat exchanger (indicated by dotted lines) for cooling the liquid returning to the evaporator at the expense of heating the compressor suction gas. In either case relatively warm liquid is throttled directly from condenser to evaporator pressure. Since this liquid must be cooled to the evaporator temperature, a definite fraction must be evaporated before any useful refrigeration can be produced. The resulting gas must be pumped back to the condenser through the total pressure ratio of the cycle.

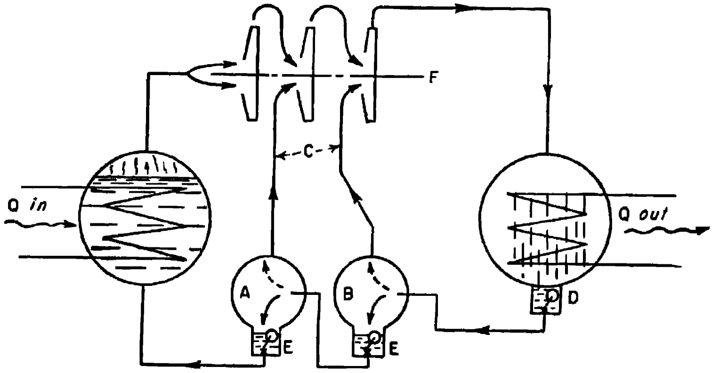

FIG. 9-3b. Refrigeration cycle with interstage liquid flash cooling. (A) Low-temperature economizer; (B) high-temperature economizer; (C) economizer gas lines; (D) condensate return trap; (E) economizer traps; (F) three-stage compressor.

Centrifugal refrigeration cycles almost always employ one or more stages of liquid "flash cooling" as shown in Fig. 9-3b. Instead of returning the condensate immediately to the evaporator, it is passed through one or more expansion chambers, or "economizers," which are connected to intermediate stages of the compressor. Means are provided in these chambers for separating the gas generated from the liquid and for passing the latter on to the next lower pressure. In this way all the useless evaporation necessary to cool the liquid from the condenser to the evaporator temperature does not occur at the lowest pressure. For the cycle shown, roughly a third of the total gas is compressed through the entire pressure range, another third is handled by the last two compressor stages, and the remainder passes through only the last wheel.

The use of intermediate liquid flash cooling in centrifugal refrigeration cycles results in important power savings, particularly when the temperature range (condenser minus evaporator temperature) of the cycle is large.

Evaporators or Brine Coolers. Centrifugal refrigeration machines almost invariably are designed with close-coupled evaporators or brine coolers because it is usually less expensive to pipe the load (either directly or by the use of a secondary refrigerant or brine) to the machine than to pipe the refrigerant to and from the load.

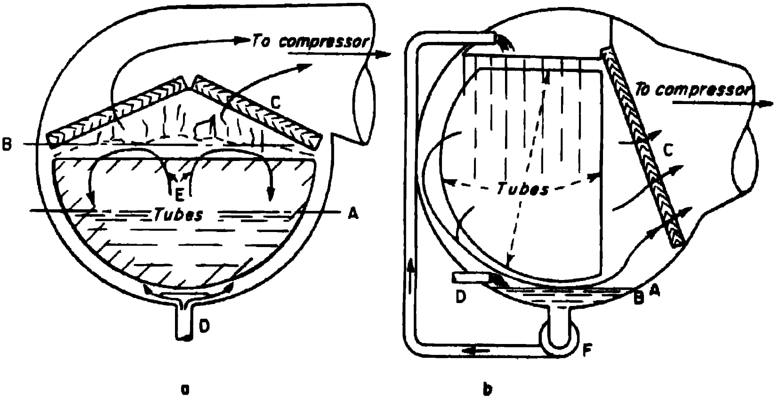

Fig. 9-4. Two common types of evaporators (*A*) Shutdown refrigerant level; (*B*) operating refrigerant level, (*C*) entrained liquid eliminator, (*D*) liquid return connection; (*E*) circulation during operation; (*F*) refrigerant pump

Two types of shell and tube evaporators are commonly used, the flooded type and the pump type, shown schematically in Figs. 9-4*a* and 9-4*b*, respectively. Flooded coolers are used whenever the loading of the heat-transfer surface and evaporator pressure is great enough to cause a violent boiling condition in and around the tube bundle. With low-vapor-pressure refrigerants like Freon 11, it is necessary to use a very compact type of finned tubing to develop the proper degree of circulation and assure high heat-transfer coefficients.

At low evaporator pressures the weight of the body of liquid refrigerant raises the pressure and hence the boiling temperature in the bottom of the shell to such an extent that the subsurface formation of gas and resultant agitation of the refrigerant is suppressed. The same result occurs if the heat-transfer coefficient on the brine side (inside tube surface) of the tube bundle is low. In either case the refrigerant-side heat-transfer coefficient and effectiveness of the surface are improved by distributing the liquid over the tube bundle with a pump as in Fig. 9-4*b*.

Condensers. The condensers of centrifugal refrigeration equipment differ only slightly from those used in other systems of equal capacity. Since the majority of machines use low-vapor-pressure refrigerants like Freon 11, the gas volumes entering condensers are relatively large, and more space must be provided within the shell for distribution of gas to the condensing surface.

The evaporator pressure of most centrifugal machines is subatmospheric. Any leaks which occur will therefore result in air, water, or brine entering the refrigerant system rather than escape of refrigerant. Condenser design must therefore provide means to collect and evacuate larger amounts of air and moisture than would normally occur in using refrigerants of higher vapor pressure.

The coexistence of air, water vapor, and Freon refrigerants in the condenser air-collection system introduces certain unique corrosion problems which must be considered in any successful design of these elements.

Purge Systems. The condenser pressures of centrifugal refrigeration units using low-pressure refrigerants may at times be subatmospheric. The system used to evacuate air and moisture must be designed accordingly. Details of the purge systems furnished by various manufacturers differ, but all normally provide the following essential functions:

1. Primary air and water-vapor concentration in the condenser shell
2. Means for transferring these gases to the purge unit
3. Additional concentration of the foul gases by condensation of some refrigerant and water vapor
4. Separation of water and liquid refrigerant and return of the latter to the machine
5. Relief of air and residual refrigerant and water vapors to the atmosphere

Besides its obvious duty of maintaining efficiency of the condensing surface on the refrigerant side, frequent observation of the purge system will reveal the incidence of any significant air, brine, or water leaks into the machine.

COMPRESSOR PERFORMANCE CHARACTERISTICS

The displacement or cubic feet per minute handled by a centrifugal compressor is not closely associated with speed. In fact, a machine running at constant speed will handle widely varying volumes with little change in pressure ratio but with a corresponding change in power requirement. This is indicated clearly in Fig 9-5. The three curves in each half of the figure apply to three different speeds, N_1, N_2, and N_3, and a constant or nearly constant suction temperature and density. The vertical coordinates are the ratio of condenser pressure to evaporator pressure or pressure ratio (lower) and input horsepower (upper).

Since the refrigeration or the cooling effect produced is almost proportional to the pounds of refrigerant circulated, the horizontal coordinate can be considered as the load in tons or Btu per hour which is imposed on the machine. The shape of the characteristic curve indicates that a considerable range of capacity can be handled at constant speed with relatively minor change in compression ratio.

The factors that influence the maximum compression ratio produced by a compressor are

1. The speed at which it is driven
2. The molecular weight of the gas handled. A higher pressure ratio is developed per stage with high-molecular-weight gases.
3. The number of wheels or stages
4. The diameter of the compressor wheels

The effect of items 1 and 4 is to establish a certain impeller "tip speed" (peripheral velocity). The compression ratio developed is closely related to this velocity (U in Fig. 9-1).

Capacity Control. The operating engineer should become familiar with the displacement-pressure ratio-power characteristics of this type of machine because they serve as a guide to operating procedures. For example, the characteristic in Fig. 9-5 indicates the relative performance obtained with different methods of control. In most instances the demand on the compressor is one of reducing pressure ratio with reducing load as represented by the line OA. This type of demand may be brought about by an interrelation between condenser water temperature and load, as in the case of air-conditioning applications. (Low outside temperature reduces both the load and condenser water temperature.) Or it may be that the only reduction in pressure ratio comes about by the reduced temperature difference required on the heat exchangers (brine cooler and condenser) at reduced load In any case, with a load-pressure ratio demand curve like OA, the lowest power will be required if the speed of the compressor is reduced. For this reason centrifugal refrigeration machines are usually driven by a variable-speed motor or turbine.

If a constant-speed driver such as a synchronous motor is used, practically equal power economy will result by interposing a slip coupling of either the hydraulic or magnetic type between driver and compressor.

If variable compressor speed is not provided, either suction throttling or condenser-water throttling may be used to balance the capacity with the demand. Suction throttling requires less power and is therefore used almost to the exclusion of the latter method.

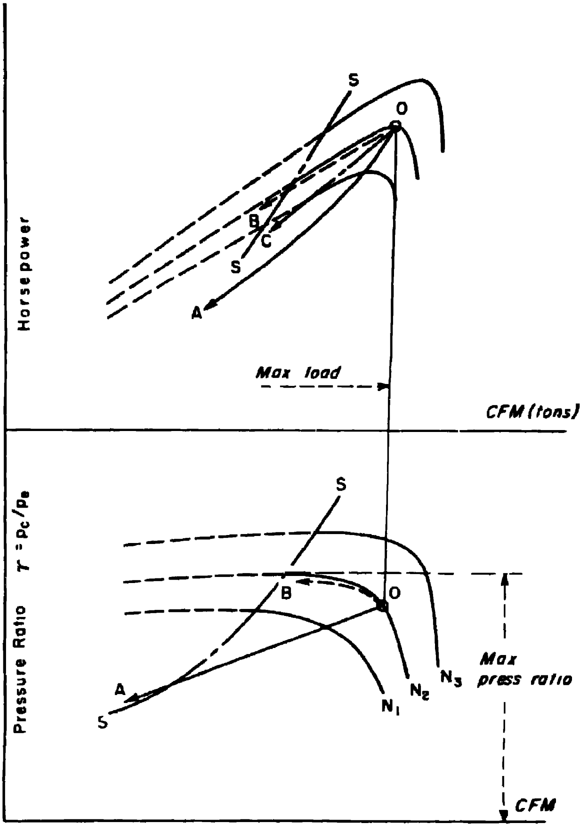

Fig. 9-5. Compressor and load characteristics.

It will be remembered that the characteristic curves shown in Fig. 9-5 are for a fixed-suction gas temperature. At higher suction temperatures the compression ratio at a given speed and cubic feet per minute will be lower. If the machine in question is designed to deliver water or brine at a constant temperature, the evaporator refrigerant temperature required will be higher at reduced loads. Since throttling does not materially alter the gas temperature, the cubic feet per minute-pressure-ratio characteristic when suction-damper or condenser-water control is used will be similar to that represented by the dotted curve *OB* in the lower part of Fig. 9-5.

The horsepower cubic feet per minute characteristic for condenser-water control will be as indicated by the dotted curve *OB* in the upper diagram. The power curves *OA* and *OB* can be compared directly because they refer to the same suction density and therefore to equivalent weight flows and approximately the same refrigeration load at particular cubic feet per minute.

The power–cubic feet per minute curve for suction throttling would only lead to confusion if placed on the same diagram because the effect of the suction damper is to reduce the pressure and density entering the compressor. The suction cubic feet per minute handled per unit of refrigeration is increased at part loads to such an extent that the operating point on the lower diagram moves back and away from point *O* very little as load is reduced. However, the weight flow of refrigerant reduces, and the power falls off accordingly.

If curves *OA* and *OB* are taken as the horsepower vs. tons characteristics for variable-speed and condenser-water control, respectively, the corresponding curve for suction-damper control will fall between them as indicated by the dash-dot line *OC* in the upper part of Fig. 9-5.

It will be realized, of course, that certain power losses result in variable-speed (slip-ring) motors and in slip couplings, and these losses must be included in any realistic power-consumption comparison. Even though these losses are added, it will be found that variable-speed drives are normally the most economical for centrifugal refrigeration units. In some instances, however, either the power saving does not justify the more costly motor, or the savings which may accrue from power-factor correction supplied by a synchronous motor offset the control losses.

Referring again to the cubic feet per minute -pressure-ratio characteristic in the lower half of Fig. 9-5, the reader will notice a line marked *SS* which cuts across the characteristic curves at their points of maximum compression ratio. To the left of this line the delivery of the compressor will be intermittent. The operation is often referred to as being unstable or surging. Generally, if the load is only slightly less than that required for stable operation, the frequency of the surge will be long. If the load is very light, the surging will be rapid or more like a flutter.

Surging is a normal characteristic of practically all centrifugal compressors. It has no detrimental effect on the compressor as long as the machine is rugged in design and the energy in the surge is limited. In the case of air or gas compressors, discharge check valves are usually installed to limit the intensity of the periodic reversal of flow through the compressor. Fortunately, in refrigeration applications the condenser (to which the compressor is usually closely coupled) acts like a check valve. The process of continuous condensation of the refrigerant leaves only a small amount in gaseous form on the discharge of the compressor at all times. The only danger normally associated with unstable operation of centrifugal refrigeration compressors is that overheating may occur at very light loads (15 to 20 per cent full load) if extended operation is undertaken. Overheating will be evidenced by abnormal temperatures of the bearing lubricant. A by-pass from condenser to evaporator is usually supplied on machines that must operate continuously at very light loads. It is not necessary to by-pass enough gas to stop the surge. This would be wasteful of power. Only that amount necessary to maintain normal bearing temperatures need be by-passed.

In some applications the sound of the surge may be objectionable. In such circumstances it is advisable to select the type of control which will reduce the point of instability to as low a load as possible. If load and condenser-water temperature fall off together or if light loads occur only in periods of low condenser pressure, variable-speed control will usually reduce surging almost to the vanishing point. This is indicated by the low-load intersection of lines *OA* and *SS* in the lower diagram of Fig. 9-5. With variable-speed control and this type of load characteristic surging occurs only at reduced compressor speeds where it is very mild in energy level.

If low loads may occur during periods of high condenser pressure and consequently high pressure ratio, a suction damper is most effective in suppressing the incidence of instability. For this reason a damper is often supplied even on machines equipped with variable-speed drives.

Capacity Range. As in the case of other types of compression refrigeration equipment, the capacity of centrifugal machines depends on their compressor displacement, the characteristics of the refrigerant used, and the operating-temperature level. Of the several Freon refrigerants, Freon 11 is normally used in machines for water cooling and for brine cooling down to about −20°F. For ultralow temperature applications, down to as low as −120°F or lower, and for large-capacity duty at higher temperature levels, Freon 12 is used. As Freon 22 becomes more available it may replace Freon 12 to some extent because of its slightly higher cycle efficiency.

Centrifugal refrigeration compressors are built for a range of displacements from about 2,000 to 20,000 cfm. Using Freon 11 for water-cooling duty, this range is equivalent to the capacity range of from 125 to 1,200 tons. Equipment of the same displacement but capable of higher compression ratios will produce from 40 to 400 tons on 0°F brine-cooling duty. The use of Freon 12 in similar machines designed for the higher pressure refrigerant results in capacities between five and six times those obtained with Freon 11. Freon 12 is, of course, used for ultralow temperature work. The relatively large displacements of centrifugal compressors make capacities of as high as 300 tons available in a single unit at −100°F suction temperature.

OPERATION

Centrifugal refrigeration equipment is relatively simple to operate. The inherently flat pressure ratio-volume characteristic for a given speed results in the tendency to maintain nearly constant evaporator and delivered brine temperatures under varying load conditions. This means fewer adjustments during operation. Turbine-driven or motor-driven units with hydraulic or magnetic slip couplings are easily provided with controls to maintain a constant delivered brine temperature by automatically adjusting compressor rpm. Machines with suction dampers can also be made automatic by the use of controlled damper positioners.

Normally there are no valves in the refrigerant circuit to be opened or closed. The equipment is usually started in the following sequence:

1. Brine and condenser-water circulating pumps
2. Refrigerant-condenser purge system unless it is designed for automatic operation
3. Lubricant cooling-water flow
4. Centrifugal compressor

If the brine is warm and condenser is free of air, the compressor will not surge when started and may be increased in speed to the limit of the safe input for the driver. Motor current should be observed and maintained below the full load rating either by the speed control or by closing the suction damper.

If the brine is cold and condenser water is warm (as in cases where several machines are operated in parallel) or if air in the condenser causes an abnormal elevation of the condenser pressure, the compressor will not pump any gas and therefore will not produce any refrigeration until its speed is increased to a point where the compression ratio it develops exceeds that established by the evaporator and condenser pressures. A rapid surge and low power demand are indicative of such conditions. Sometimes if machines have been opened for inspection or repair, large concentrations of air will be mixed with the refrigerant vapors. This has the effect of reducing the average molecular weight of the gas handled by the compressor and hence reduces the maximum ratio which can be developed. In such circumstances it may be impossible to get the machines to absorb load until the air concentration has been reduced by the purge equipment.

Shutdown equipment is normally stopped in the following sequence:

1. Compressor
2. Brine and condenser-water circulating pumps
3. Purge equipment unless it is designed for automatic operation
4. Lubricant cooling-water flow

ROUTINE SERVICING AND TROUBLE SHOOTING

In centrifugal refrigeration equipment the item of mechanical wear is reduced almost to the vanishing point. The major danger to bearings and other internal parts is corrosion. Annual inspection of the thrust and shaft bearings is advisable.

Besides the routine duty of maintaining heat-exchange surface (evaporator and condenser) in a clean and efficient condition, the operator's major concern should be to prevent as far as possible the contamination of the refrigerant with air, water, and oil. The presence of air and water in Freon refrigerant systems is the primary cause of corrosion. Since inward leakage of air always carries with it some water vapor and since even the most effective purge-recovery system must discharge some refrigerant with purged air, the operator who maintains his equipment "vacuum tight" will reduce both internal corrosion and refrigerant loss. *Machines should be checked periodically for inward air leakage and water or brine leakage from the heat exchangers into the refrigerant. The purge system should be maintained in good condition* so that it can remove air and water from the system if it enters and hence serve to indicate trouble.

Ordinarily the capacity and efficiency of centrifugal compressors are not directly related to the hours or years of operation. It is not unusual for a machine to maintain its original performance for 10 years or more with only minor servicing of auxiliaries and maintenance of heat exchangers. In fact, the only circumstances which can alter the performance of a centrifugal compressor are the development of excessive by-passing of gas between stages or the formation of deposits of scale or other foreign material in the gas passages of impellers and diffusers.

Interstage by-passing can result from increased clearances at the labyrinth packings caused by whipping of the rotor shaft. If no violent vibration has ever been noticed during operation, it is unlikely that labyrinths have been damaged. Likewise the deposit of foreign materials in the gas passages will eventually result in noticeable unbalance. Hence, abnormal vibration originating in the compressor is a sign of trouble, and inspection of the compressor rotor and diffusers is advisable. Otherwise the compressor assembly should require no servicing.

The gas-temperature rise from intake to discharge is an approximate indication of the power per pound used to compress the refrigerant. An unusually high discharge temperature, therefore, indicates poor performance and possibly internal recirculation. Compressor efficiency normally varies so much with load and speed, however, that only expert interpretation and careful measurements can definitely establish the facts. However, a cool or cold discharge temperature is indisputable evidence that entrained liquid refrigerant is entering the compressor with the vapor either from the evaporator or the economizers. Limited entrainment has no material effect on power requirement. In fact, liquid injection is sometimes used to limit discharge temperatures at high compression ratios. Excessive entrainment will increase power consumption and may be the cause of dirt or scale deposits in impeller and diffuser channels. The causes of a cold discharge temperature should, therefore, be located and corrected.

Centrifugal compressors should use very little, if any, oil. Absorption of Freon refrigerants in the oil will increase its volume and cause foaming at the beginning of operation. Observation of oil level should only be made at the end of operating periods when oil is warm and comparatively free of refrigerant. Oil should be added only when absolutely necessary, and excessive consumption (more than 3 or 4 qt per year) should be investigated.

APPLICATIONS

Centrifugal refrigeration equipment is applicable to practically all types of commercial and industrial services. Machines may be applied for cooling water brines, and liquids in processes of all sorts, and for the direct condensation and cooling within the evaporator of hydrocarbon vapors, ammonia, chlorine, and other industrial gases.

Such equipment may also be applied as a "heat pump" where fuel is scarce but electric power is available. The heat-pump principle involves the extracting of low-level heat (42 to 70°F) by the evaporator from the atmosphere or river and the delivery as high-level heat (120 to 150°F) from the condenser. Many such applications have been made on the European continent.

In general, centrifugal refrigeration machines are most applicable in sizes above 100 hp, although for ultralow-temperature applications this figure might be reduced.

A type of application that should not go unmentioned is to be found in the petroleum and chemical industries. Here a cooling effect is often produced by the direct evaporation, compression, and condensation of the materials in process or of a solvent injected for specific reasons. Centrifugal compressors are applicable to such processes which may be generally classified as autorefrigeration or solvent-recovery cycles.

Although all forms of commercial drive equipment may be used for centrifugal refrigeration compressors, the rotative speed and other characteristics of the steam turbine are ideal. Furthermore, in industrial applications where by-product steam is required or is available (breweries, meat-packing plants, refineries, etc.) the item of operating cost chargeable to refrigeration can usually be reduced by fitting the refrigeration turbine into the over-all steam-power balance of the plant.

RECIPROCATING COMPRESSION SYSTEMS

BY H. B. POWNALL

President, Freezing Equipment Sales, Inc.

The purpose of this subsection is to describe briefly the component parts and equipment making up a reciprocating compression refrigeration system. It is not the intent to go into detailed design but merely to describe the basic pieces of equipment as well as to discuss their operating characteristics.

RECIPROCATING COMPRESSORS

Reciprocating compressors may basically be subdivided into two classes, *viz.*, vertical- and horizontal-type compressors. In vertical compressors the center line of the cylinders is perpendicular to the floor, whereas horizontal compressors have the center line of the cylinders parallel to the floor. These two classifications may be again subdivided into single- and double-acting compressors. In a single-acting compressor the piston compresses gas during only one part of the stroke, whereas a double-acting compressor will compress gas in both ends of the cylinder.

These compressors may also be classified as being enclosed or semienclosed, depending on the way the gas is taken into the machine. In an enclosed-type compressor, the crankcase will be equalized to the low side pressure, and the lubricating oil is in contact with the refrigerant at all times. On a semienclosed machine a crosshead design is used so that the stuffing box is between the cylinder and the piston rod with the result that the crankcase can be equalized to atmospheric pressure. The most common compressor would be the enclosed type wherever it can be used since, having less moving parts, it is the more economical to build. It is necessary in some instances to use a semienclosed compressor where the refrigerant will break down the oil in the crankcase or where considerable foreign material is expected to flow back to the compressor.

The recent trend has been to design towards multicylinder, high-speed compressors. These compressors can be made either in line, radial, or in blocks forming Vs and Ws.

The capacity of reciprocating compressors will vary from a small fraction of a ton, such as the 1/6- and 1/8-hp compressors used on the domestic refrigerators, to several hundred tons capacity such as will be found in compressors of 14-in. bore by 13½-in. stroke, four cylinders in a line. From practical considerations, the top limit of bore and stroke is around 20 in. For single-stage compression where pressure differentials may approach 200 psi, the bearing pressures become excessive for larger bores, and good design in piston speeds will force the rotative speed of the compressor down. By using multicylinder radial or V-W compressors, the cylinder bore and stroke may be kept small, which allows high rotative speeds with resultant large increments of capacity.

TYPE OF REFRIGERANTS

The type of refrigerant to be used in conjunction with a reciprocating compressor must be carefully chosen. First, the refrigerant must have thermodynamic characteristics such that the volume of gas handled, commonly referred to as cubic feet per minute per ton, is maintained within the range of capacity of a positive-displacement type of compressor. The pressure differential of the refrigerant must be such that reasonable bearing loadings will be obtained. The density of the refrigerant will control the necessary valve area needed. The polytropic expansion coefficient must be such that reasonable discharge temperatures will be obtained with ratio of compressions as high as 8 to 1 and 9 to 1. It is also necessary to consider the characteristic mixtures which are obtained between lubricating oil and the refrigerant.

Making a brief comparison between ammonia and Freon 12, it might first be stated that the cubic feet per minute per ton required for ammonia is approximately 50 per cent less than that required for Freon 12. However, the pressure differential, when working from a constant evaporator temperature to a constant condensing temperature, is approximately 50 per cent greater for ammonia than for Freon 12. It is for this reason that on ammonia compressors designers will generally pick cylinder dimensions of equal bore and stroke such as 10- by 10-in. compressors. In designing this compressor for Freon 12, it is possible to go to a larger cylinder diameter for the same stroke and maintain the same bearing loading. Thus, it will generally be noted that on Freon 12 compressors the bore is generally larger than the stroke.

Since Freon 12 gas is approximately five times as heavy as ammonia gas, it is necessary to handle considerably more pounds per minute through the ports to the compressor with the result that considerably more valve area is required in a Freon 12 compressor than in an ammonia compressor.

The solubility of ammonia gas in oil is practically negligible. Thus the crankcase oil is almost always pure and has very little tendency to foam upon reduction of pressure. Ammonia liquid and oil are also insoluble, and therefore, the oil, being heavier than the ammonia liquid, will separate and go to the bottom when a quiescent condition exists.

Freon 12 gas is soluble in all proportions in oil. There is a very definite pressure-temperature relationship established over oil and Freon mixtures. This solubility results in considerable foaming upon sudden reduction of pressure, and so it is very difficult to keep Freon oil in the crankcase of Freon compressors when the suction pressure is suddenly reduced.

Freon liquid and oil are also soluble and follow the same equilibrium curve as the Freon gas and oil. This means that, as the oil concentration builds up in Freon 12 liquid, the oil will not separate but tends to raise the boiling point of the refrigerant.

REFRIGERATION CYCLE

Figure 9-6 shows a schematic mechanical refrigerating system. It should be noted that the compressor is acting merely as a heat pump in that it forces heat to flow from a low temperature level to a high temperature level. This is accomplished by removing vapor which is formed by evaporating liquid on the low side of the system. This liquid is evaporated by the heat which it removes from the space or fluid to be cooled. The compressor raises the pressure of this vapor sufficiently high so that, when discharged into a condenser, this heat may be rejected to the atmosphere or cooling water by condensing the high pressure vapor to a liquid where it is again returned to the evaporator through a throttle or expansion valve.

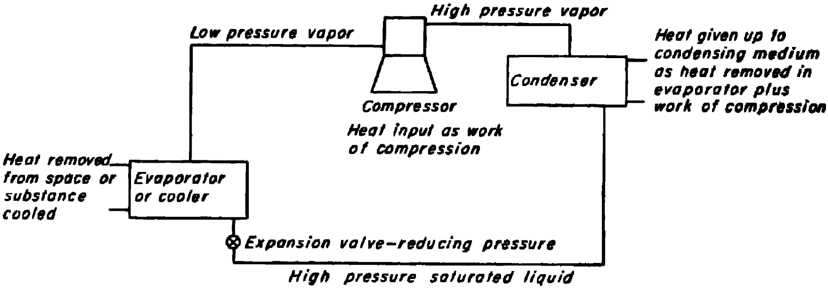
High pressure vapor
Low pressure vapor
Heat given up to condensing medium as heat removed in evaporator plus work of compression
Condenser
Compressor
Heat input as work of compression
Heat removed from space or substance cooled
Evaporator or cooler
Expansion valve—reducing pressure
High pressure saturated liquid

FIG. 9-6. Schematic mechanical refrigerating system.

Without going into too much detail it can readily be seen from such a cycle that, the lower the evaporator pressure, the more inefficient this system will become when operating at a constant condensing pressure. This condition is easily explained since, as the evaporator pressure is reduced, the vapor created becomes rarer with the result that the compressor in handling a constant volume will handle less total pounds of the vapor. The refrigeration effect is naturally a function of the number of pounds of refrigerant evaporated in the evaporator. Thus, it can be said that, as the suction pressure of a compressor is lowered, the refrigeration effect will decrease approximately proportional to the absolute suction pressure; the total brake horsepower driving the compressor will also decrease; however, the brake horsepower per ton of refrigeration required by the compressor will increase.

The condensing pressure is controlled by the temperature of the cooling medium used, such as water or air. Generally speaking, a decrease in condensing temperature will increase the capacity of the refrigeration system and will decrease the total brake horsepower required by the compressor. Likewise, an increase in condensing temperature will decrease the refrigeration effect of the compressor and will generally increase the total brake horsepower required by the compressor.

TYPES OF CONDENSERS

Air-cooled Condensers. Air-cooled condensers are quite common on small commercial condensing units for capacities up to and including approximately two tons of refrigeration. They are seldom used for capacities greater than this since they require too much surface and air circulation to be economical. When using an air-cooled condenser, the condensing temperature corresponding to pressure can be as high as 130°F, with an air temperature as high as 100°F entering the condenser. For this reason air-cooled condensers are generally used only with refrigerants that have relatively low vapor pressures. For example, Freon 12 will have a corresponding vapor pressure of 180 psi at 130°F, whereas the corresponding ammonia would be approximately 315 psig. Thus, air-cooled condensers are very rarely used for ammonia applications.

Evaporative-type Condensers. Evaporative-type condensers are a combination of air-cooled and water-cooled condensers. Water is sprayed over the condensing surface while a fan pulls air through the condensing surface with the result that the condensing-surface temperature is maintained at the approximate average wet-bulb temperature between the leaving and entering air. Entering dry-bulb temperature has very little effect upon the operation of an evaporative-type condenser This is true since the entering air will saturate itself almost immediately upon coming in contact with the water. These condensers are becoming quite popular for both Freon 12 and ammonia use since they use approximately the same amount of water as a cooling tower, which consists of the theoretical $\frac{1}{4}$ lb per min per ton of refrigeration plus losses, as well as the fact that they combine the benefits of a cooling tower and water-cooled condenser into a single unitary piece of apparatus.

Water-cooled Condensers. Water-cooled condensers can be subdivided into vertical open types and horizontal closed types. In either case all heat removed is given up to the water flowing through the condenser. The vertical open-type condenser allows the water to flow down through the tubes by gravity, and as the name implies, the tubes are open on both ends which allows them to be cleaned during normal operation.

The horizontal closed-type condenser is generally a multipass condenser to enable high water velocities with resultant optimum heat transfer regardless of water quantity. The increasing popularity of extended surface tubes is responsible for the industry developing smaller and smaller shell sizes for a given capacity in the line of horizontal multipass condensers. This type of condenser must naturally be used either with city water, well water, or, on larger installations, cooling-tower application. The main disadvantage of the new highly efficient small multipass condensers is that, in areas having bad scale-forming water, the condenser tubes must be cleaned frequently.

TYPES OF EVAPORATORS

The subject of evaporators for refrigeration plants is one which in itself could be responsible for several volumes. Basically, the main purpose of an evaporator in a refrigeration plant is to boil the liquid refrigerant from a liquid phase into a vaporous phase. The refrigerant, in changing state, absorbs heat from the space or object to be cooled. The most common types of evaporators could be classified as follows:

1. Room pipe
2. Brine and water coolers
3. Air-cooling coils

METHODS OF CONTROL IN REFRIGERATION PLANTS

Most refrigeration plants today require quite a few automatic controls. These plants may be classified either as automatic or semiautomatic. Semiautomatic plants will have a considerable number of automatic controls in an effort to simplify operation but will require the attendance of an operator for such duties as to add oil to the compressor crankcase and make various minor adjustments. Commercial refrigeration has developed fully automatic plants which require no attention other than periodic inspection and cleaning.

Capacity Control of Reciprocating-type Compressors. Capacity control of reciprocating-type compressors can generally be accomplished in one of the following manners:

1. Speed control of the compressor is by nature the best type of control which can be used. This is true since the efficiency of the compressor is maintained when operating at reduced capacity as well as the fact that the percentage capacity reduction will be directly proportional to the speed reduction regardless of operating conditions. The disadvantage of speed control is the method of obtaining it, since the majority of installations today are electric drive, with the result that multispeed motors must be used which considerably complicate the electrical side of the plant. It is necessary to use two-speed motors or slip-ring motors in conjunction with resistance banks.

2. The use of multiple compressors is another way in which capacity control can be gained since compressors may be cut on and off the line to match plant capacity. This method of control, while thermodynamically efficient, is generally undesirable from a cost standpoint when more than two or three steps of capacity reduction are required. With this type of control on automatic plants, the crankcase of the compressors must be equalized. This sometimes presents a considerable problem.

3. Capacity control may be achieved automatically through the use of gas by-pass. Compressors of one design have a so-called "partial by-pass" which means that a portion of the gas pumped by the cylinder is by-passed back into the suction main. This method has the advantage that there is always some gas flowing through the cylinder in order to absorb some of the frictional heat.

An external by-pass from the high to low side may be used for capacity reduction, provided that some means of cooling the gas entering the suction of the compressor is used. This method is the least efficient since full compressor horsepower is required regardless of capacity.

4. Clearance pockets may be used to reduce capacity by designing some mechanical means of increasing the clearance in the cylinder. This type of capacity reduction is highly efficient, provided the clearance can be obtained without wire drawing. The only objections are purely mechanical, in that it is difficult to obtain sufficient clearance to give necessary capacity reduction.

It is of prime importance to design the controller actuating the capacity-reduction means so that there is sufficient time lag between steps of capacity to allow the plant to rebalance. It should be remembered that, wherever finite steps of capacity reduction are taken, the temperature difference between the initial and final temperature of the fluid being cooled will decrease by the amount of the capacity reduction. This means that either a very insensitive instrument will have to be used, which is objectionable because of the lack of control, or else sufficient time lag must be placed in the control circuit either to allow the compressor to rebalance under the reduced capacity condition or slowly cycle back and forth between two steps of capacity. Short cycling of refrigeration plants due to improper setting or application of controls can be very annoying from the standpoint of dependable operation as well as maintenance costs for repairs at the end of the year.

PLANT OPERATION

One of the most important requirements in the operation of a reciprocating-compressor type of refrigeration plant is to be sure that the compressor is handling dry gas on the suction side of the machine. Suction superheat is theoretically the measure of dryness, but it is not always possible to use. When using flooded-type evaporators where the vapor disengages from the body of liquid, with no superheating surface between the evaporator and compressor, this suction gas can enter the machine at saturation temperature. If the gas is carrying considerable quantities of liquid with it, the suction temperature will still remain at saturation. It is also possible on plants operating with refrigerants having high latent heat, such as ammonia, to have small quantities of liquid rolling along the bottom of the suction line with superheated gas flowing over top of the liquid, since insufficient time of contact was allowed for equilibrium conditions to be reached. In the above two cases suction-gas temperature is not a true measure as to whether liquid is returning to the compressor.

The discharge temperature from a refrigeration compressor is probably one of the best indications of liquid slop-over as well as one of the most important readings to be watched in any refrigeration plant. Normally the discharge temperature from a compressor should run 20 to 30°F higher than the expected adiabatic discharge temperature. It is possible to run at the expected adiabatic temperature for most compressors without danger, although this is generally an indication of wet compression. When discharge temperatures are running lower than adiabatic, it means that certain amounts of liquid are entering the cylinder and cooling the gas during compression but that this liquid is completely evaporated at the end of the compression stroke.

A continuous log of discharge temperature on a compressor is generally the best indication of trouble occurring in the machine. A valve breakage or any blowback from high pressure to low pressure within the machine will be apparent by a partial increase in the actual operating discharge temperature.

The presence of air or other noncondensable gases is a detriment to the operation of a refrigeration system in that, for a given cooling-water temperature, the presence of air results in an increased condensing temperature which results in lowered capacity and over-all plant efficiency. Air normally collects in the condensing or high-pressure side of the system and is purged at this point either by hand or by automatic purge devices. The presence of noncondensable gases in the condenser may be detected by checking the temperature of the condensate liquid against the temperature corresponding to pressure in the condenser. Normally, under stand-by conditions, if the temperature of the liquid condensate is more than 3 or 4°F below that corresponding to saturation at the observed pressure, air is present, and purging is necessary. Whenever the evaporator side of the system is operating in a vacuum, particular attention must be given to prevent possible air leakage through valve packing, stuffing boxes, etc.

For good operation, it is essential for the condenser tubes to be free of scale. The presence of scale in an air-free condenser is noted by an increase in the temperature difference between the water and the saturation temperature corresponding to pressure. Whenever this temperature increases appreciably over that for known clean conditions, the tubes should be cleaned.

AMMONIA-ABSORPTION REFRIGERATION SYSTEMS

BY W. H. BICKFORD

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The continuous ammonia-absorption refrigeration system performs the same basic function as the reciprocating compression system but in a somewhat different manner. In the absorption system the transfer of refrigerant from evaporator pressure to condensing pressure is by virtue of the solubility relationships between ammonia and water.

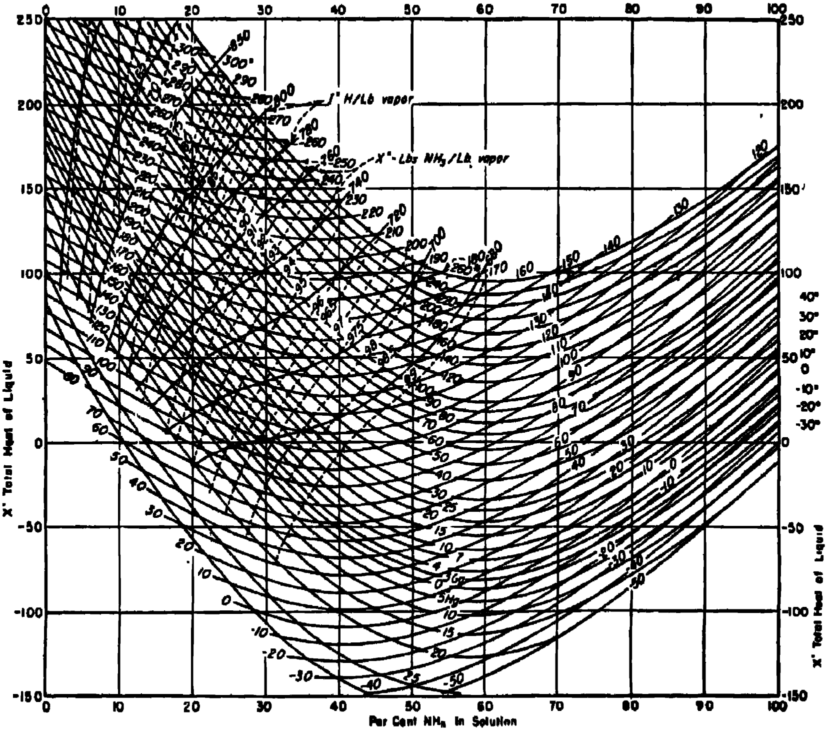

FIG. 9-7. Thermal properties of ammonia-water solutions (copied from A. B. Stickney chart).

The thermal properties of ammonia-water solutions are shown in Fig. 9-7 from which it may be seen that

1. At constant pressure the quantity of ammonia in a solution decreases with an increase in temperature.
2. At constant temperature the quantity of ammonia in a solution decreases with a decrease in pressure.
3. Heat must be removed for a solution to absorb more ammonia at constant pressure.
4. Heat must be added to decrease the concentration of a solution at constant pressure.

In a typical system (Fig. 9-8), the ammonia refrigerant is vaporized in the evaporator, and the vapor passes to the absorber and is absorbed in a weak solution of ammonia and water (weak aqua) forming a strong solution (strong aqua). The strong aqua is transferred to the generator through exchanger by means of aqua pump. In the generator the refrigerant vapor is boiled out of solution through the bubble column to condenser where the vapor becomes liquid. The weak aqua solution formed in the generator passes under condensing pressure through exchanger, undergoes a reduction in pressure, and enters absorber at essentially evaporator pressure. In the exchanger heat is transferred from the hot weak aqua leaving the generator to the cool strong aqua from the absorber, decreasing the amount of heat-transfer surface required in absorber and generator as well as generator heat and over-all cooling-water requirements.

The rectifier is used to remove water carried by the refrigerant vapor leaving the generator, thus ensuring essentially pure ammonia vapor to the condenser. A small amount of condensed liquid, known as reflux, is returned to the rectifier for this purpose.

Each piece of equipment in the cycle has a part in the *plant heat balance* as follows:

Generator—Heat is added at high temperature.

Evaporator—Heat is added at low temperature.

Pump -- Heat is added in the form of work performed.

Absorber--Heat is removed by water

Condenser—Heat is removed by water.

Rectifier—Heat is exchanged between liquid ammonia reflux and hot vapor from the generator.

Exchanger--Heat is exchanged between strong and weak aqua.

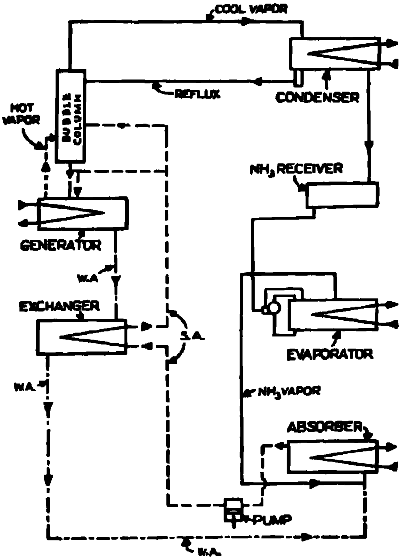

FIG. 9-8. Absorption-system flow diagram.

The heat exchanged in rectifier and exchanger is of internal nature and, while affecting the over-all plant balance, is not considered as a separate item.

In a typical system, the heat quantities per pound of refrigerant supplied to the evaporator are

Q_G = heat supplied to generator

Q_E = heat removed in evaporator

Q_P = heat equivalent of pump work

Q_A = heat removed from system by absorber water

Q_C = heat removed from system by condenser water

Q_L = heat losses from system through radiation, etc.

Neglecting losses and incorporating heat equivalent of pump work into generator heat supplied, the heat balance is as follows:

$$Q_G + Q_E = Q_A + Q_C$$

The above quantities may be evaluated through use of table of thermodynamic properties of ammonia, steam, and ammonia-water solutions.

The absorber load Q_A is comprised of three factors: (1) the heat of condensation of ammonia vapor, (2) the heat of solution as the ammonia is absorbed, and (3) the heat removed in cooling the weak aqua solution to absorber temperature. Obviously, once the evaporator, condenser, and absorber loads are known, the generator load may be obtained by difference. Approximate values of absorber and condenser loads may be obtained from Table 9-25.

Table 9-25. Approximate Values of Absorber and Condenser Loads

Aqua concentration, per cent NH_3 by weight		Load, Btu/lb ammonia evaporated							
		Condenser				Absorber			
		Pressure, psig				Pressure, psig			
Strong	Weak	205	185	165	145	40	30	20	10
45	35	593	590	587	584	807	815	820	830
44	36	598	595	592	590	795	804	810	818
43	37	603	600	598	591	775	784	790	798
40	30	623	620	610	614	844	851	859	866
39	31	630	627	624	621	830	840	846	855
38	32	638	635	632	628	811	820	826	835
35	25	666	662	658	654	880	887	895	904
34	26	677	673	668	665	867	875	883	891
33	27	689	685	680	676	845	855	862	865
30	20	730	725	718	714	915	925	932	940
29	21	747	742	735	729	905	913	920	930
28	22	765	759	751	745	885	893	900	910

Current preference in equipment is toward the use of horizontal shell and tube construction for condensers, absorbers, generators, and distillation column-type rectifiers. Centrifugal-type aqua pumps are preferred with either motor or turbine drive.

From an operating viewpoint, the primary requisite is the avoidance of sudden changes. Adjustments to operating variables should be made gradually.

For correct economical operation it is necessary that proper balance be maintained throughout the unit. The heat input to the generator increases as the aqua solution decreases in strength. Therefore, the aqua concentrations should be held as high as possible regardless of the load on the system. The amount of heat input to the generator also increases with an increase of "compression ratio," so the absorber pressure should be maintained as high as possible while doing refrigeration at the required temperature.

The concentration of the strong aqua is dependent upon the amount and concentration of the weak aqua supplied to the absorber, together with the amount of vapor that is absorbed. Therefore, in order for the system to operate at best conditions, the quantity of weak aqua circulated should be in proportion to the amount of vapor from the evaporator. If the full-load quantity of aqua is circulated when the load is decreased, the final concentration of the strong aqua from the absorber is decreased from the design condition. This results in a lower absorber pressure, and unless the evaporator pressure is controlled by throttling the vapor, the evaporator pressure will also decrease. If the evaporator temperature is controlled by a valve causing a pressure drop between the evaporator and the absorber, the system operates on a lower aqua concentration than necessary, with the result that the heat input to the generator is excessive.

The evaporator pressure may be controlled by increasing the concentration of the weak aqua from the generator by lowering the steam pressure. However, with this control, the full amount of aqua in circulation must be heated from exchanger temperature to the final generator temperature, thus requiring more steam than if the aqua quantity were reduced to correspond with the reduction in refrigerating load.

The quantity of aqua circulated should be decreased as the load decreases on the unit, with the aqua flow being decreased a little more rapidly than the decrease in the amount of vapor going to the absorber from the evaporator. This gradually changing ratio between the quantity of aqua circulated and the quantity of vapor to the system is difficult to maintain accurately either with manual operation of the plant or by the use of automatic control, so that in most cases the aqua flow is reduced in direct proportion to the reduction in quantity of vapor being absorbed.

Loss of capacity in an absorption system may be attributed to any of the following causes:

1. Presence of noncondensable gases in absorber and condenser as indicated by temperature-pressure relationships
2. Insufficient cooling water supply to condenser and absorber
3. Excessive fouling of absorber and condenser tubes
4. Insufficient refrigerant or aqua charge in plant. (Proper levels should be maintained in all equipment.)
5. Unsteady steam pressure. (The generator should have even source of heat.)
6. Accumulation of condensate on steam side of generator. (Tubes must drain properly.)

The concentration of the strong and weak aqua solution may be obtained with fair accuracy from temperature and pressure conditions in the absorber and generator, respectively, provided the system is purged of noncondensibles. Exact determination of solution concentration may be made by chemical analysis of samples withdrawn from the system.

The operating conditions for a given evaporator pressure are fixed by the steam pressure and cooling-water temperature. Generally, it may be stated that, for optimum operation, steam and water conditions should allow at least an 8 per cent difference in concentration between the strong and weak aqua streams, although it is quite possible to operate systems with larger or smaller concentration differences.

The absorption refrigeration system finds its chief application where a sufficient quantity of waste heat is available in the form of steam. hot oil, etc. Systems may be designed to operate single or double stage, with series or parallel flow of cooling water through condenser and absorber, allowing considerable flexibility when properly operated.

The quantity and temperature level of the generator heat supply as well as the temperature and quantity of cooling water to absorber and condenser is largely a function of design. The most advantageous applications are in cases where suitable amounts of waste or low-grade heat are available. This heat may be available as exhaust steam, quench oil, or similar heat which might normally be lost but which could be used to produce refrigeration.

Table 9-26. Steam and Water Rates per Ton of Refrigeration

(Single-stage—steam and water rates are based on series flow of water through condenser and absorber, 85°F H_2O on condenser and 8 per cent aqua split)

Absorber pressure, psig	Steam pressure, psig	Steam rate, lb/hr/ton	Cooling-water rate, gpm/ton
5	46.5	37.7	5.40
10	35.0	34.3	5.06
15	27.5	32.0	4.84
20	21.5	29.8	4.66
25	16.5	28.3	4.52
30	12.3	27.0	4.39
35	8.6	25.8	4.28
40	5.8	24.7	4.18

Table 9-26 gives typical values of steam and cooling-water rates for several absorber pressures with fixed cooling-water temperatures.

Industrial absorption refrigeration machines have been designed and operated for refrigeration loads from 100 to 2,000 tons. Where suitable cooling water is available, usually below 85°F, water may be used in series, passing first through condenser and then absorber, realizing an appreciable saving in pumping costs, etc.

Two-stage absorption machines may be used where refrigeration is required at different temperature levels, as well as for single low-temperature loads with heat source below that allowable for single-stage operation.

The absorption machine should not be overlooked where waste heat is available. Unlike reciprocating compression machines, which are limited by volumetric displacement, the absorption machine can maintain capacity at lower back pressures by increasing the flow of heat to the generator. A further advantage is that the absorption machine may be located outdoors, requiring no special building. With the application of suitable controls, these machines operate with a very small amount of attention.

ABSORPTION AND ADSORPTION DEHUMIDIFYING SYSTEMS

BY GILBERT A. KELLEY

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Dehumidification is the removal of moisture from air or other gases. The term "dehumidification equipment" is usually applied to equipment whose primary function is removal of moisture from a gas, usually air. This is a contrast to the conventional refrigeration systems where moisture is removed from a gas by condensation simultaneously with the reduction in sensible heat of the gas by spray or surface cooling with the cooling medium below the dew point of the gas. *There are two general types of dehumidifying systems, viz.*, absorption systems and adsorption systems.

Absorption systems are those in which moisture is removed from a gas with a change, either physical or chemical or both, of the absorption medium during the process. The absorption medium may be either solid or liquid. Calcium chloride is an example of a solid absorbent, and aqueous solutions of lithium chloride, lithium bromide, calcium chloride, and solutions of the ethylene glycols are examples of liquid absorbents.

Adsorption systems are those in which moisture is removed from a gas without any physical or chemical change of the adsorbing medium during the process Silica gel, activated alumina, and activated bauxites are examples of solid materials having this adsorbing property.

ABSORPTION SYSTEMS

Principle of Operation. The liquid absorbent or hygroscopic solution has the property of absorbing moisture from, or adding moisture to, the air depending upon the vapor-pressure difference between the air and the solution. The equilibrium vapor pressure of the solution is dependent upon the temperature and the concentration of the solution. The liquid absorbent is capable of either dehumidifying or humidifying the air, depending upon the requirements on the system, by control of the temperature or the concentration of the solution.

Air saturated in contact with an aqueous solution of lithium chloride will contain approximately one-eighth as much water as air saturated in contact with water. In other words, air in equilibrium with a concentrated aqueous solution of lithium chloride would have a relative humidity of approximately 12 per cent since the relative humidity of the air is the amount of moisture in the air compared to the amount of moisture the air would contain if it were saturated with water vapor at the same temperature.

In most of the commercial equipment of this type, it is usual practice to maintain the concentration of the solution constant for a given application and to cool the solution where it is desired to remove moisture from the air and to heat the solution when it is desirable to add moisture to the air.

The heat of absorption, owing to moisture being absorbed from the air by an absorbent liquid, is given up to the surface of the liquid. This heat will cause the absorbent liquid to rise in temperature so that it is necessary to cool it. Otherwise the whole body of absorbent liquid would rise in temperature to such a point that the loss of the sensible heat to the air by conduction and radiation would equal the gain in latent-heat absorption of the water vapor by the liquid.

As contrasted with adsorption systems, the dry-bulb temperature of the air from an absorption dehumidifier is not a function of the amount of water removed from the air but is dependent upon the temperature of the absorbent liquid. If the temperature of the absorbent liquid is lower than the dry-bulb temperature of the air entering the dehumidifier, the air will be cooled as well as dehumidified in passing through the dehumidifier. If the temperature of the liquid absorbent is higher than the dry-bulb temperature of the air, the air will tend to increase in temperature while passing through the dehumidifier.

Liquid-absorbent Summer Operating Cycle. The operating cycle of a liquid-absorption system with an external solution cooler and heater is shown in Fig. 9-9 For summer operation the strong hygroscopic solution is pumped from the sump of the unit through a solution cooler. The solution is continuously cooled in the heat exchanger with city, well, refrigerated, cooling-tower water, or in some instances with an evaporative cooler. The quantity of water required is a function of the water temperature and the total heat, either sensible, latent, or both, removed from the air by the hygroscopic solution.

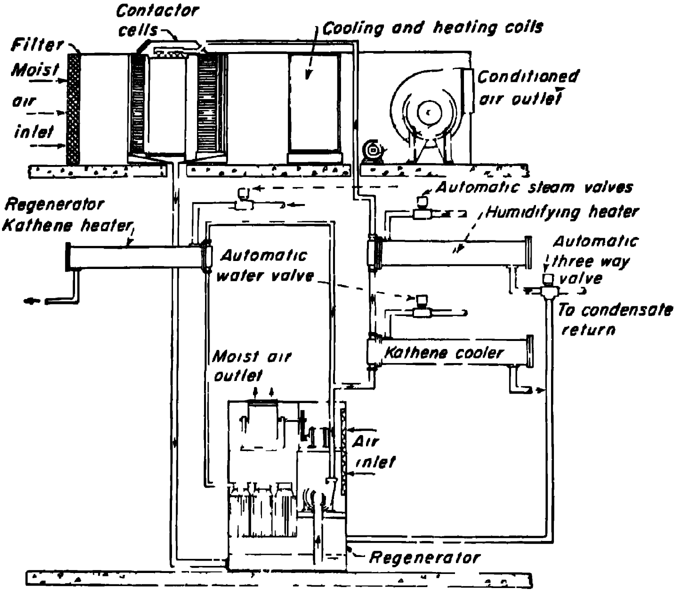


FIG. 9-9. Flow chart for liquid-absorbent system with external liquid cooler and heater.

The hygroscopic solution at the required temperature and concentration is brought into intimate contact with the air to be dehumidified in the contactor chamber of the unit The moisture is absorbed from the air by the solution owing to the vapor-pressure difference between the air and the solution The amount of water vapor removed from the air is usually controlled by automatically regulating the flow of water through the cooler with a modulating water valve

The dry-bulb temperature of the air leaving the liquid-absorbent contactor is dependent upon the temperature of the liquid absorbent and the amount of contact surface between the air and solution. In most commercial equipment, the dry-bulb temperature of the air leaving the dehumidifier will be within 1 to 5° of the liquid-absorbent temperature. If the temperature of the absorbent liquid is lower than the dry-bulb temperature of the air entering the dehumidifier, the air will be cooled as well as dehumidified in passing through the dehumidifier If the temperature of the liquid absorbent is higher than the dry-bulb temperature of the air, the air will increase in temperature.

Cooling and heating coils are installed in the duct work if cooling or heating of the air from the dehumidifier is required.

In absorbing moisture from the air, a certain amount of heat is generated which is the heat of condensation of the water vapor in the solution. This consists of the latent heat of condensation of water vapor and the heat of solution or the heat of mixing. The heat of mixing varies with the liquid absorbent used and with the concentration and temperature of the absorbent. It is 120 Btu per lb of water vapor absorbed for an aqueous solution of lithium chloride at 95°F temperature and at the concentration used for air-conditioning applications.

The temperature of the liquid absorbent will increase owing to the heat of condensation of the water vapor absorbed from the air by the solution. The concentration of the liquid absorbent is lowered because of the dilution of the liquid with water absorbed from the air. The warm dilute solution flows from the contactor cells to the sump.

The liquid absorbent is maintained at the proper concentration for moisture removal by automatically removing water from the liquid. A small percentage of the solution, usually 10 to 20 per cent of the flow to the contactor cells, is passed through a heater where the liquid is heated to 200 to 235°F, depending upon the steam pressure available. At this temperature the vapor pressure of the liquid absorbent is considerably higher than that of outside air. The hot solution at the relatively high vapor pressure is then contacted with outside air in the regenerator, where water is absorbed from the solution by the outside air owing to the vapor-pressure difference between the outside air and the hot solution. The solution is cooled as well as concentrated in the regenerator because of the absorption of water from the solution and the sensible cooling effect of the outside air. The process in the regenerator is the reverse of the absorption process in the contactor cells where the air is dehumidified. The hot moist air from the regenerator is discharged to the outside.

The majority of the absorption dehumidifier systems are controlled by a float control in the sump of the unit. This control holds a constant level of liquid in the sump. Since no salt is added or removed, a constant level or volume in the system represents a constant concentration. The volume of liquid absorbent in the unit and the level of the liquid in the sump will rise as water is absorbed from the air. This rise in level actuates the float which opens the steam valve and admits steam to the regenerator heater to remove water from the solution until the control level is reached. The float then turns off the steam to the regenerator heater, and no further regeneration takes place until there is another rise in level.

The hot concentrated liquid absorbent from the regenerator mixes in the sump with the warm diluted liquid from the contactor cell. The mixture, which is at the proper concentration for dehumidification, is pumped through the cooler where the cycle is repeated.

Liquid-absorbent Winter Operating Cycle. The operating cycle of a liquid-absorption system for winter operation is shown in Fig 9-9. The hygroscopic solution is pumped through a humidifying-solution heater where the solution is continuously heated in proportion to the humidifying load.

The hygroscopic solution at the required temperature and concentration is brought into intimate contact with the air to be humidified. The moisture is given up to the air by the solution owing to the difference in vapor pressure between the air and the solution. The amount of water vapor added to the air is usually controlled by automatically regulating the flow of steam through the heater with a modulating steam valve. The water valve for summer operation is automatically closed when humidification is required.

The temperature of the solution will decrease because of the heat of evaporation of the water absorbed by the air from the solution. The concentration of the solution will increase owing to the removal of water from the solution. The cool concentrated solution flows from the contactor cells to the sump.

The solution is maintained at the proper concentration by adding steam condensate or distilled water to the solution to replace the water given up to the air. The addition of city or well water to the chloride or bromide solutions is not recommended because of the possible effects of minerals in the water.

The density control for humidifying operation controls the addition of water to the solution by operating a valve in the make-up water line to maintain a constant concentration. The condensate from the humidifying heater is usually used for make-up water as shown in Fig. 9-9. The density control operates the three-way valve to pass condensate into the sump when make-up water is required and to pass the condensate into the condensate-return line when no make-up water is required.

Liquid-absorbent Equipment. *Absorbents.* Absorbents for liquid dehumidifying systems should have the following characteristics:

1. Suitable vapor-pressure characteristics. The absorbent should not crystallize at temperatures 10 to 15° below the temperature range used in the operating cycle
2. Noncorrosive
3. Odorless
4. Stability. It will not break down over range of use or vaporize
5. Low viscosity, good heat-transfer characteristics
6. Nontoxic and noninflammable
7. Capable of being regenerated at temperatures obtainable with l-p steam
8. Widely available and economical to manufacture

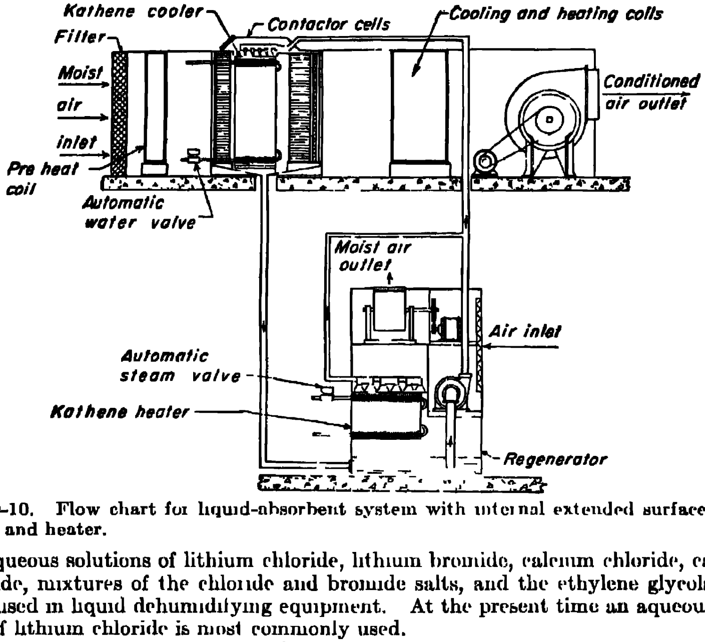


FIG. 9-10. Flow chart for liquid-absorbent system with internal extended surface liquid cooler and heater.

Aqueous solutions of lithium chloride, lithium bromide, calcium chloride, calcium bromide, mixtures of the chloride and bromide salts, and the ethylene glycols have been used in liquid dehumidifying equipment. At the present time an aqueous solution of lithium chloride is most commonly used.

Contactors. The contactor is that part of the apparatus in which the air to be dehumidified is brought in contact with the liquid absorbent. The contactor should have a high water-removal efficiency per vapor-pressure difference between the air and absorbent with a relatively low resistance to the flow of air. It should be relatively light, compact, and inexpensive. It should retain its original characteristics after extended use.

The contactor may consist of an atomized spray of the liquid absorbent, but most equipment has a contact surface providing a large area of the absorbent exposed to the air. This extended surface contactor provides higher water-removal efficiency per volume of space and eliminates atomization of the liquid. This simplifies the removal of entrained drops of the absorbent from the air stream.

An extended surface fin coil is used as the contactor in most of the standard absorbent dehumidifiers. The cooling medium, usually water, flows inside the tubes, and the air and liquid absorbent flow over the external surface of the coil as shown in Fig. 9-10. This arrangement combines the plate-contact surface and the cooler shown in Fig. 9-9 in one assembly. This arrangement is less expensive, more compact, and is more adaptable for packaged units assembled in the factory.

The plate-contact surface usually consists of a series of parallel plates made of metal or glass. The air passes between the plates in contact with the liquid flowing over the surface of the plates. The solution is precooled as shown in Fig. 9-9. This type of contact surface is usually used in heavy-duty industrial applications in which the individual parts of the system are built at the factory and installed in the field

Regenerators. The regenerator is that part of the apparatus in which water is removed from the liquid absorbent. The regenerator requires the same performance and physical characteristics as the contactor.

An extended surface fin coil is used as the regenerator in most of the standard absorbent dehumidifiers. The heating medium, usually steam or hot water, flows inside the tubes, and the liquid absorbent and air flow over the external surface of the coil as shown in Fig 9-10. This arrangement combines the plate-regenerator contact surface and the solution heater shown in Fig. 9-9 into one assembly

Some heavy-duty regenerators are constructed with a series of parallel plates

Table 9-27. Unit Specifications*

Cfm	Resistance to air flow, in water gauge		Pump hp	Regenerator fan hp
	100% flow	50% flow		
15,400	2.0	0.6	3	†
11,000	2.0	0.6	3	†
5,000	1.9	0.5	3/4	3/4
3,500	1.9	0.5	3/4	3/4
2,500	1.9	0.5	3/4	1/2
1,350	1.9	0.5	3/4	1/2

* Courtesy Surface Combustion Corporation.

† Regenerator in the 11,000- and 15,400-cfm units are separate assemblies and selected for the water removal.

Table 9-28. Regenerator Specifications*

Model No	132	115	101	94	66	52	47	33	26
Cfm regenerator air	4,650	4,400	4,000	3,620	2,540	2,000	1,810	1,270	1,000
Regenerator air fan motor hp	3	3	3	3	3	2	2	2	2
Water-removal capacity, lb per hr at 25 lb gauge steam pressure	1,320	1,180	1,040	940	660	520	470	330	260

* Courtesy Surface Combustion Corporation

Table 9-29. Water-removal Capacity at Various Steam Pressures*

Gauge steam pressure, lb	25	20	15	13	10	5	3
Removal capacity, per cent	100	86	76	71	66	55	47

* Courtesy Surface Combustion Corporation.

Table 9-30. D Factor—Heat from Regenerator, 1000 Btu per Hour*

Inlet air grains/lb	Outlet air, grains/lb											
	60	55	50	45	40	35	30	25	20	15	10	8
130	7.33	7.68	8.05	8.34	8.8	9.28	9.9	10.58	11.6			
120	7.25	7.55	7.9	8.25	8.65	9.13	9.7	10.38	11.3			
110	7.15	7.45	7.78	8.2	8.55	9.0	0.55	10.2	10.98			
100	7.06	7.34	7.66	8.1	8.42	8.85	9.36	10.0	10.65	12.1		
90	7.05	7.26	7.6	8.0	8.35	8.75	9.24	9.8	10.48	11.75		
80	6.95	7.2	7.55	7.94	8.27	8.65	9.1	9.64	10.25	11.4		
70	6.9	7.15	7.5	7.85	8.2	8.52	8.95	9.48	10.08	11.1	12.0	
40						8.2	8.54	9.04	9.55	10.15	11.15	
20						.	.		...	9.85	10.55	12.0

* Courtesy Surface Combustion Corporation

NOTE: Heat from regenerator is *D* factor × regenerator model number. Example for No 66 regenerator Btu per hour from regenerator = *D* factor × 66.

made of glass or metal for the contact surface. A solution heater as shown in Fig. 9-9 is required to preheat the solution with this type of surface.

Pumps. Vertical self-priming centrifugal pumps with the pump casing immersed in the liquid absorbent are used in most commercial absorbent dehumidifiers.

Equipment Ratings. Figure 9-11 and Tables 9-27 to 9-33 present equipment ratings of lithium chloride equipment commercially available and operating according to the basic cycle shown in Fig. 9-10.

Table 9-31. Maximum Steam Demand, Pounds of Steam per Pound of Water Removed*

Exit grains/lb	Steam pressure						
	3	5	10	12½	15	20	25
60	3.18	3.1	2.9	2.8	2.7	2.55	2.4
50	3.29	3.2	2.97	2.86	2.75	2.6	2.45
40	3.4	3.3	3.05	2.92	2.8	2.65	2.5
30	3.51	3.4	3.13	2.98	2.85	2.7	2.55
20	3.62	3.5	3.2	3.05	2.9	2.75	2.6

* Courtesy Surface Combustion Corporation.

Table 9-32. Dry-bulb Temperature of Outlet Air with 95°F Inlet-air Temperature*

Inlet air grains/lb	Outlet air grains/lb												
	60	55	50	45	40	35	30	28	26	24	22	20	18
130	113	111	106	101	96	91	84	80	76	72	66	61	
120	115	113	108	102	97	92	85	82	78	75	69	64	
110	117	115	109	103	98	93	86	83	80	77	72	67	
100	118	116	110	104	99	94	88	85	82	79	75	70	63
90	119	117	111	105	100	95	90	86	83	81	78	71	67
80	120	118	112	106	101	96	91	88	84	83	79	73	70
70			113	107	102	97	92	90	85	84	81	76	74
60				109	103	99	93	92	89	85	84	78	76
50					105	100	95	93	90	86	85	80	77
40								95	91	87	86	82	78

* Courtesy Surface Combustion Corporation.

Note. Deduct 1°F from outlet air dry-bulb temperature for 85°F inlet air. Deduct 2°F from outlet air dry bulb temperature for 75°F inlet air. Deduct 3°F from outlet air dry bulb temperature for 65°F inlet-air.

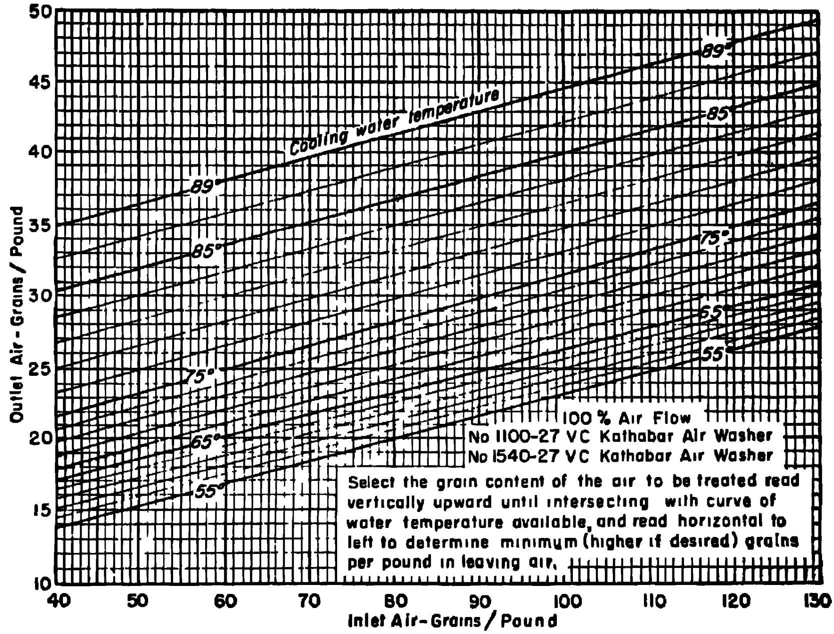
Table 9-33. Gpm of Water per 1000 Btu per Hour of Heat Absorbed by Water*

Water temperature deg F	Outlet air grains/lb												
	60	55	50	45	40	35	30	28	26	24	22	20	18
89	0.120	0.125	0.130	0.135	0.139								
85	0.077	0.082	0.087	0.092	0.120	0.170							
80	0.066	0.070	0.074	0.078	0.090	0.110	0.160	0.180					
75	0.062	0.064	0.066	0.068	0.070	0.074	0.110	0.122	0.140	0.170			
70	0.057	0.060	0.061	0.063	0.065	0.067	0.080	0.093	0.110	0.130	0.150		
65	0.055	0.056	0.057	0.058	0.059	0.060	0.065	0.073	0.082	0.092	0.105	0.120	0.140
60	0.049	0.050	0.051	0.052	0.054	0.055	0.057	0.062	0.068	0.080	0.092	0.105	0.119
55	0.043	0.044	0.045	0.046	0.047	0.048	0.050	0.053	0.056	0.067	0.078	0.091	0.101

* Courtesy Surface Combustion Corporation.

Note. For parallel coil arrangement multiply above values by 1.45.

Selection of Lithium Chloride Equipment. *Given:* a five-story office building in Houston, Texas, with 595,150 cu ft of space to be air conditioned. There are 685 occupants in the building



Select the grain content of the air to be treated read vertically upward until intersecting with curve of water temperature available, and read horizontal to left to determine minimum (higher if desired) grains per pound in leaving air.

FIG 9-11 Performance curve of lithium chloride dehumidifier (*Surface Combustion Corporation*)

A *Design data*

1 Outside design
   - *a* 130 grains of moisture per lb of dry air for latent load
   - *b* 95°F DB for sensible load

2 Inside design—80° DB and 50 per cent relative humidity, 77 grains per lb. absolute humidity

3 Utilities
   - *a* 85° cooling-tower water
   - *b* 20 psi steam pressure

4 216,000 Btu hr—internal latent load from people and infiltration

5 Fresh-air requirements
   - *a* 15 cfm of fresh air per person has been specified for ventilation.

$$15 \times 685 = 10{,}275 \text{ cfm fresh air for ventilation}$$

B *Cfm through the dehumidifier*

In general, the dehumidifier is applied to the fresh air only for comfort installations of this type The general pattern is to place the dehumidifier in a penthouse of the building to be air conditioned and to discharge the dehydrated air to individual cooling units The entire external and internal latent load is ordinarily removed in the dehumidifier

1 Try removing the internal and external latent-heat loads with all fresh air through the dehumidifier

$$\frac{216{,}000}{0.68^{*} \times 10{,}275} = 31.0 \text{ grains per lb difference required between maintained air and fresh air from the dehumidifier}$$

From performance curve Fig. 9-11, for lithium chloride dehumidifier with 85° water, the air will be reduced from 130 grains per lb to 45 grains per lb. Since the entire latent load can be absorbed by the fresh air passing through the dehumidifier, the 11,000-cfm unit is selected to handle 10,275 cfm of fresh air. If the required outlet-air grains were lower than could be obtained with the dehumidifier, it would be necessary to obtain a lower cooling-water temperature or mix some recirculated air with the fresh air and pass more air through the dehumidifier in order to assume the latent load.

In this problem, the calculations will be based on reducing 10,275 cfm of air from 130 to 45 grains per lb.

C. Dry-bulb temperature of outlet air

From Table 9-32, the outlet-air dry-bulb temperature from the dehumidifier will be 101°F.

D. Regenerator selection

$$\frac{10{,}275 \times 0.075 \times 60 \times (130 - 45)}{7{,}000} = 560 \text{ lb per hr water removed from the air}$$

The regenerator capacity for 20 lb steam pressure from Table 9-29 is 86 per cent of the rated capacity.

$$\frac{560}{0.86} = 651 \text{ lb per hr rated selection required for 560 lb per hr removal at 20 lb steam pressure}$$

From Table 9-28 select a No 66 regenerator which will remove 660 lb per hr of water at 25 lb steam pressure. At 20 lb steam pressure, the water-removal capacity will be 0.86 × 660 = 568 lb per hr.

E. Total heat to be removed by the 85° water

1. Sensible heat. From step *C* the air enters dehumidifier at 95° and leaves at 101°.

$$10{,}275 \times 1.08^{\dagger} \times (101 - 95) = 66{,}600 \text{ Btu/hr sensible-heat addition to the air}$$

2. Latent heat. From step *D* 560 lb per hr of water are removed from the air.

$$560 \times 1{,}170 = 655{,}200 \text{ Btu/hr latent heat to the water}$$

NOTE. Average latent heat of condensation of 1 lb of water in lithium chloride is 1170 Btu for concentration used in air-conditioning equipment.

3. Heat load from regenerator. From Table 9-30 the *D* factor is 8.34 × 1000 Btu per hr. The heat load from the regenerator will be

$$8.34 \times 1000 \times 66 = 550{,}000 \text{ Btu per hr.}$$

655,200 Btu per hr- latent heat from air
550,000 Btu per hr - heat load from regenerator
1,205,200 Btu per hr.
66,600 Btu per hr sensible heat to the air
1,138,600 Btu per hr— total heat to be removed by the cooling-tower water

---

* Factor $0.68 = \frac{0.075 \times 60 \times 1.055}{7{,}000}$ from formula

Latent Btu/hr $= \frac{\text{cfm} \times \text{specific volume } (0.075) \times 60 \times 1{,}055 \times \text{grains per lb removed}}{7{,}000 \text{ (grains in 1 lb)}}$

77 grains per lb in space
31 grains per lb required grain difference for 10,275 cfm of air
46 grains per lb required outlet air from dehumidifier to remove total latent load.

† Factor 1.08 = 60 × 0.075 × 0.24 from formula

‡ Sensible Btu per hour = cfm × 60 × 0.075 × 0.24 × temperature difference

F. Operating data

1. Gpm of cooling-tower water. From Table 9-33, for 85° water and 45 grains per lb outlet air, the gpm of water per 1,000 Btu per hr of heat absorbed by the water is 0.092. From step *E*, the total heat to the water is 1,138,600 Btu per hr.

$$0.092 \times 1{,}138.6 = 104.8 \text{ gpm of } 85^\circ \text{ water}$$
$$1{,}138{,}600/8.34 \times 60 \times 104.8 = 21.8^\circ \text{ water-temperature rise}$$

2. Steam requirements. From Table 9-31 the maximum steam demand is 2.63 lb of steam per lb of water removed from the air for 20 lb steam pressure.

$$560 \times 2.63 = 1{,}475 \text{ lb per hr of steam for maximum load.}$$

Note: The steam rate is proportional to the load. At half load the steam requirements will be 50 per cent of the steam rate for full load.

3. Electric requirements. A 3-hp motor for the pump from Table 9-27 and a 3-hp motor for the regenerator fan from Table 9-28.

G. Humidifying operation for winter cycle

1. Fresh air from the lithium chloride dehumidifier to be discharged to the local zone units at 72° DB and 30 grains per lb.
2. The air will be preheated in a preheating coil to 89° DB. All the heat for humidification is supplied in this type of unit with the preheating coil. Assuming outside air at 0° DB, the air from the preheating coil will be 89° DB—53.3° WB. Neglecting the radiation and conduction losses, the air from the lithium chloride humidifier will be 72° DB and 30 grains per lb. The amount of humidification is controlled by automatically regulating the steam input to the preheating coil.

H. Sensible-heat removal for summer operation

The internal and external (fresh-air) latent loads are removed with the dehumidifier. The cooling coils for the individual-zone cooling units will be selected for sensible heat removal only and will operate without condensation of water vapor on the coil surface. The refrigeration system will be selected in the conventional manner for the sensible cooling load. The flow chart for this problem is shown in Figure 9-12.

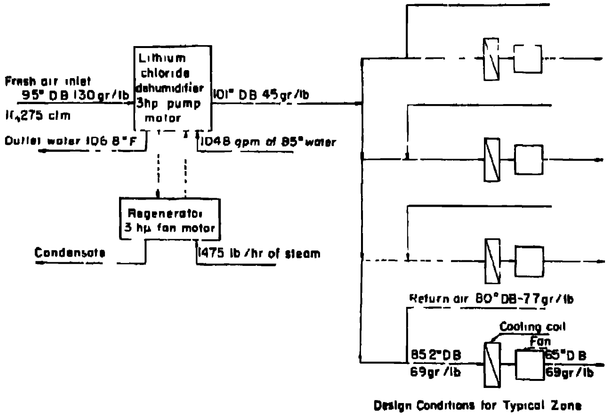


Fig. 9-12. Flow diagram for problem using liquid-absorbent dehumidifier.

Control of Absorbent Dehumidifiers. There are two basic control methods used with absorbent dehumidifying systems, *viz.*, constant solution temperature and variable solution temperature.

Constant Solution Temperature. The liquid-absorbent temperature is held constant regardless of the load imposed on the unit. The temperature is such that, at full load on the unit, the air emerging will be at the desired moisture content. At any load less than 100 per cent, the air from the unit will be dryer than the required design conditions. The density control inside the unit maintains the absorbent concentration constant.

In general, constant solution-temperature control is suitable for comfort conditioning and for process work where a lowering in leaving air dryness from the liquid-absorbent unit is not important. The majority of dehumidifying applications are of this type.

Constant solution-temperature external controls for this control method are as follows:

1. Immersion thermostat in the liquid absorbent which is set at the proper control temperature. For installations such as shown in Fig. 9-9, this thermostat would be placed between the solution cooler and the contactor cells. For an installation in which the contactor unit and the cooler are combined in one assembly, as shown in Fig. 9-10, the immersion thermostat would be placed in the path of the solution from the extended surface contactor.
2. Modulating valve in the water line to the solution cooler. If humidification is required, a modulating valve is also placed in the steam line to the humidifying heater. The two valves are controlled by the immersion thermostat.

Variable Solution Temperature. The liquid-absorbent temperature is varied over a range with varying loads so that, regardless of the load imposed on the absorbent unit, the air to be controlled will be at a constant moisture condition. In general, variable solution temperature is suitable for process and industrial applications where a constant condition of moisture must be maintained and where any fluctuation would be detrimental.

The following controls are required for this control method:

1. A master wet-bulb stat, humidistat, or dew-point control installed at the control point.
2. A submaster immersion solution thermostat placed in the liquid absorbent The location would be the same as for the immersion thermostat under constant solution-temperature control
3. Modulating valve in the water line to the cooler and a modulating steam valve in the steam line to the heater.

The master wet-bulb thermostat with this control arrangement resets the submaster liquid-absorbent-temperature thermostat to vary the absorbent temperature in accordance with the load. It is important with this type of control always to include a submaster thermostat in the control circuit. In addition to the above absorbent external controls, a dry-bulb thermostat should be placed at the control point for most applications to control the dry-bulb temperature through an afterheating coil or an aftercooling coil, or both, if required The solution concentration si maintained at a constant value with the internal density control.

ADSORPTION SYSTEMS

In an adsorption system, water vapor is condensed on the surface of the adsorbent and drawn into the interior of the adsorbent by capillary attraction. The adsorbent continues to adsorb water vapor from the air until an equilibrium condition is reached where the internal vapor pressure of the water in the pores of the material is in equilibrium with the vapor pressure of the air passing through the material.

For a given adsorbent material, the moisture adsorption is dependent upon the adsorbent-bed temperature, the moisture content of the air or gas to be dehumidified, and the flow rate through the adsorbent bed.

The air passing through the adsorbent dehumidifier increases in temperature owing to the conversion of latent heat to sensible heat resulting in a rise in temperature of the air. The rise in temperature of the air passing through the dehumidifier is somewhat higher than the theoretical rise computed from the transformation of latent heat into sensible heat This extra heat is from the residual heat in the bed and structural parts owing to the reactivation process. An approximate figure for the rise in the temperature of the dehumidified air caused by the adsorption heat is 0.75°F rise for each grain of moisture removed per pound of air. The actual temperature rise of the air in the adsorbent dehumidifier for a specific application should be obtained from manufacturer's rating tables.

Additional capacity may be obtained from the adsorbent dehumidifier by precooling the air before it goes through the dehumidifier when the air to be dehumidified is warm. Precooling of the air serves to lower the adsorbent-bed temperature and thus increase the performance. Some manufacturers install cooling coils in the adsorbent beds or stage the adsorbent bed with a cooling coil between two beds in order to obtain additional capacity and lower the working temperature of the bed.

The moisture is removed from the adsorbent by heating the bed to approximately 300 to 450°F. The exact temperature of the air for reactivation depends upon the type of adsorbent used. The reactivation air containing the water given up by the adsorbent is discharged to the outdoors. After the water has been removed from the adsorbent, outdoor air is usually passed over the bed to reduce the temperature of the bed before it is switched to the dehumidifying cycle.

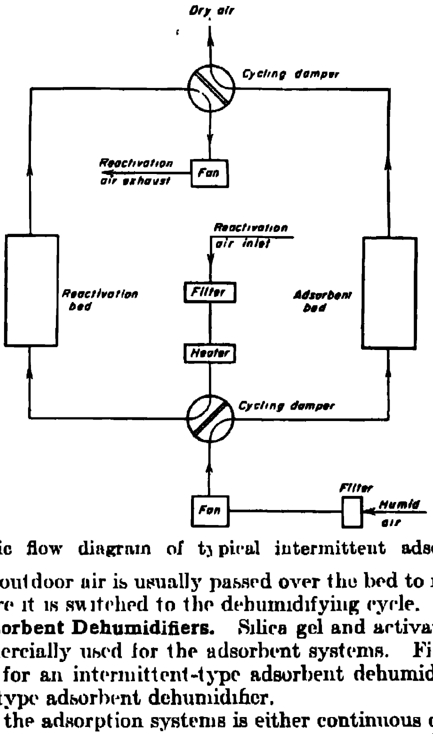


FIG. 9-13. Schematic flow diagram of typical intermittent adsorbent dehumidifier.

Operation of Adsorbent Dehumidifiers. Silica gel and activated alumina are the two materials commercially used for the adsorbent systems. Figure 9-13 is a schematic flow diagram for an intermittent-type adsorbent dehumidifier, and Fig. 9-14 shows a continuous-type adsorbent dehumidifier.

The operation of the adsorption systems is either continuous or intermittent. In order to provide continuity of moisture removal, there are two adsorbent beds in the intermittent equipment, as shown in Fig. 9-13. While one bed is being reactivated and purged, the other bed or adsorbing bed is removing moisture from the air. After the adsorbing bed has adsorbed water vapor to the point where it will not continue efficiently to remove moisture from the air, dampers are switched in the air stream so that the adsorbing bed which now contains a relatively large amount of water vapor is put on the reactivating cycle. At the same time, the air to be dehumidified is switched to the other bed which has been reactivated and purged. This is accomplished automatically in most equipment with time relays.

In a continuous-adsorption dehumidifier, the unit is separated into two compartments with dividing seals located on the inside and the outside of the drum. Figure 9-14 shows the details of construction of a continuous-adsorbent dehumidifier. The air entering the right-hand portion of the drum is dehumidified and discharged through the right-hand fan, while at the same time the left-hand portion of the drum is being reactivated.

Water vapor is adsorbed by a given particle of the adsorbent as it rotates counterclockwise from *Z* to *X*. The adsorbent particle moves into the reactivation compartment at position *X* where the water vapor previously adsorbed is driven out of the material.

The adsorbent in the reactivation compartment is in contact with air heated by the gas burner. The products of combustion from the burner are mixed with the air drawn in through the reactivation air inlet. The temperature of the mixture is high enough to drive the water vapor from the adsorbent. This temperature is 325 to 350°F for the silica gel equipment of one manufacturer. The water vapor is discharged to the outdoors by the reactivation fan. High-pressure steam may be used for heating the reactivation air by placing steam coils in the reactivation air supply to the unit.

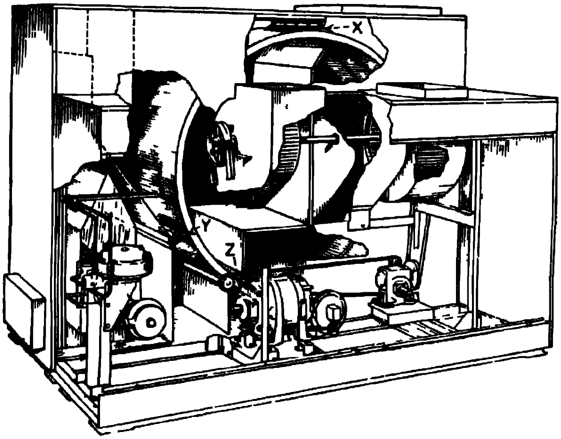


FIG. 9-14. Rotary-type silica gel adsorbent dehumidifier. (*Bryant Heater Company*)

The unit as shown in Fig. 9-14 is designed so that the water vapor has been driven out of the adsorbent when it reaches point *Y*. Outdoor unheated air is pulled through the segment *YZ* by the reactivation fan to reduce the temperature of the bed before it enters the adsorption compartment.

Equipment Ratings. Figures 9-15, 9-16, and 9-17, represent equipment ratings of standard silica gel adsorbent units commercially available.

Selection of Silica Gel Dehumidifier. *A. Problem.* A laboratory 72 by 30 by 12 ft is to be air-conditioned. Ground floor is concrete. One outside wall is constructed of 13 ft brick and plaster with three coats of vaporproof paint. All windows are sealed and caulked. Inside walls and ceiling are coated with three coats of vaporproof paint on both sides. One door 6 by 8 ft is to be air-locked. There are 10 people in the laboratory. One pound per hour of water is evaporated in the laboratory.

B. Design data

1. Outside design
   a. 122 grains of moisture per lb of dry air for latent load
   b. 95° DB, for sensible load
2. Inside design
   85° DB and 28 per cent relative humidity—52 grains per lb
3. Utilities
   a. 85° water for the dehumidifier
   b. 1000 Btu natural gas

4. 11,900 Btu per hr, internal latent load from people, infiltration, and product
5. Fresh air
   a. $72 \times 30 \times 12$ ft = 25,920 cu ft volume of space

$$25{,}920/60 = 432 \text{ cfm} = 1 \text{ air change per hr}$$

One and one-half changes per hr of fresh air will be discharged to the space.

$$1.5 \times 432 = 648 \text{ cfm of fresh air, say } 650 \text{ cfm}$$

The amount of fresh air discharged to the space through the dehumidifier is usually estimated from experience.

C. Cfm through dehumidifier

In applications of this type, it should first be checked to see if the latent load can be removed by passing fresh air through the dehumidifier. If the entire internal and external latent-heat loads cannot be removed with fresh air, some recirculated air is mixed with the fresh air, and the mixture is circulated through the dehumidifier.

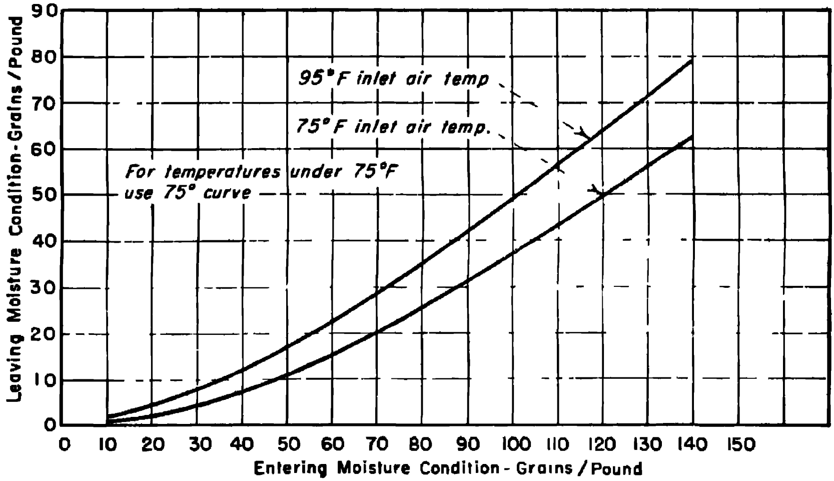


Fig. 9-15. Effluent moisture curve for rotary adsorbent dehumidifier. (*Bryant Heater Company.*)

With air entering at 122 grains, the outlet air from Fig. 9-15 will be higher than the maintained condition of 52 grains per lb, so it will be necessary to treat a mixture of fresh and return air through the dehumidifier.

Try 1,300 cfm of air through a No. 14 unit. The mixture of 650 cfm fresh air and 650 cfm return air will be at 90° DB and 87 grains per lb. From Fig. 9-15 the outlet air will be 37.5 grains per lb.

$$11{,}900/0.68^{*} \times 1{,}300 = 13.5 \text{ grains/lb rise on } 1{,}300 \text{ cfm of air}$$

52 grains/lb = maintained condition

13.5 grains/lb = required diffusion grain/pound difference between maintained condition and air from dehumidifier with 1,300 cfm of air

38.5 grains/lb = required air from dehumidifier

Since the dehumidifier will deliver air at 37.5 grains/lb, use No. 14 unit with 1,300 cfm of air. If the moisture removal could not have been obtained with 1,300 cfm of air, it would be necessary to use more return air and a larger unit or precool the fresh air.

D. Dehumidifier after-cooling coil selection

From Fig. 9-16 the air from the adsorbent dehumidifier will be 134°F for air entering at 90° DB and 87 grains per lb. The cooling coil is selected in the conventional manner. A 7° approach to the water temperature would be an economical selection. 1,300 × 1.08* × (134 − 92) = 58,968 Btu per hr to be removed by the water. Assuming a 14° temperature rise, the water quantity would be 8.4 gpm of 85° water, leaving the coil at 99°.

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* See page 666 for derivation of factor 0.68.

E. Gas consumption

From Fig. 9-17 the full-load gas consumption is 145,000 Btu per hr. From Fig. 9-17 with 87 grains per lb inlet air at 90° DB, 85 per cent of the full-load gas consumption will be required.

$$0.85 \times 145{,}000 = 123{,}250 \text{ Btu/hr gas consumption}$$

F. Operating data

8.4 gpm of 85° water from step *D*

123,250 Btu/hr, gas consumption from step *E*

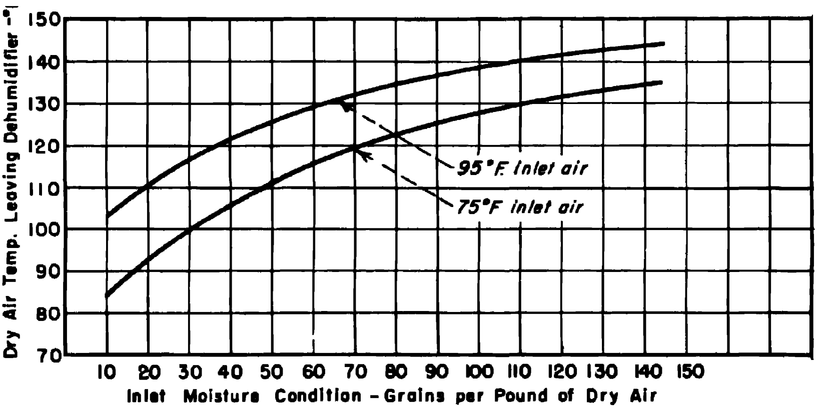
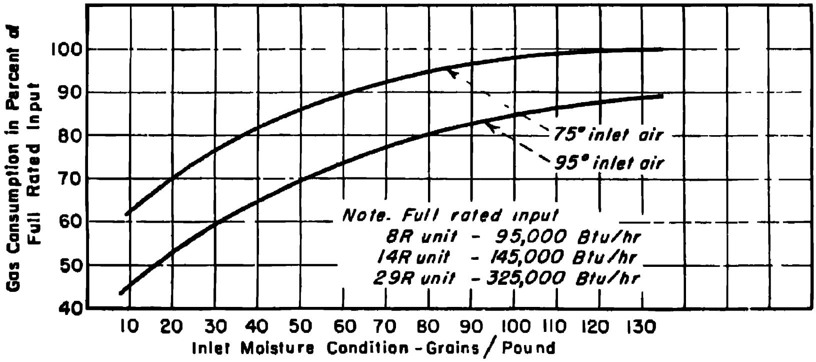


FIG. 9-16. Temperature of effluent dry air from rotary adsorbent dehumidifier. (*Bryant Heater Company.*)



Note. Full rated input
8R unit - 95,000 Btu/hr
14R unit - 145,000 Btu/hr
29R unit - 325,000 Btu/hr

FIG. 9-17. Gas consumption on rotary adsorbent dehumidifiers. (*Bryant Heater Company.*)

A 1-hp motor is supplied with this unit for the fan handling the main air supply and the reactivation air. The motor size varies with the depth of the bed and the air flow. The flow chart for this problem is shown in Fig. 9-18.

Control of Adsorbent Dehumidifiers. *Variable Control of Reactivation of Steam-fired Adsorbent Dehumidifiers.* A wet-bulb thermostat, humidistat, or dew-point stat is placed at the control point to operate a modulating damper in the reactivation air supply to the dehumidifier. The moisture-removal capacity of the dehumidifier is controlled by regulating the amount of activation-heat input into the dehumidifier by controlling the volume of reactivation air with a modulating damper control. The dehumidifier operates continuously with this arrangement, and the equipment is modulated to meet the varying moisture-removal loads.

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* See page 666 for derivation of factor 1.08.

The control equipment required consists of the following:

1. Wet-bulb thermostat, humidistat, or dew-point thermostat placed at the control point
2. Damper with modulating damper motor in the reactivation air supply between the steam-reactivation coil and the dehumidifier under control of the humidistat

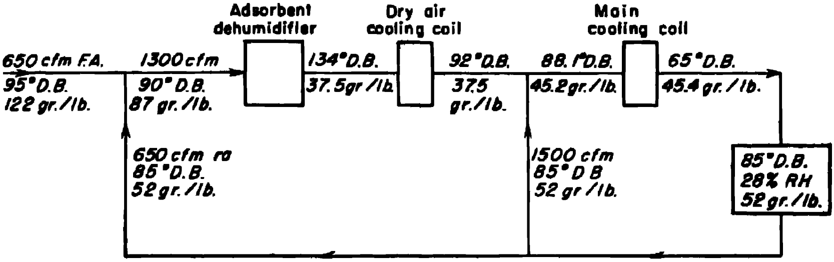

FIG. 9-18. Flow chart for adsorbent dehumidifier problem.

Variable Control of Reactivation of Gas-fired Adsorbent Dehumidifiers. The humidistat with the gas-operated dehumidifier controls the gas input to the dehumidifier. The operation and controls are the same as the above with the exception that the modulating control is placed on the gas input to the dehumidifier, and the modulating damper in the reactivation air supply is eliminated.

Face and By-pass Damper Control. In certain applications, the modulating control is obtained with adsorbent dehumidifiers by placing face and by-pass dampers around the dehumidifier. Face and by-pass dampers are installed in the dehumidified air stream under the operation of the humidistat or wet-bulb thermostat controller. The controls and equipment are designed so that, under maximum load, all air passes through the dehumidifier. At less than maximum load, the humidistat operates the face and by-pass dampers to by-pass a certain amount of air around the dehumidifier in order to modulate the capacity of the dehumidifier to the load requirements.

Off and On Control. In certain applications, the adsorbent dehumidifier is cycled from a wet-bulb thermostat or humidistat in the controlled space. This type of control is chiefly used in comfort installations and other installations where some variation in the relative humidity is permissible.

APPLICATION OF ABSORBENT AND ADSORBENT DEHUMIDIFIERS

Absorbent and adsorbent dehumidifiers are employed where it is desirable to control the removal of moisture from the air independent of the dry-bulb temperature of the air. Dehumidifiers are used for this purpose in industrial processing air conditioning and in comfort air conditioning in combination with refrigeration.

Dehumidifiers are favorably applied to remove moisture from outside air in applications requiring delivery of air at low relative humidities. Figure 9-19 is an interesting application of dehumidification equipment with refrigeration for aftercooling the air. This arrangement is applied to a pharmaceutical plant where it is necessary to maintain various spaces at different relative humidities and dry-bulb temperatures.

The dehumidifier reduces the fresh air to an outlet condition of 97° DB and 40 grains per lb. This air is distributed to the various air-conditioned spaces through booster units. Small booster dehumidifiers are provided for those departments or spaces requiring greater dehumidification than that supplied by the main air supply. Cooling coils and heating coils are provided when control of the dry-bulb temperature of the air in the various spaces is required.

Dehumidifiers are applied to many industrial drying problems where low relative humidities are required for the processing, manufacturing, and storage of various materials. Figure 9-18 is a typical arrangement of equipment for control of a space or drying tunnel at relative low humidities. A cooling coil or heating coil is usually applied after the dehumidifier for dry-bulb-temperature regulation.

Dehumidification equipment is employed to advantage in air-conditioning problems requiring low relative-humidity control and no temperature control. The preservation of inactive naval vessels by holding the interior of the ship at a relative humidity below 30 per cent to prevent mold, corrosion, and other moisture damage is an example of low-humidity control with no temperature control.

The application of dehumidification equipment to comfort air conditioning in combination with refrigeration for aftercooling provides the following advantages without the use of reheat:

1. Independent control of relative humidity and temperature with varying latent- and sensible-heat loads. The humidistat or wet-bulb thermostat directly controls the moisture removal in the dehumidifier, and the sensible cooling is controlled by the thermostat regulating the amount of refrigerating effect.

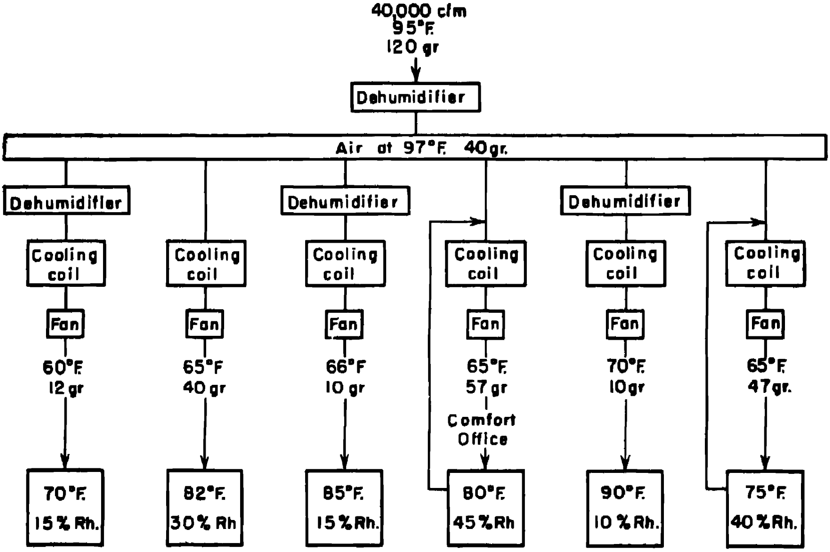

Fig. 9-19. Flow chart for central system dehumidifier with booster units.

2. Operation of the refrigeration equipment at higher evaporator temperatures for most applications owing to the use of refrigeration equipment for only sensible cooling.

3. The first cost and operating cost of a combination system compared with a straight conventional refrigeration system is dependent upon the design air conditions, the size of the installation, the ratio of latent heat to total heat load, the amount of fresh air, the relative cost of steam or gas and electric power, and the cost of cooling water. In general, it may be said that the operating cost tends to be lower for a combination system, and the first cost tends to be higher.

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SECTION 10

INDUSTRIAL INSTRUMENTATION AND CONTROL

BY DOUGLAS M. CONSIDINE

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INDUSTRIAL INSTRUMENTATION AND CONTROL

BY DOUGLAS M. CONSIDINE

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The importance of industrial instrumentation to the daily activities and responsibilities of the plant and process engineer has increased manyfold in the past few years. Continuous manufacturing and processing techniques, plus new and improved methods of measurement and control, are principally responsible for the rapid strides made in instrumentation. In hundreds of industrial organizations, instruments have created many new departments and jobs. Instrument Department and Instrumentation Engineer are becoming familiar phrases in plant terminology.

Because of the tremendously wide field embraced by industrial instrumentation, it is impossible thoroughly to cover or even mention all phases of the subject. The phases covered have been selected because of the significance which they bear to plant and process engineers. A list of excellent references is included at the end of the section. Broad coverage of other phases of instrumentation is contained in the sources listed.

Since plant and process engineers frequently must present valid and convincing reasons to management for proposed expenditures, including those for instrumentation, the basic answers to the question, "Why measure and control?" are quite important. A part of the section is devoted to this consideration. While many plant engineers have a very detailed knowledge of many instruments or controls, it has been found that an over-all understanding of the process variables, their nature and relation to one another, is often lacking. An especial effort has been made here to classify, compare, and discuss the nature of the variable. Measurement is an extremely important part of instrumentation since accurate control can be based only upon reliable measurements. Fundamental measuring means are discussed here in a broad, informal manner. The approach is designed to convey how certain very basic criteria are utilized for the measurement of many different variables. Detailed considerations of measurement must be obtained from references specifically covering this subject. Particular stress has been placed here upon the newer measuring means, such as those for determining chemical composition, dew points, etc., which are not yet thoroughly covered in the literature. Automatic control is a very broad and complex subject, and volumes have been devoted to it alone. Here only the fundamentals of the subject can be presented. To stress the practical application of measuring and controlling instruments, a section on automatic-control-system applications has been included. And finally, a subject of interest to most plant and process engineers, *viz.*, instrumentation costs, is briefly covered.

WIDE SCOPE OF INSTRUMENTATION

Few products of modern industry escape passage through some process environment which must be controlled within rather narrow limits. Temperature, pressure, humidity, flow, and other factors, commonly referred to as *process variables*, literally govern the environment to which a product is exposed and often will determine whether the product will be of high quality or a reject. Wherever stabilization of a process variable will increase product quality, decrease wastes or rejects, save steam or fuel, or enable an operator to do a better job in less time, instruments and automatic controls should be considered. Since there are several process variables of major importance and since these variables are encountered in hundreds of different industries, the scope of instrumentation is extremely wide.

WHY MEASURE AND CONTROL?

In the final essence, most advantages claimed for instrumentation can be resolved in terms of economics. Instruments and controls intelligently applied generally will enable the manufacture of a better product at a lower cost. Instruments are also applied widely in the interest of safety to personnel and equipment.

Product Quality. Most industrial products are manufactured to meet certain specified physical or chemical properties. As the quality of a product goes up, the tolerances governing deviation from these properties become narrower. And, as tolerances grow narrower, the need for instruments and automatic controls generally increases.

Thousands of examples of how final product quality is tied in with manufacturing conditions could be cited. Only a few can be mentioned here. In the examples given, the critical process variables are in italics.

The hardness, ductility, resistance to corrosion, and several other properties of a metallurgical product are representative of the care exercised in (1) proportioning, by *weight*, of the original ingredients and (2) maintaining the proper operating *time* and *temperature* in melting, refining, pouring, and heat-treating.

The purity of a chemical product may depend upon (1) the proper proportioning of original ingredients by *weight* or *flow* (volume), (2) holding the *pressure* of the reaction vessel constant for a specified *time*, and (3) adjusting the *pH* of the final product through addition of a corrective agent.

The weight, strength, and other properties of a paper product are partly the result of (1) digesting the pulp at the proper *temperature* or *pressure* and for a specified *time*, (2) regulating the *liquid level* of the head box, (3) adjusting the *pH* of the pulp, (4) maintaining the proper *speed* of the paper machine, (5) drying the paper at the desired *temperature* and to the proper *moisture content*, and (6) storing the paper at the most desirable *humidity*.

Product quality and uniformity can result only when processing conditions are held closely from hour to hour and from day to day. Automatic controls not only eliminate the drudgery of manually maintaining these conditions but eliminate the risks of human errors and carelessness. At the same time, the operator is freed to do other important tasks.

Direct Materials Savings. By consistently maintaining product quality, the number of rejects produced is cut to a minimum, thus effecting savings of raw materials as well as the productive time of machinery and men. These savings in time and material cut over-all manufacturing costs which means greater profits and a lower selling price to the consumer.

Instrumentation has done an outstanding job in cutting the nation's fuel bill and in so doing has contributed to conservation of our natural resources. Automatic controls through eliminating overheating of ovens, furnaces, driers, kettles, and other processing equipment have cut to a minimum the number of Btu going up the stack. Heating controls are generally based upon temperature measurement, and there is no human fireman or operator who can sense temperature with the accuracy and sensitivity of a thermocouple or thermometer bulb. Fuel savings through instrumentation, then, are not difficult to understand.

Conditioning of trade wastes is another example of where instruments have effected large savings in materials Often the wastes from a plant are dumped into a public stream. Frequently, these wastes contain objectionable materials, such as acids, which destroy fish and other wildlife. To avoid these harmful results, many operators add some corrective agent such as lime to the wastes to neutralize the objectionable effects. By measuring the pH of the treated wastes, considerable savings can be realized through elimination of overdosage of corrective agent.

Cost Accounting. Cost accountants are helpless without reliable data upon which to base their studies. Recording and integrating instruments comprise one of the accountant's principal tools in collecting data. Flowmeters are used widely to measure steam, water, fuel, and other utilities used by various departments in a plant. Where solid materials are involved, scales are used to measure the transfer and consumption of materials of interest to the accountant.

Product Testing. Most manufacturers perform certain physical and chemical tests on all products made, or at least upon representative samples taken from the production lines. Such testing could not be carried out without the aid of modern instruments. Viscosimeters, X-ray spectrometers, checkweighing scales, hydrometers, and refractometers—these are but a few of the instruments in daily use for product testing.

Plant Safety. Most plants have one or more areas where fire, explosion, poison, or other hazards are present and would be disastrous unless carefully watched and controlled. A gas-fired oven, a solvent-recovery area, a paint spray booth—these are examples. Fortunately, modern instruments are available to detect the presence of dangerous concentrations of gases and vapors, of flame failure in combustion units, and similar conditions. These instruments not only detect the existence of dangerous conditions but sound visual or audible alarms and in many cases immediately institute protective measures automatically, such as shutting off fuel supplies, closing fire doors, or starting auxiliary ventilating systems. Instruments have saved many lives.

Advantages—ad Infinitum. A complete delineation of the advantages of instrumentation cannot be given here. One other advantage which is often overlooked, however, is that which enables many complex processes to exist. This is the advantage of practically instantaneous coordination. With the hundreds of valves and dampers which must almost continuously be adjusted in many processes, it is difficult to comprehend how the process could operate manually, even with a man at every control valve. Possibly, one man with a hundred hands and the brain to coordinate those hands could control such a process. But certainly, a hundred men, each with two hands and a brain, but without the instantaneous coordination required, could not tackle the control of such a process.

PROCESS VARIABLES

The process variables are extremely difficult to classify. They can be grouped, however, with some logic. One method of grouping is followed here.

Energy Variables. Energy variables are so termed because they are related to the energy content of a process stream or piece of process equipment. While these variables are large factors in determining the final properties of a product, they are unimportant once this product has been made. Energy variables include (1) temperature and (2) pressure (positive and vacuum).

Temperature. The existence of three states of matter—solid, liquid, and gas—is convincing proof that temperature vitally affects every substance at at least two points on the temperature scale. Advantage of the changing state of substances at definite temperatures is taken in processing, wherein freezing and boiling points constitute the basis for separating materials. Two substances or different boiling points, for example, can be separated by distillation. Temperature also affects the rate of chemical reaction between substances, the rate of reaction increasing two- or threefold for each 10°C rise in temperature. The solubility of solids in liquids and of gases in liquids is also dependent upon temperature. In general, the solubility of solids in liquids increases with an increasing temperature, while the solubility of gases in liquids increases with a decreasing temperature. Thus, it should be obvious why temperature is so important a factor to the chemical operations, crystallization and absorption. The relationship between crystallization of solids and temperature explains the need for accurate temperature control in many heat-treating operations. The rate of bacterial and enzyme action is also critically affected by temperature. The range over which fermentation and other biochemical actions occur, therefore, is relatively narrow. Since there are few substances which are unaffected by temperature over a wide range, the measurement and control of temperature constitutes the largest field of application for industrial instruments.

Pressure. Under many conditions encountered in industrial processing, pressure and temperature are interlocked by nature so that one variable is fixed by the other. Often the plant or instrumentation engineer will be presented with making a decision as to which variable should be selected as the basis of control for a piece of process equipment. Pressure, like temperature, markedly affects the boiling points of substances and, to a lesser degree, freezing points. Pressure also affects the rate of chemical reaction, especially where the products of reaction are gases. Where the reactants and products of reaction are solids or liquids, however, pressure is relatively unimportant. Pressure control is extremely important to boiler operation where the safety of personnel and equipment depends upon maintaining pressure within prescribed limits. Recent advances in both extremely h-p and high-vacuum techniques have accentuated the importance of pressure measurement and control.

Quantity and Rate Variables. Under quantity and rate variables are variables that bear a direct relationship to the quantities and flow rates of materials in a process They include (1) fluid flow rate, (2) liquid level, (3) weight and weight flow rate of solids and liquids, (4) thickness and other dimensions of solids, and (5) speed of processing machinery. Unlike the energy variables, these variables do not affect the properties of the substances being processed so much as they determine the efficiency and operation of the process itself. For example, the physical properties of water flowing at the rate of 20 gpm are the same as those of water flowing at 2,000 gpm. On the other hand, an item of processing equipment designed to handle 20 gpm will be quite incapable of handling 2,000 gpm.

Flow. Although flow rates of single components in a process do not affect the properties of the components, the rates of flow of two or more components to form a mixture, or of two or more constituents to form a chemical compound, do greatly affect the results obtained. For example, in proportioning a heavy oil to a light oil, the flow rates of both components will determine the viscosity of the mixture. Since substances react chemically in definite proportions one to the other, accurate flow proportioning is required to avoid the presence of excess reactant in the final product. The importance of flow measurement to continuous processing, where equipment is designed to operate at maximum efficiency at certain production rates, cannot be overemphasized. Similarly, the importance of flow measurement to the gathering of statistics for plant cost accounting cannot be overlooked. It is interesting to note that flow often is a dependent variable fixed by a key variable, such as temperature or pressure. The flow control of steam or of cooling water to a heat exchanger to maintain a desired temperature is an example of this latter situation.

Liquid Level. Like flow, liquid level in itself does not affect the properties of a substance being processed. Nevertheless, it must be measured and controlled so that processing equipment will not run dry or overflow, with possible disastrous results.

Weight. While of principal concern to the proper proportioning of solid or liquid ingredients to a process, weight is also a specific property of a substance which may be desired to be held within definite limits. In extruding tire tread or tubes, for example, it is desired to obtain a definite weight per running foot of the extruded material, since this weight is vital to the balance, weight, and physical dimensions of the finished product. Measurement of weight in various stages of manufacturing is also frequently done as an aid to plant cost accounting. Counting large numbers of parts by weight, for example, is widely practiced.

Thickness. Thickness and other physical dimensions of substances being processed often have a definite bearing upon the final product. For example, sheet steel is rolled to a definite gauge; pipe is extruded to a definite bore; and paper is calendered to a definite caliper.

Speed. Speed is quite similar to flow rate in that it determines to a large degree the rate of production or processing. In itself, speed does not affect the properties of a substance. For example, paper running at 300 or 1,500 fpm on a machine is not affected by the speed itself. It is affected by the time with which it is in contact with the drying cylinders, however, and that time will be affected by speed. Since processing equipment, such as a paper machine, is designed to operate at a certain speed, control of speed is important. Too fast a speed on a paper machine, for example, will result in wet, improperly dried paper, while too slow a speed will result in overdried paper.

Substance Property Variables. Substance property variables are definite, specific properties of substances and include (1) density and specific gravity; (2) viscosity and consistency; (3) pH or hydrogen-ion concentration; (4) electrical conductivity;

(5) thermal conductivity; (6) calorific value; (7) explosibility and flammability; (8) humidity; (9) color, opacity, smoke density, and turbidity; (10) moisture content; and (11) actual chemical composition.

Several of the above variables are of principal importance in determining, partly or wholly, the chemical composition of a substance. These variables include electrical conductivity, thermal conductivity, and specific gravity. Electrical and thermal conductivity, for example, are relatively unimportant physical characteristics of most substances (insulating materials would be an exception). On the other hand, electrical conductivity is used widely as an index of the moisture content of a substance, while thermal conductivity is used to measure the composition of gases.

To illustrate how difficult a classification of the process variables really is, it is interesting to note that weight and thickness, classified previously as quantity or rate variables, may just as logically be classified here as substance property variables.

Density and Specific Gravity. Density and specific gravity are particularly important to evaporating and crystallizing operations where a liquid solution must be evaporated to a certain concentration before crystals will begin to form. Where reasonably pure substances are involved, a measurement of the specific gravity proves to be a useful index to concentration.

Viscosity and Consistency. Lubricating oils are the most fitting examples of substances which are produced having viscosity as one of the final specified properties. Viscosity is also extremely important to the production of paints and varnishes. Consistency, closely related to viscosity, is of especial importance to paper production. This variable property of paper pulp greatly affects formation of the sheet on the paper-machine wire.

pH or Hydrogen-ion Concentration. As a measure of the effective acidity or alkalinity of a process liquor, pH is of significant importance to many of the chemical unit operations, including precipitation, coagulation, neutralization, and fermentation. Usually the pH of the final product is not nearly so important as the pH of the various liquors during production. For example, certain organic solvents and compounds are produced by the action of living enzymes or molds. These living organisms usually yield a maximum quantity of the desired substance at a definite pH value, and, therefore, close control over pH is important. On the other hand, the actual pH of the finished product is not usually of particular significance.

Electrical Conductivity. Electrical conductivity finds limited value as an index to chemical composition of liquids and also as a measure of the moisture content of such substances as paper and textiles. With the exception of wire and electrical insulators, electrical conductivity is not a particularly significant property of a final product.

Thermal Conductivity. Measurement of thermal conductivity finds its widest application as an index to the chemical composition of gases. With the exception of heat-insulating materials, it is not an important final property of a product.

Calorific Value. Calorific value is an extremely important property of fuels, of course, and in this connection is measured widely. This property is also used to a limited extent in determining the chemical composition of gases.

Explosibility and Flammability. Explosibility and flammability apply mainly to gases and vapors and are measured, of course, in the interest of safety to personnel and equipment.

Humidity. Generally, humidity is considered a term to signify some function of the concentration of water vapor in air. It is also being applied today to denote the concentration of other vapors in air or in other gases. This is another example of a variable which is not important as a final property of a substance but which is extremely important as a processing condition. It is highly important in comfort and process air conditioning and to such operations as drying, absorption, adsorption, and storing.

Color. Color is of principal importance as a final product quality, although it is used to a limited extent in determining chemical composition. Chlorine and nitrogen dioxide, for example, are gases, the colors of which vary with concentration. Hydrogen sulfide reacts with lead acetate to produce black lead sulfide. By subjecting a lead-acetate-treated paper to hydrogen sulfide gas and measuring the intensity of the black color photoelectrically, an indirect measure of the gas concentration is obtained.

Moisture Content. Moisture content is an important property to materials in production as well as to final products. The moisture content of warp yarn as it emerges from the slasher is very important to the success of the subsequent weaving or knitting process. The moisture content of the paper sheet as it emerges from the dry end of the paper machine is important to subsequent converting and printing operations.

Chemical Composition. Product purity is of especial significance to the chemical, petroleum, and pharmaceutical industries. In many cases, temperature, pressure, and other process variables are measured today as means to control product purity. Advances made in the direct measurement and control of chemical composition over the past few years undoubtedly will have a marked effect upon industrial-instrumentation techniques. Why use the boiling point of a liquid as an index to purity if an equally practical means for directly analyzing the composition of the liquid is available?

BASIC CRITERIA OF MEASUREMENTS

The means for measuring the numerous process variables are about as difficult to classify or group as the variables themselves. Since the process variables are measured and controlled because they change, it follows that some phenomena must be measured which reliably represent these changes. For many measuring instruments, these phenomena fall into one of the following categories: (1) change of pressure and (2) change of electrical voltage, resistance, or current.

The discussion of measuring instruments to follow does not correspond to the conventional manner of presentation followed in most handbooks and texts. Usually each process variable is taken up individually, first exhausting temperature measurement, pressure measurement, and the like. In this discussion, an attempt is made to interlock the major measuring instruments on the basis of the phenomena which they utilize as their bases of measurement. If it is desired first to study all temperature-measuring instruments, then all pressure-measuring instruments, etc., reference to texts listed in the bibliography is suggested.

PRESSURE-ACTUATED INSTRUMENTS

Pressure-actuated instruments are of two general types, *viz.*, (1) those designed to measure pressure as a process variable and (2) those designed to measure pressure as an index of some other process variable, such as temperature, flow, or liquid level. In either case, the measuring means used are strikingly similar.

Elements for Measuring Pressure Changes. Whether a pressure change is due to a change in flow, a change in liquid level, a change in temperature, or a change in process pressure itself, there are seven basic elements which will measure these pressure changes. These are (1) the helix, (2) the spiral, (3) the bellows, (4) the compensated bellows, (5) the bourdon tube, (6) the mechanical manometer, and (7) the electrical manometer (see Fig. 10-1).

The helix, spiral, bellows, and bourdon tube have much in common in that they comprise metal enclosures which expand when exposed to internal pressure. Expansion of the helix, spiral, and bourdon tube causes a twisting action. By attaching a pen or pointer to these elements by means of a link, an indication or record of movement of the element and, consequently, of pressure is obtained. Expansion of a bellows causes a linear movement, but attachment of pen or pointer via links is similar.

In industrial instruments, the manometer generally takes the form of a U tube filled with water, mercury, or other suitable liquid. Pressure is applied to both sides of the manometer and causes a float in one side to rise or fall. This float may be directly connected to a pen or pointer by means of a link, or the float may raise or lower an armature which is part of a solenoid. This latter arrangement permits electrical transmission of the measurement over almost unlimited distances.

Elements for Creating Pressure Changes (see Fig. 10-2). Obviously, where pressure itself is the process variable being measured, the changes in pressure are created

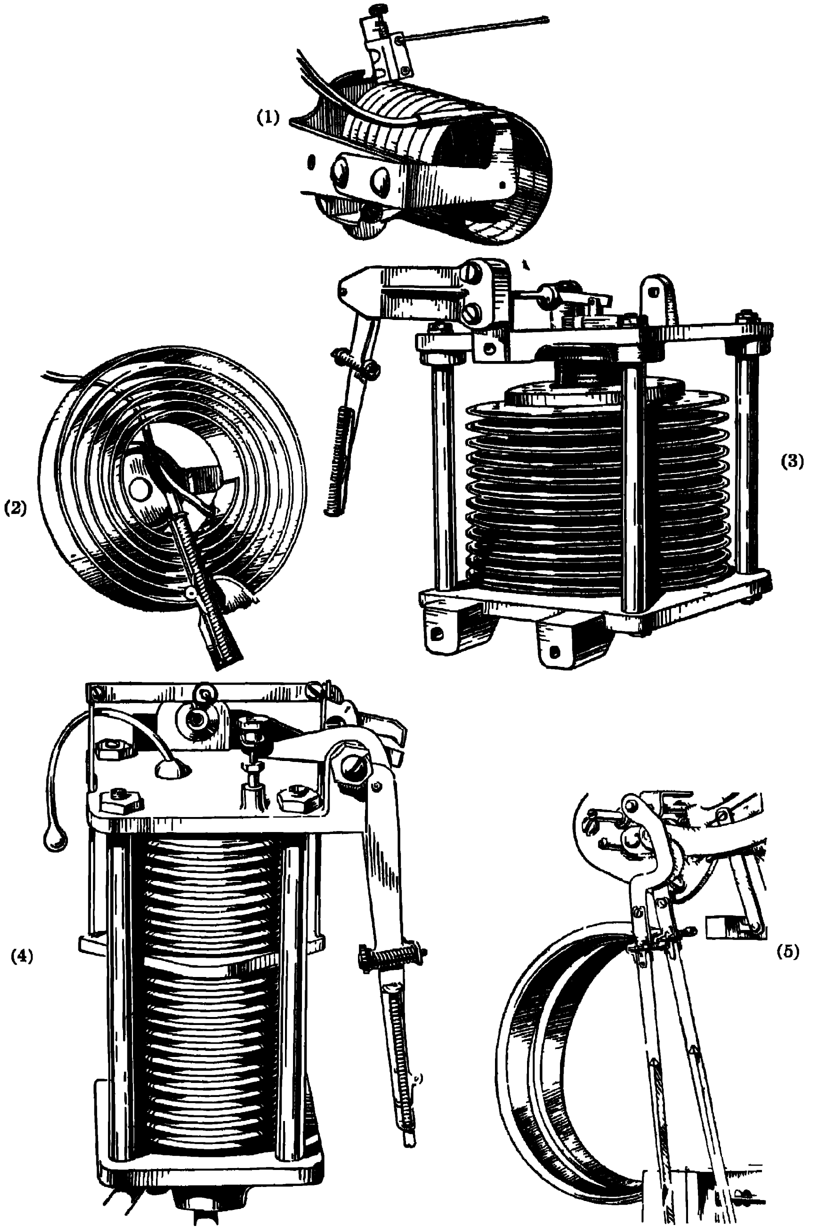
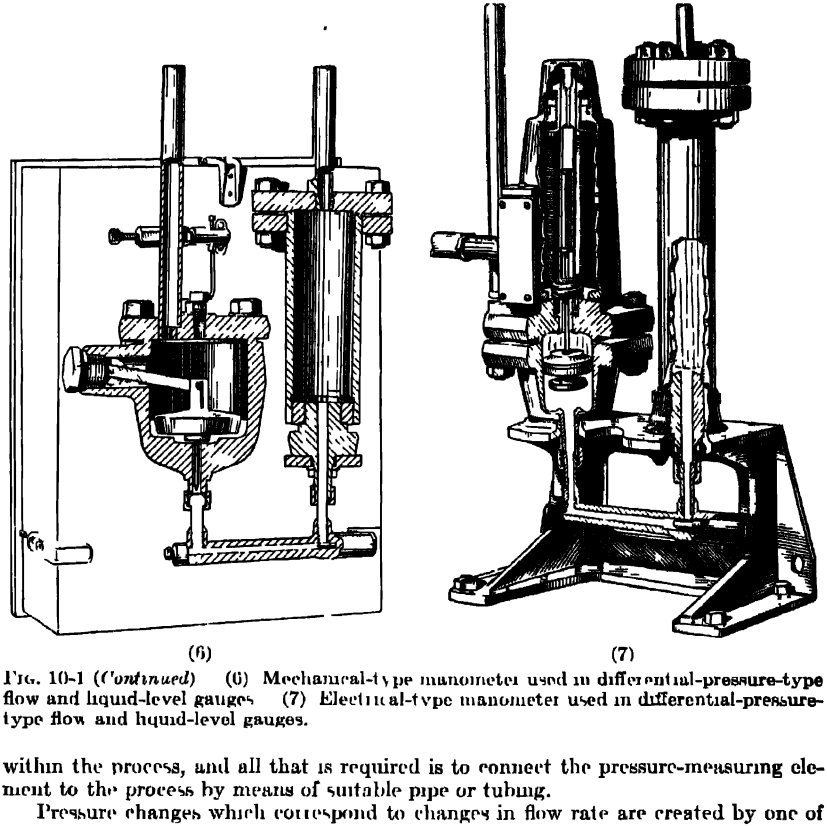


FIG 10-1 (1) Helix used in pressure gauges and pressure-type thermometers (2) Spiral used in pressure gauges and pressure-type thermometers (3) Bellows used in pressure and vacuum gauges and in bubble- or diaphragm-type liquid-level gauges. (4) Bellows with barometric compensation used in absolute pressure gauges. (5) Bourdon tube used in pressure gauges.



(6) (7)

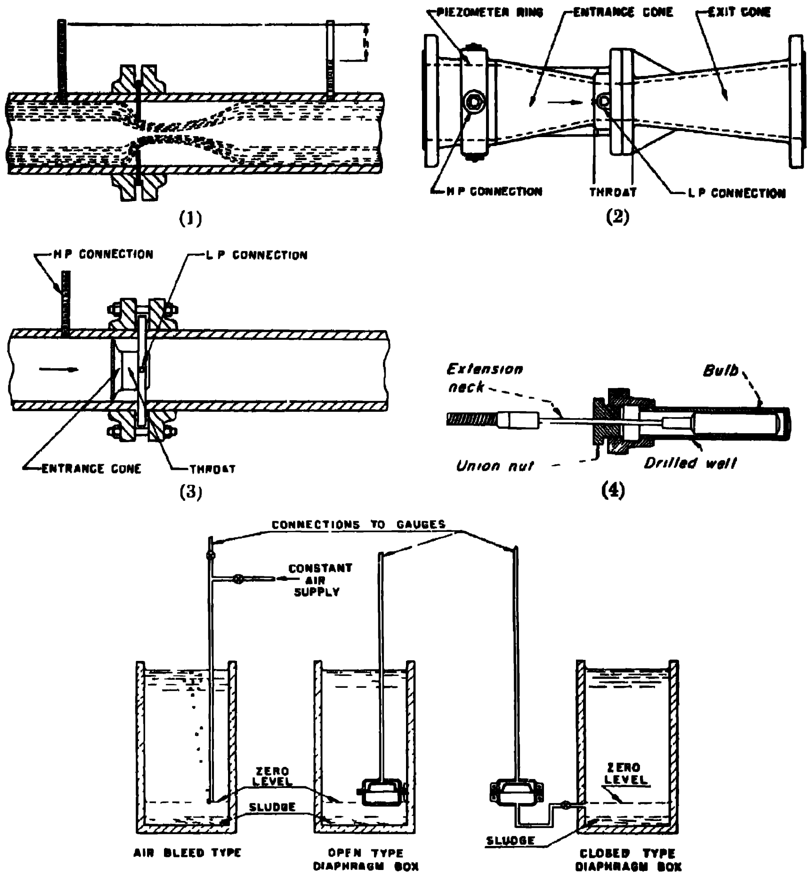
Fig. 10-1 (*Continued*) (6) Mechanical-type manometer used in differential-pressure-type flow and liquid-level gauges (7) Electrical-type manometer used in differential-pressure-type flow and liquid-level gauges.

within the process, and all that is required is to connect the pressure-measuring element to the process by means of suitable pipe or tubing.

Pressure changes which correspond to changes in flow rate are created by one of several devices, including (1) orifice, (2) venturi, and (3) flow nozzle. If a restriction is placed in a pipe line containing a flowing fluid, a pressure differential will be created by the restriction. This differential in static pressure bears a definite relationship to the rate of flow past the restriction and, therefore, can be used as a measure of fluid flow. Two pressure taps, one on either side of the restriction, are connected to a manometer.

Pressure changes which correspond to changes in temperature are created by thermometer bulbs (see Fig. 10-2(4)) of three principal types, *viz.*, (1) mercury-filled, (2) gas-filled, and (3) liquid-plus-vapor filled. When any closed vessel is subjected to a rise in temperature, the internal pressure of the vessel will be increased. The opposite action occurs when a fall in temperature is involved. The mercury-filled system comprises a bulb, connecting capillary tubing, and helix or spiral, all filled with mercury. As the bulb, which is located in the medium to be measured, is heated, the mercury expands causing the measuring element to expand. In the gas-filled system, the mercury is replaced with a suitable gas—the principle is exactly the same. In the liquid-plus-vapor system, the bulb is filled with a liquid and its vapor. As the bulb is heated, more of the liquid vaporizes, increasing the internal pressure of the measuring system. Systems of the latter type have the advantage of not being affected by ambient-temperature changes along the capillary connecting tubing.

Pressure changes which correspond to changes in liquid level employ a manometer or bubble pipe. In the manometer system, two taps are made in the side of the vessel containing the liquid. As the liquid level rises or falls with relation to these



(1) (2) (3) (4) (5)

FIG. 10-2. (1) Orifice plate used in differential-pressure-type flow meters. (2) Venturi tube used in differential-pressure-type flow meters. (3) Flow nozzle used in differential-pressure-type flow meters. (4) Typical measuring bulb of pressure-type thermometer. (5) Bubble- and diaphragm-type liquid-level gauges.

taps, the pressure differential changes. By connecting these taps to a manometer, just as with an orifice-type flowmeter, the position of the manometer float will indicate the level in the vessel. In the bubble-pipe system (see Fig. 10-2(5), an open-ended pipe is placed in the vessel, and air under pressure is forced into the pipe, causing bubbles to emerge from the end of the pipe. By regulating the supply air pressure so that a definite number of bubbles rise from the pipe over a given time and by measuring the back pressure in the system required to maintain this condition, a measure of the head of liquid in a vessel is obtained. The back pressure is measured by means of a spiral or bellows.

ELECTRICALLY ACTUATED INSTRUMENTS

A change in electrical resistance, electromotive force, or current is used to measure any number of the process variables. Some of the more important variables measured in this manner include *temperature* (thermocouple or thermopile), *pressure* (thermocouple, electrical resistance, or ionization), *moisture* (electrical conductivity), *pH* (glass electrode), *liquid level* (electrodes), and *chemical composition* (Geiger counter, dropping mercury electrode, etc.).

Instruments for Measuring Electrical Changes. There are two basic types of instruments for measuring electrical changes, *viz.*, (1) deflectional-type millivoltmeter and (2) null-balance potentiometer (galvanometer and electronic types). Where precision is required, the potentiometer-type instrument generally is used.

Millivoltmeters (see Fig. 10-3) employ the D'Arsonval galvanometer. Deflections are obtained by the reaction between an energized coil and the field of a permanent magnet. The coil of fine wire is connected across the circuit to be measured through spiral springs. It is supported by pivots and free to move in the annular space between a soft-iron core and the pole pieces of a permanent magnet. Owing to the shape of the pole pieces and core, the strength of the magnetic field is uniform. Hence, the deflections of the pointer are directly proportional to the current, and the scale divisions are uniform.

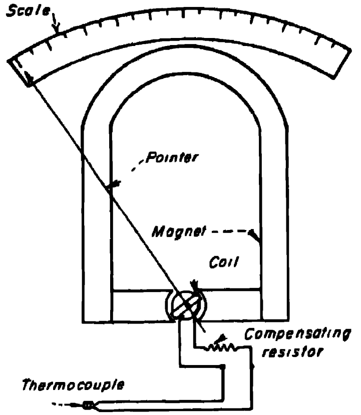


Fig. 10-3. Deflectional-type millivoltmeter.

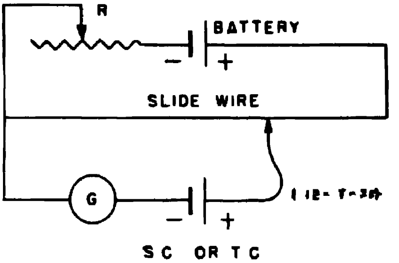


Fig. 10-4. Basic circuit of null-balance-type potentiometer.

Although it is possible to measure the temperature of a thermocouple directly by means of a millivoltmeter, this method introduces as a possible variable the resistance of the leads and various connections. Consequently, a potentiometer is generally used as the measuring circuit in precision instruments, since in this method resistance is ruled out as a variable owing to the fact that the electrical potential applied is balanced against an equal and opposite potential produced by a battery. An elementary potentiometer circuit is shown in Fig. 10-4. In the upper circuit, the battery produces a potential drop in the slide-wire which may be adjusted by resistance *R*. In the lower circuit, an opposite potential is produced by a standard cell of definitely known voltage, and the drop in the slide-wire can be made equal and opposite to that produced by the battery by an adjustment of the slider, this point being indicated by a zero reading of the galvanometer. A thermocouple can then be substituted for the standard cell and the galvanometer again brought to zero by adjusting the slider. The relation between the first and second slider readings is a measure of the difference between the thermocouple potential and that of the standard cell, and the slide-wire positions may be calibrated in terms of the temperatures of the hot junction corresponding to the various potentials. The potentiometer just described is manually balanced. In industrial instruments, the balancing action is accomplished automatically, and the thermocouple voltage (in terms of temperature) is indicated continuously by mechanical, electrical, or electronic means.

The detecting and rebalancing means used in one type of electronic potentiometer are illustrated in Fig. 10-5. Briefly, the d-c millivoltage created by the thermocouple is changed to an a-c voltage of proportional magnitude in a converter. The converter is essentially a flat metal reed oscillating between two contacts connected to the opposite ends of the primary winding of an input transformer. The unbalanced d-c voltage is impressed across the converter and the center tap of the primary winding on the input transformer. As the reed moves from one contact to the other, any unbalanced d-c voltage will cause direct current to flow first in one direction through one-half the primary winding, then in the opposite direction through the other half. This action generates an alternating flux in the input transformer core which, in turn, induces an alternating voltage on the transformer secondary.

The action of the converter is related to the a-c supply voltage by the energizing coil which is excited by the a-c supply voltage through a stepdown transformer. The reed is polarized by a permanent magnet and, therefore, is actuated by the energizing coil to oscillate in synchronism with the a-c supply voltage. The reed closes one contact to the transformer for one-half the supply-voltage cycle and the other contact for the other half, making one complete oscillation for each cycle. The direct current flowing in each half of the transformer primary winding, therefore, will create an alternating voltage in the transformer secondary of the same frequency as the supply voltage. This alternating voltage is amplified in voltage and power to where it will actuate a two-phase balancing motor.

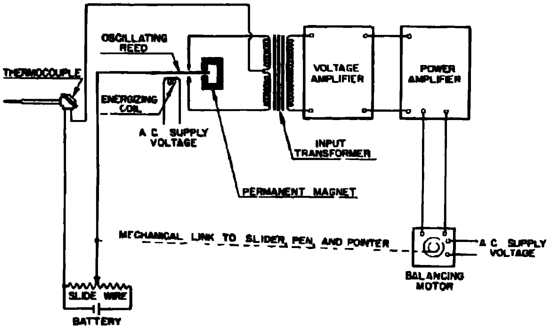


FIG. 10-5. Schematic diagram of electronic-type potentiometer.

If there is an unbalance between the slide-wire voltage and the thermocouple voltage, the balancing motor functions to rebalance the two voltages by moving the slider on the slide-wire. The direction in which the motor turns is determined by a definite phase relationship between the a-c supply voltage and the amplified a-c thermocouple voltage. This relationship depends, of course, upon whether the measured variable is increasing or decreasing in magnitude. The use of electronic and electromechanical components in this potentiometer has virtually eliminated the multiplicity of moving mechanical parts common to previous mechanically balanced potentiometers.

Elements for Creating Electrical Changes. In the foregoing description of the millivoltmeter and potentiometer, the thermocouple was used as the detecting means. This was done merely for convenience. Instruments of the type just described can be used to measure electrical changes produced by other devices such as pH electrodes, resistance thermometers, tachometer generators, and others.

Temperature-measuring Devices—Thermocouple. The thermocouple (see Fig. 10-6) consists of two dissimilar metal wires joined at one end, generally known as the hot junction. Often the wires are contained in tubes for protection against excessively oxidizing or reducing furnace atmospheres and from other types of corrosion. When a thermocouple is heated at one end (hot junction), while the free ends (terminals) are kept at a known temperature, a potential is set up between the terminals which is related to the hot-junction temperature. This potential can be measured by a millivoltmeter or potentiometer, the latter instruments being calibrated in terms of temperature.

As will be pointed out later, thermocouples not only are used in the direct measurement of temperature but are also used in instruments for measuring high vacuums, in spectrometers for measuring chemical composition, etc. The most commonly used combinations of wire include platinum and platinum–10 per cent rhodium, iron-constantan, chromel-alumel (Hoskins), and copper-constantan. Temperature-millivolt equivalents for these couples can be obtained from any instrument manufacturer specializing in pyrometry.

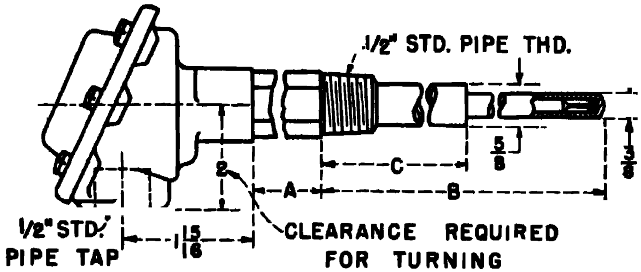


Fig 10-6 Typical thermocouple and protecting tube. Hot junction is shown inside tip at right, head with terminals for connecting extension lead wire is at left.

Thermopile. To increase the sensitivity and the emf output of a thermocouple, several thermocouples can be connected in series to form a thermopile, as illustrated in Fig 10-7. These devices are used widely in radiation pyrometers, which focus energy from a furnace or other hot object onto the thermopile by a lens system. Thus, the temperature-detecting device does not have to contact directly the object of measurement but can be located several feet away.

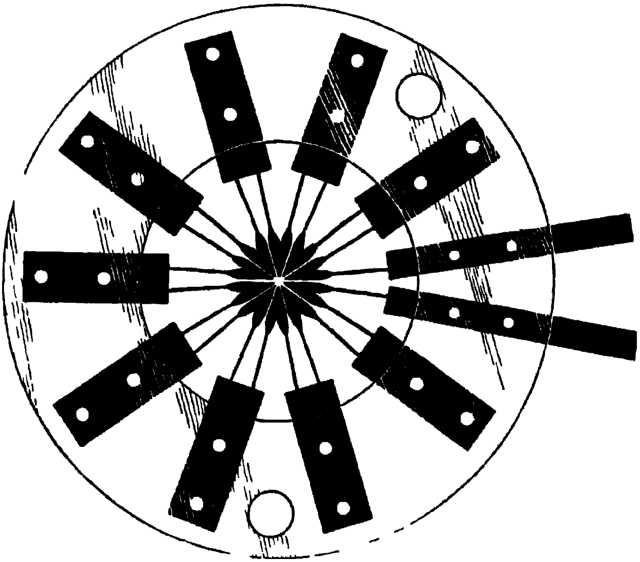


Fig 10-7 Enlarged view of thermopile (group of thermocouples connected in series) used in radiation-type pyrometer.

Resistance Thermometers. The electrical resistance of metal wire, usually a coil of nickel or platinum, is the temperature-sensing device employed in resistance thermometers. Since there is a definite relation between temperature and the resistance of the wire, the electrical resistivity of the wire can be measured and indicated or recorded in terms of temperature.

Electrometric pH Measurement. In electronic pH measurement, two electrodes are immersed in the unknown solution. One electrode, the *measuring* electrode, is sensitive to the presence of hydrogen ions; the other electrode, the *reference* electrode, is required to complete an electric circuit through the solution and to supply a constant potential. The reference electrode is not affected by pH. Since to date it is impossible to measure the potential of a single electrode, the reference electrode must be used. Each electrode is sometimes referred to as a *half cell.* The potential measured is the algebraic sum of the two electrodes and is conveniently measurable with a potentiometer.

Measuring Electrodes. Four outstanding electrodes have been developed which are sensitive to pH, *viz.*, (1) the hydrogen electrode, (2) the quinhydrone electrode, (3) the antimony electrode, and (4) the glass electrode. The glass electrode is the modern and most practical means of pH measurement and has practically replaced the other electrodes for industrial application.

Glass Electrode. When a thin membrane of special glass separates two solutions of different hydrogen-ion concentrations, a potential is created which is a function of the two concentrations. If the pH of one solution is maintained constant, that of the other solution can be determined from the potential produced. A typical glass electrode consists of a glass tube about one-half inch in diameter with a bulbous glass tip. The solution of constant pH is contained inside the tip. Several theories regarding the operation of glass electrodes have been advanced, but their explanation must be left to complete texts on the subject.

Reference Electrodes. A good reference electrode must possess a reasonably constant potential regardless of temperature changes, must maintain this potential over long periods of time, must not easily be poisoned by solutions, must be simple and easy to maintain, and must not in itself be affected by change in pH. Years of experience have proved the calomel electrode, saturated or unsaturated, to be the best answer to the above requirements. The saturated calomel electrode, most widely used, comprises mercury and mercurous chloride (calomel) with saturated potassium chloride as an electrolyte. In a typical electrode, a liquid junction between the electrode and the process solution is established by means of a small hole covered by a ground-glass sleeve. Diffusion of electrolyte from the electrode to the solution is slow but sufficient to provide excellent electrical conductivity.

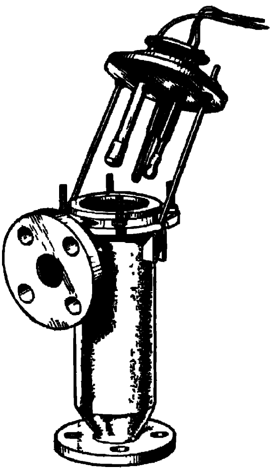


FIG. 10-8. pH electrode assembly designed for installation in pipe lines. (*National Technical Laboratories.*)

Electrode Assemblies. Sturdy construction of electrode assemblies has been responsible for a wider industrial use of pH measurement during the past few years. The flow assembly illustrated in Fig. 10-8 can be installed directly in process pipe lines and is capable of operating under moderate pressures. Numerous materials of construction, including porcelain-covered iron, rubber, stainless steel, nickel alloys, and glass, are available. An assembly available for immersion in tanks, vats, and other process vessels also is available.

pH Recording and Controlling Equipment. Null-balance potentiometers rather than deflectional instruments are required for pH measurement since the measuring instrument must not draw any appreciable current from the electrodes. Any current flow would cause polarization at the electrodes, immediately upsetting the accuracy of measurement. Potentiometers are available in four basic ranges: 0 to 7, 3 to 10, 6 to 13, and 2 to 12 pH. Pneumatic and electrical control components available with thermocouple potentiometers are also available with pH potentiometers.

Oxidation-reduction Potential. The measurement of oxidation-reduction potential is a reasonably well-known technique of physical chemistry. However, its application to industrial processes is just emerging from the experimental stage. Basically, oxidation-reduction potential is to oxidation and reduction chemistry what pH is to double-decomposition chemistry or neutralization of acids and bases. Kolthoff and Furman state that "It is the potential an unattackable metal electrode assumes in a solution relative to that of the normal hydrogen electrode." The measurement of oxidation-reduction potential in practice involves the use of a standard reference electrode, such as a calomel electrode, and a platinum, gold, or other nonattackable metallic electrode. The potential from the electrodes is measured by an instrument quite comparable to a potentiometer designed for pH measurement, but with the scale graduated to read in millivolts rather than in pH units. The measured potential may be expressed as millivolts or as E_h or R_h.

Because oxidation-reduction-potential measurement has been so infrequently used, it is a fertile field for investigation. In addition to concentration control of reducing and oxidizing solutions, it has been used to measure the chlorine content of water to less than one part per million, using a calomel and a silver–silver chloride electrode. Chemical engineers will immediately visualize many other applications which can be corroborated only by test data. Much can be expected from oxidation-reduction-potential measurements within the next few years.

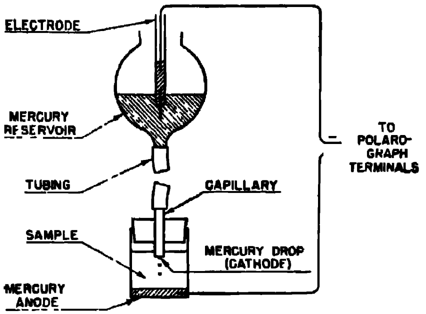


Fig. 10-9. Dropping mercury electrode used in polarograph. (*E. H. Sargent & Co.*)

Dropping Mercury Electrode. The polarograph (employing the dropping mercury electrode) is an analytical instrument, both qualitative and quantitative, for determining the chemical constituents of either aqueous or nonaqueous solutions. Although extremely small quantities of material can be detected, it is not limited to microanalysis. Both in micro- and macroanalytical work, it replaces tedious and slow wet-chemical methods for determining metallic and organic constituents.

The dropping mercury electrode, illustrated schematically in Fig. 10-9, comprises a very fine bore capillary tube connected by a flexible tube to a mercury reservoir and placed in the solution so that very fine drops of mercury are formed beneath the surface. The rate of mercury flow can be adjusted by varying the height of the reservoir and normally amounts to a drop every one to three seconds. The mercury reservoir is connected to one terminal of the polarograph; the other cell terminal is a pool of mercury in the bottom of the solution vessel.

In operation, a few drops of mercury added to the sample act as one electrode of the cell. The dropping mercury electrode next is inserted. An inert gas, usually nitrogen, then is bubbled through the solution to remove atmospheric oxygen which is reducible and gives a polarographic wave not usually of interest to the analyst. As the voltage impressed on the cell is raised slowly, the current remains nearly constant until a voltage is reached at which a component of the solution is reducible. The current then rises sharply to a new level at which it remains constant until the voltage is reached at which a second component is reducible. Another sharp rise in current to a new constant value then occurs. These sharp rises in current are referred to as polarographic steps or waves. Since the polarographic steps for various components occur at different voltages, several substances can be determined in one solution on a single polarogram.

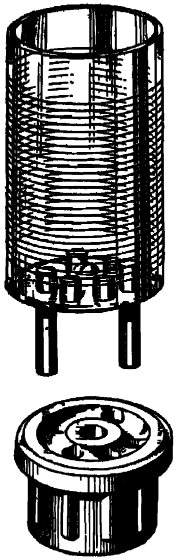


FIG. 10-10. Electrical humidity-sensing element. (*American Instrument Co.*)

The output from the polarographic cell is fed to the recorder so that the displacement of the pen from zero is determined by the amount of current flowing through the cell. The chart is driven by a constant-speed synchronous motor. The polarographic bridge which supplies voltage to the cell is also driven by a synchronous motor so that the chart displacement is a linear function of the voltage applied to the cell, *e.g.* the time ordinate of the chart becomes, in essence, the voltage ordinate. The record obtained, therefore, is current vs. voltage.

Electric Humidity Measurement. A comparatively new development in hygroscopic measurement is illustrated in Fig. 10-10. Here a double coil of palladium wire is wound on an insulating core, the wires being coated with a film of hygroscopic lithium chloride suspended in a water-soluble coating. The electrical conductivity of the coating varies with the relative humidity and is measured by a microammeter provided with a-c energy through a transformer and a full-wave copper-oxide rectifier. To secure a long range, several such sensitive elements must be used in series, each different in over-all resistance from the others. One instrument of this type is available with electronic recorders and electric or pneumatic control.

Electronic Dew-point Recorder. Referring to Fig. 10-11, which is a schematic diagram of one type of automatic dew-point recorder, note that the gas under test enters the gas chamber where it comes in contact with a thin metallic mirror to which a thermocouple is attached. The mirror is refrigerated to −90°F. Consequently, moisture from the gas under test forms on the surface, thus fogging the mirror. Coincident with the formation of fog or dew, the mirror is heated to the temperature at which the dew vanishes This temperature is the dew point of the gas under test. The electronically controlled heater maintains the mirror at this temperature until the dew point of the gas under test changes, at which time the mirror

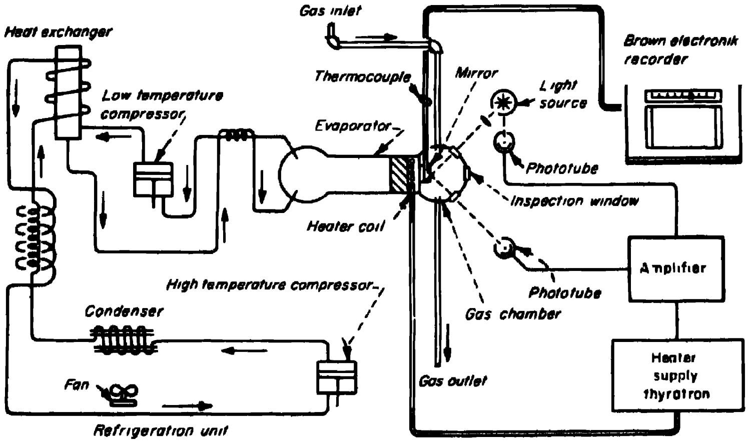


FIG. 10-11. Schematic diagram of dew-point-measuring and -recording system. (*General Electric Co.*)

temperature is changed to coincide with the formation of dew and is held at this new temperature. By recording the output voltage from the thermocouple attached to the mirror by means of a potentiometer, a continuous record of the dew point of the gas is obtained.

In operation, the mirror is constantly refrigerated by a gas flow against the back of the mirror. The refrigerating gas flow is cooled to −90°F by the action of a two-stage compressor and a dual refrigerating system. A heater mounted directly behind the mirror is capable of raising the temperature of the mirror from −90°F to ambient against the opposing cooling effect caused by the refrigerating system. An automatic optical system views the mirror surface and operates by means of a beam of light which passes through one of the windows of the gas chamber and is reflected from the mirror through the other window to the phototube. As the test gas passes through the gas chamber and dew condenses on the mirror, the reflectivity of the mirror lowers, thus decreasing the amount of light reaching the phototube and initiating a change in the magnitude of the heater current.

To reduce the effect of voltage variations and aging on the phototube characteristics, the lamp also shines on a second phototube which is balanced in the electrical circuit against the first tube. The amplified difference between the two phototube currents controls the heater current.

Measuring Moisture Content by Electrical Conductivity. Moisture content of solids can be measured in various ways in addition to the fundamental method of weighing and drying. Probably the commonest method is to measure the electrical conductivity. In the case of crystals or powdered materials it is necessary to ensure a standard density of material before making the measurement. Measuring the humidity of air in contact with the substance is another useful method.

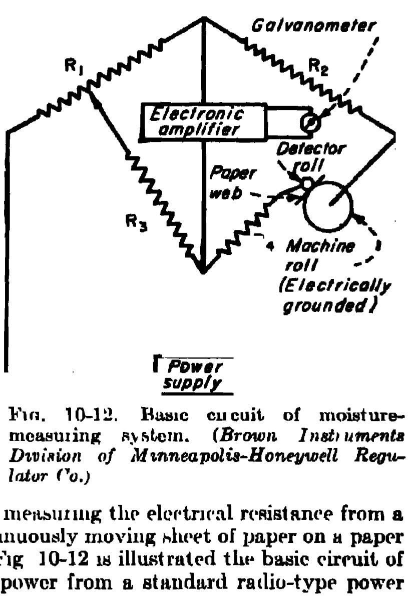


FIG. 10-12. Basic circuit of moisture-measuring system. (*Brown Instruments Division of Minneapolis-Honeywell Regulator Co.*)

In one method, moisture is detected by measuring the electrical resistance from a detector roll in direct contact with the continuously moving sheet of paper on a paper machine or warp on a textile slasher. In Fig 10-12 is illustrated the basic circuit of this device. The bridge is supplied with power from a standard radio-type power supply consisting of a transformer, rectifier tube, and filter.

The bridge consists of resistors R_1, R_2, R_3, and R_4 and the resistance of the object of measurement, which is in series with R_4. R_1 is the slide-wire in the instrument. This resistor is automatically adjusted by the mechanical linkage of the recorder, which, in turn, is positioned by the galvanometer, thus keeping the bridge in balance for changing values in resistance of the unknown. Resistor R_4 is used principally to control the scale shape and because of its extremely high resistance excludes any possible danger to personnel coming in contact with the detector roll.

Inasmuch as the resistances being measured are extremely high, the current in the bridge is extremely low with the safe voltages used. Therefore, it would require a most sensitive galvanometer of the laboratory type to detect small unbalances. It is necessary, therefore, electronically to amplify the unbalances occurring before feeding them into a standard commercial galvanometer. The amplifier consists simply of a twin-triode tube used in a balanced circuit in which the bridge current is fed into the grids, and the galvanometer is placed across the plates. When the grid voltages are at the same potential, the plate currents are equal, and the galvanometer does not deflect. As soon as any unbalance occurs in the bridge, the grid potentials change which, in turn, changes the plate currents, causing the galvanometer to deflect.

X-ray Diffraction Spectrometer. The use of X-ray diffraction for analytical purposes has become an extremely important aid to several of the process industries. The time and technique required to conduct an analysis by this method through photographic means have formerly limited its usefulness. However, development of the Geiger-counter X-ray spectrometer, together with the high-speed recording potentiometer, has greatly simplified the procedure and reduced the time required to make an analysis.

Basically, the diffraction of X rays in a solid serves to analyze its structure. The arrangement of atoms within the solid and their interrelationship provide the analytical information necessary to evaluate such properties as simple chemical identity, particle size and distribution, grain size, and similar properties. In such processes as the following, analysis by X-ray diffraction serves (1) to control addition of dyes, fillers, and accelerators to the rubber batch in the rubber industry; (2) to select the required manganese dioxide ores in battery manufacture; (3) to select clay of proper particle size in the manufacture of high-frequency insulators; and (4) to control the activator in fluorescent materials used in lamps and cathode-ray-tube screens. These applications are representative of a host of others. At present, a tool of the product-development and process-control laboratory, future research may adapt it to continuous process control.

Principle of X-ray Diffraction. When a beam of X rays is directed into a sample undergoing analysis, a series of secondary beams is reflected from the specimen in the form of a diffraction cone. Analysis of this cone with properly sensitized photographic film shows a series of diffraction lines in the form of concentric circles. These lines occur as a result of the cancellation and reinforcement of the reflections from the various atomic planes of the specimen. No two different materials have been found which show identical patterns, and, hence, identification by this means is positive. However, identification by the examination of photographic film is somewhat inadequate.

The use of the Geiger-Muller counter tube in place of photographic film eliminates these inadequacies, making possible an accurate, automatic, relatively fast recording device. The Geiger-Muller tube actually counts quanta of X-ray energy reflected from the specimen. The tube is an adaptation of the type used for cosmic-ray investigations but differs in that it is much more sensitive in the X-ray range and actually produces one count for almost every X-ray quantum entering the tube.

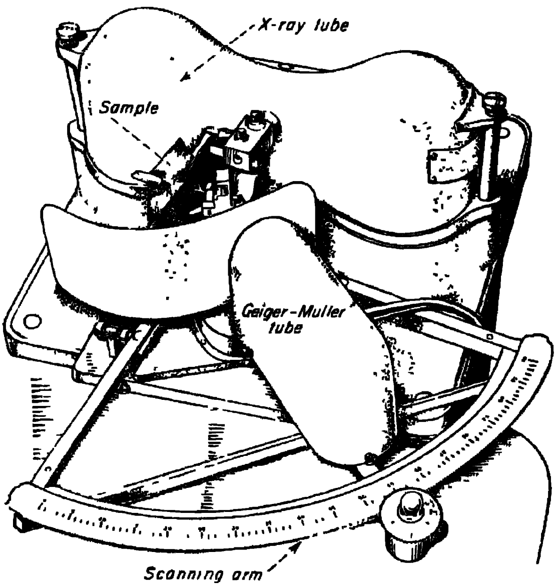
The Geiger-Muller tube comprises a sealed tube filled with an ionizable gas Radiation traveling through the gas within the tube results in partial ionization of the gas which, in turn, allows an electric current to flow through the tube. This current is amplified and put through an electronic integrating circuit so that it can be evaluated in terms of the intensity of X-radiation.

In order to obtain a diffraction pattern with the counter tube, it is necessary to scan a cross section of the diffraction cone. This is accomplished by rotating the counter tube and the sample in such a manner that the former moves at twice the angular speed of the latter so that the tube will be progressively exposed to the various beams which are reflected from the sample. As illustrated in Fig. 10-13, the counter tube is rotated through 90 deg of arc by means of a motor-driven, calibrated scanning arm (goniometer) whose speed of rotation is precisely coordinated with the chart speed of a recording potentiometer. In this way, an exact space relationship between the representative peaks and dips of reflected radiation is reproduced on the recorder chart.

Ultraviolet Spectrometer. Continuous, automatic analysis of a flowing gas or liquid now is possible with the ultraviolet spectrometer and is accomplished by recording the per cent transmission of light through the sample at the frequency selected. Major components of the instrument include a suitable light source, a monochromator, and a measuring system. Automatic analyses of more than one sample flow can be effected by means of electric time-cycle control and associated solenoid valves and fittings in the sample lines.

A tungsten lamp is used to supply light at wave lengths of from 320 to 1,000 millimicrons, while a hydrogen-discharge lamp is used to supply ultraviolet radiation from

220 to 350 millimicrons. Which lamp is used is determined solely by the wave-length-absorption characteristics of the sample under analysis. The monochromator fundamentally is an optical system serving to isolate the wave length necessary to the analysis of the sample. Light from the selected source is focused by an optical arrangement in the lamp housing so that a crescent-shaped image is projected through an aperture and into the separately housed monochromator unit. By means of adjustable slits, the desired wave lengths within the beam of light which enters the monochromator are segregated and directed through the sample cell and thence to the measuring system. The monochromator is provided with a sensitivity and a wave-length control. Light passing through the sample is intercepted by the measuring cell which comprises a phototube whose electrical characteristics are sensitive to changes in light intensity at wave lengths of the transmitted light. The phototube and its associated amplifier develop an output voltage whose value is fixed by the intensity of the light received by the phototube. By measuring and recording this voltage with a potentiometer, the per cent transmission of light through the sample is directly indicated and recorded. Potentiometers for such use generally are calibrated in per cent transmission.



Partial view of X-ray spectrometer, showing X-ray tube, sample location, Geiger-Muller tube, and scanning arm (*North American Philips Co., Inc.*)

The ultraviolet spectrometer is designed principally for recording variations in per cent transmission at a fixed wave length rather than for scanning a spectral region to record absorption spectra. Thus, the instrument can be adjusted to record continuously per cent transmission at a wave length which indicates the purity of a substance. Indications of less than 100 per cent will be produced when the substance absorbs some of the radiation; indications of 0 per cent will be produced when the substance absorbs all the radiation. Since potentiometers suitable to these measurements are available with pneumatic or electric automatic-control components, the possibilities of automatic process-stream purity control are obvious.

Infrared Spectrometer. The most important prerequisite of an infrared spectrometer is a suitable source of infrared radiation. Its final function is to measure the intensity, at various wave lengths, of a beam of such radiation which has passed through the substance under test. Before measurements of the intensity at various wave lengths can be made, the infrared beam which has been passed through the test substance must be broken up into different wave lengths and arranged in numerical sequence. This is accomplished in a fashion similar to that just described in connection with the ultraviolet spectrometer. Since infrared waves are invisible, measurement of their intensity can best be accomplished by measuring their heating effect. By measuring this effect at each wave length, the absorption pattern of the substance under test can be ascertained.

Figure 10-14 shows schematically the path of infrared radiation. Note that radiation from the Nernst glower which constitutes the source *A* of this radiation passes through a rotating shutter *B*, thence through a rock-salt window *C*, and to a condensing mirror *D*. The condensing mirror forms an image of the source in the focal plane of the concave rock-salt lens *E*. The collimated beam passes through the shutter *F* and the liquid cell compartment *G* to a lens *H* which converges the beam through the gas cell or sample compartment *I* and the lens *J* to the upper entrance slit *K* where an image of the source is formed. Radiation from the slit is collimated by a spherical mirror *L* and dispersed by the prism *M*. The desired wave length of the dispersed radiation is reflected from the Littrow mirror *N* back through the prism *M* to the collimator *L*. The collimator focuses on the plane of the exit slit (which lies just below the entrance slit), and the plane mirror *O* reflects the beam against the condensing mirror *P* which focuses an image of the exit slit on the energy receiver *Q* where the heating effect of the wave length, which is selected in accordance with the position of the Littrow mirror *N*, is measured by means of a thermocouple located in the energy receiver. By rotating the Littrow mirror, different wave lengths of the dispersed beam are directed to the energy receiver where their intensity is measured by the thermocouple and recorded by a potentiometer. Thus, a record of the absorption pattern of the substance under test is obtained.

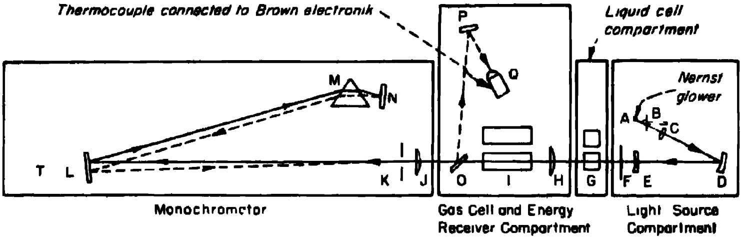


FIG. 10-14. Optical path in infrared spectrometer. (*Perkin-Elmer Corp.*)

AUTOMATIC PROCESS CONTROL

Despite the fact that automatic process control involves the use of highly technical and often complex mechanisms, the *application* of instruments and automatic controls to an industrial process has only relatively recently graduated from an art to a science. Even today, the science has not been sufficiently simplified and condensed to the point where the average engineer can apply instruments to certain types of process without utilizing cut-and-try methods. Fortunately, during the past few years, a few valuable textbooks on instrumentation have appeared; hundreds of fine technical articles on the subject have been written, a society devoted solely to instrumentation has been organized on a national scale; and, of utmost importance, the ASME has compiled a long list of well-classified instrument terms and definitions. Progress thus is being made and undoubtedly will continue at an even more rapid pace.

Considerable chaos was in evidence in the field of instrumentation before the adoption of a consistent nomenclature. There were as many definitions for a control term as there were manufacturers of such controls. Wherever possible, the new nomenclature is used throughout this section. Unfortunately, many automatic-control terms are somewhat complex and, to the uninitiated, often quite difficult to comprehend. To aid the plant engineer who has had little contact with the new terms, the most important and basic of the ASME definitions are illustrated diagrammatically. In some cases, the definitions appear wordy and cumbersome to handle, but a further analysis will prove that oversimplification would not be wise in these early stages in the development of a nomenclature.

Most of the following sections on terminology are taken from "Graphical Representation and Analysis of Automatic Control Terminology" by J. G. Horn, Development Engineer, Brown Instruments Division, Minneapolis-Honeywell Regulator Co., Philadelphia, Pennsylvania, and were presented at the September, 1946, meeting of ASME in Pittsburgh.

Elements of the Automatically Controlled Process. A certain number of minimum separate elements are required to control a process automatically. These elements are defined (ASME) as follows:

101 An *automatic controller* is a mechanism which measures the value of a variable quantity or condition and operates to correct or limit deviation of this measured value from a selected reference. It includes both the measuring means and the controlling means. (*Automatic regulator* is a synonymous term.)

301 A *process* comprises the collective functions performed in and by the equipment in which a variable is to be controlled.

303 The *controlled variable* is that quantity or condition which is measured and controlled.

304 The *controlled medium* is that process energy, or material in which a variable is controlled.

305 The *manipulated variable* is that quantity or condition which is varied by the automatic controller so as to affect the value of the controlled variable.

306 The *control agent* is that process energy, or material of which the manipulated variable is a condition or characteristic.

The automatic controller comprises the measuring means and the controlling means as shown in Fig. 10-15, illustrating a fluid-heat-exchange process with automatic temperature control. The measuring means may consist of a potentiometer pyrometer system measuring the temperature of the controlled medium, which is a fluid leaving the process. The controlling means consists of a mechanism embodying predetermined controller action affecting the condition or characteristic of the control agent, which is a fluid entering the process. This action upon the control agent produces an effect which corrects or limits the temperature of the controlled medium.

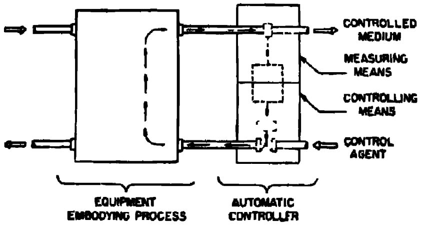


FIG. 10-15. Schematic diagram of process and automatic controller.

The process equipment and the automatic controller are installed in such a manner that the initial action within the process and the counter action of the automatic controller form a closed loop or circuit of action. The term "equipment" as embodied in this definition does not include automatic-control equipment.

The process is an operation or series of operations in which the value of a quantity or condition is controlled. It includes all functions which directly or indirectly affect the value of the controlled variable.

Figure 10-16 illustrates the same fluid-heat exchanger shown in Fig. 10-15, with additional detail of the process elements pertinent to automatic control. The controlled variable (temperature) of the controlled medium (fluid leaving the process) is measured by the automatic controller which acts in a predetermined manner to vary the value of the manipulated variable (flow) of the control agent (fluid entering the process). The action of the automatic controller restores, or tends to restore, the value of the controlled variable to the desired value.

The controlled variable is a condition or characteristic of the controlled medium. The manipulated variable is a condition or characteristic of the control agent.

Process Characteristics. All industrial processes possess characteristics which determine to a large degree how easily automatic control can be applied and the results obtained from automatic control. If all processes were alike, automatic controllers of one design or one mode would suffice. Unfortunately, processes vary widely in the number of characteristics which they possess and the extent to which these characteristics are manifest. Some of the basic characteristics of a process are defined (ASME) below:

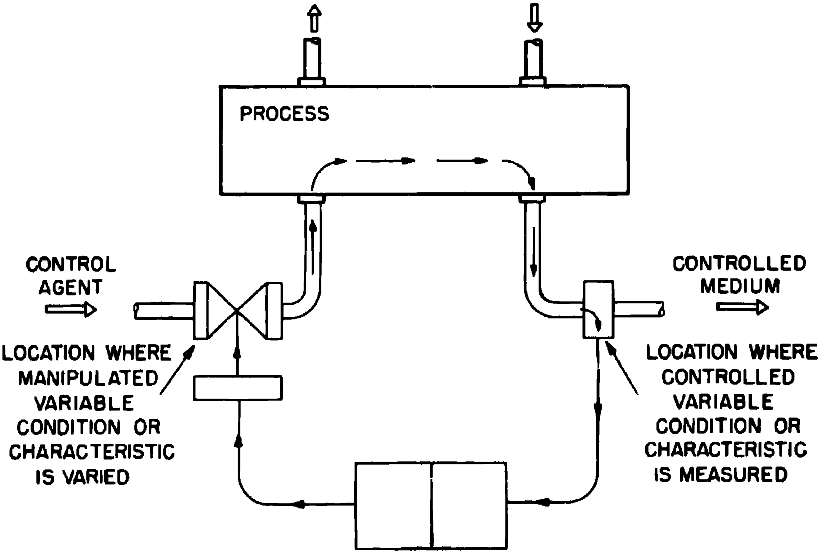

PROCESS
CONTROL AGENT
CONTROLLED MEDIUM
LOCATION WHERE MANIPULATED VARIABLE CONDITION OR CHARACTERISTIC IS VARIED
LOCATION WHERE CONTROLLED VARIABLE CONDITION OR CHARACTERISTIC IS MEASURED

FIG. 10-16. Schematic diagram showing relationship of process elements.

201 *Capacity* is a measure of the maximum quantity of energy or material which can be stored. It is measured in units of quantity.

202 *Capacitance* is the change in quantity contained per unit of change in some reference variable. It is measured in units of quantity divided by the reference variables.

203 *Resistance* is opposition to flow. It is measured in units of potential change required to produce unit change in flow.

204 *Dead time* is any definite delay period between two related actions. It is measured in units of time.

302 *Self-regulation* is a sustained reaction inherent in the process which assists or opposes the establishment of equilibrium.

Various forms of the term "lag" are commonly used to express the delaying or retarding effect inherent in the dynamic characteristics of a controlled system. The characteristics involved in this effect are capacitance, resistance, and dead time, which may exist separately or in combination. These three terms cover the basic concepts involved and, in the interest of clarity, should be used in place of the less exact term "lag."

The type of capacity is determined by the energy or material contained. Likewise the type of capacitance is determined both by the type of quantity contained and by the type of reference variable. Several types of capacity and capacitance may exist together in one process.

A comparison of capacity and capacitance is illustrated in Fig. 10-17. In this example, the capacities are equal, the capacitances are unequal, and the liquid-level surface areas do not vary with change in liquid head. The capacity or "static quantity" is the cubic volume of liquid contained in each vessel. The volume capacitance or "dynamic quantity" is equivalent to the liquid-level surface area of each vessel.

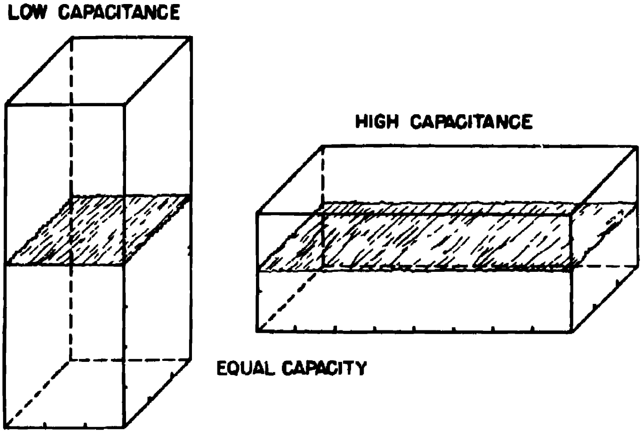


FIG. 10-17 Relationship of liquid volume capacity to volume capacity.

In this example, the capacitance is equal to the change of volume capacity divided by the change of reference variable, the liquid head.

If the shape of the vessel causes the liquid-level surface area to vary with change in liquid head, the capacitance will likewise vary with head. Vessels of different shape have equal capacities if the quantities of material contained are equal. Vessels of different shape have equal volume capacitances if the liquid-level surface areas are equal.

Another comparison between capacity and capacitance is exemplified in a compressed-air tank. The weight capacity at a stated pressure is the weight of air within

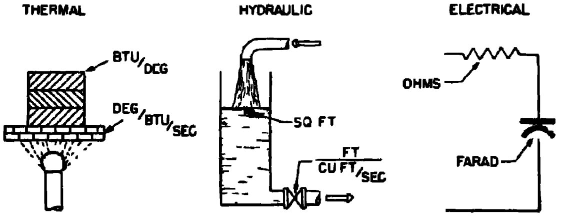


FIG. 10-18. Resistance-capacitance analogies.

the tank at that pressure. The weight capacitance with respect to pressure is the change in contained weight of air divided by the corresponding change in pressure of the tank.

Combinations of resistances and capacitances are frequently found in many industrial processes. Figure 10-18 illustrates single resistance-capacitance analogies for thermal, hydraulic, and electrical circuits.

Thermal resistance is measured in degrees per Btu per second. It is equal to the reciprocal of thermal conductance. Thermal capacitance is measured in Btu per degree and is equal to the specific heat (Btu per degree per pound) times the mass (pound).

The hydraulic resistance illustrated is measured in feet of liquid head per unit volume or mass flow of liquid from the vessel. The hydraulic capacitance is measured in square feet of liquid-level surface or pounds of total liquid per foot of liquid head.

The electrical resistance is the voltage per unit ampere or ohm. The electrical capacitance is the farad.

Frequently, the resistance is not a linear quantity, since different values of resistance may be obtained for different values of potential. Likewise, the capacitance may not be a linear quantity with changes in reference variable. Therefore, the value of the capacitance should be based on a definite value of reference variable

The dimensional units of the basic characteristics for several types of processes are summarized in Table 10-1. In this tabulation, capacity represents a static quantity, capacitance, a dynamic quantity. The potential represents a condition at a point in the process which determines the energy at the point. Resistance is the potential per unit flow The controlled variable of the process may be the potential, capacity, or other condition of the process.

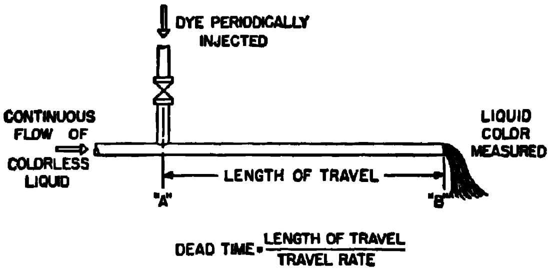

DYE PERIODICALLY INJECTED
CONTINUOUS FLOW OF COLORLESS LIQUID
LIQUID COLOR MEASURED
LENGTH OF TRAVEL
"A"
"B"
DEAD TIME = LENGTH OF TRAVEL / TRAVEL RATE

Fig. 10-19. Schematic diagram showing effect of dead time.

Dead time is illustrated in Fig. 10-19 by the use of a fluid-flow process. A small amount of dye is injected periodically at point *A* into a continuously flowing colorless liquid. The liquid mixture flows through a long length of pipe to point *B* before its color is detected The dead time is the time required for the dye to travel from *A* to *B* and is equal to the length of travel divided by the travel rate Point *A* may correspond to the location within a process where a variable change is initiated, and point *B* may correspond to the point at which the variable is measured. In this example, the measuring element does not respond to variable change during the dead-time period of the process

Table 10-1. Units of Measurement of Basic Process Characteristics

Units of	Thermal	Pressure	Liquid level	Electrical
Capacity	Btu	Cubic foot	Cubic foot-pounds	Coulomb
Potential	Degree	Lb psi	Foot	Volt
Capacitance	$\frac{\text{Btu}}{\text{Deg}}$	$\frac{\text{Cu ft}}{\text{Psi}}$	$\frac{\text{Cu ft}}{\text{Ft}}$ or $\frac{\text{Lb}}{\text{Ft}}$	$\frac{\text{Coulomb}}{\text{Volt}}$ = farad
Resistance	Deg/Btu/sec	$\frac{\text{Psi}}{\text{Cu ft/sec}}$	$\frac{\text{Feet}}{\text{Cu ft/sec}}$ or $\frac{\text{Feet}}{\text{Lb/sec}}$	$\frac{\text{Volt}}{\text{Coulomb/sec}}$ = ohms

Elements of Automatic Controllers. The fluid-heat-exchange process illustrated in Fig. 10-20 illustrates the basic elements of an automatic controller. These elements are defined (ASME) as follows:

701 The *measuring means* consists of those elements of an automatic controller which are involved in ascertaining and communicating to the controlling means either the value of the controlled variable the error, or the deviation.

701a The *primary element* is that portion of the measuring means which first either utilizes or transforms energy from the controlled medium to produce an effect in response to change in the value of the controlled variable. The effect produced by the primary element may be a change of pressure, force, position, electrical potential, or resistance.

702 The *controlling means* consists of those elements of an automatic controller which are involved in producing a corrective action.

702a A *power unit* is a portion of the controlling means which applies power for operating the final control element.

702b The *final control element* is that portion of the controlling means which directly changes the value of the manipulated variable.

Referring to Fig. 10-20, it may be assumed that the measuring means is a thermometer system, and the controlling means consists of a pneumatic-type controller mechanism and a diaphragm-operated control valve.

A change in value of the controlled variable, the temperature of fluid leaving the process, first is ascertained by the primary element (the thermometer bulb) of the measuring means. This change is communicated by the measuring means to the controlling means. The controlling means produces a corrective action affecting the value of the manipulated variable, which is the flow of fluid entering the process. The corrective action operates through the final control element (the control valve) which directly changes the value of the manipulated variable. The power unit (the diaphragm motor) supplies power to the final control element.

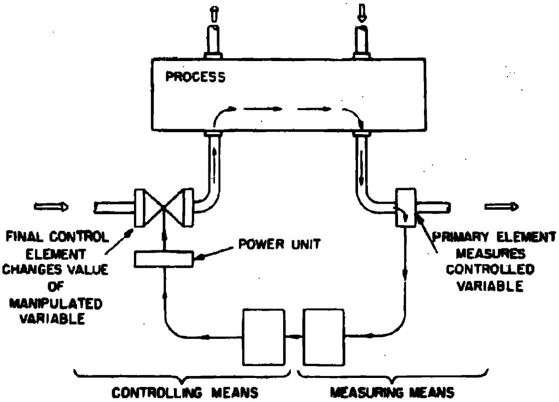

PROCESS
FINAL CONTROL ELEMENT CHANGES VALUE OF MANIPULATED VARIABLE
POWER UNIT
PRIMARY ELEMENT MEASURES CONTROLLED VARIABLE
CONTROLLING MEANS
MEASURING MEANS

Fig. 10-20. Schematic diagram showing relationship of automatic controller elements.

Other examples of primary elements are the thermocouple for measuring temperature; the diaphragm and bellows for measuring pressure; and the orifice, rotameter, venturi, and flow nozzle for measuring rate of flow. All these elements first utilize or transform energy from the controlled medium while performing the duties of measurement.

An interesting example of a final control element is the control switch of a pyrometer for electric furnaces, where all current to the furnace is carried by the control switch without the use of auxiliary relays. In this case, the switch is the final control element. If an auxiliary relay were used, this relay would become the final control element.

Characteristics of Automatic Controllers. Just as the process proper has certain characteristics which affect control, so does the automatic controller. These characteristics are defined (ASME) and described briefly below.

102 A *self-operated controller* is one in which all the energy necessary to operate the final control element is derived from the controlled medium through the primary element.

701b A ***self-operated measuring means*** is one in which all the energy necessary to actuate the controlling means of an automatic controller is derived from the controlled medium through the primary element.

702c A *self-operated controlling means* is one in which all the energy necessary to operate the final control element is derived from the measuring means.

103 A *relay-operated controller* is one in which the energy transmitted through the primary element is either supplemented or amplified for operating the final control element by employing energy from another source.

701c A *relay-operated measuring means* is one in which the energy transmitted through the primary element is either supplemented or amplified for actuating the controlling means of an automatic controller by employing additional energy.

702d A *relay-operated controlling means* is one in which the energy transmitted from the measuring means is either supplemented or amplified for operating the final control element by employing additional energy.

A self-operated controller is illustrated in Fig. 10-21 which shows a diaphragm-type pressure-reducing valve operated by energy obtained only from the controlled medium through the primary element. A self-operated controller must have both self-operated measuring means and self-operated controlling means. It must derive all its energy through the primary element and may not derive energy separately from the controlled medium to obtain a relay-operated control.

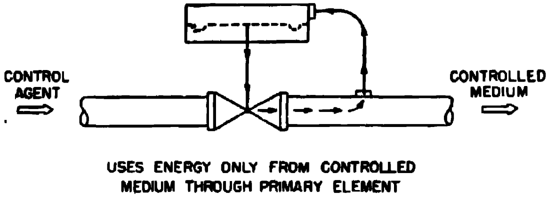

CONTROL AGENT
CONTROLLED MEDIUM
USES ENERGY ONLY FROM CONTROLLED MEDIUM THROUGH PRIMARY ELEMENT

Fig. 10-21. Schematic diagram of self-operated controller.

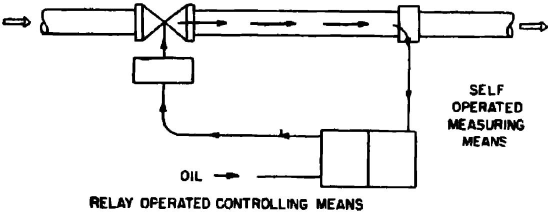

SELF OPERATED MEASURING MEANS
OIL
RELAY OPERATED CONTROLLING MEANS

Fig. 10-22. Schematic diagram of relay-operated controller, with self-operated measuring means and relay-operated controlling means.

The self-operated measuring means of the pressure-reducing valve shown in Fig. 10-21 is the diaphragm, which also is the primary element. The diaphragm measures the pressure of the controlled variable and transforms this pressure to a force.

The self-operated controlling means of Fig. 10-21 consists of the diaphragm of the measuring means and the valve body and plug. The diaphragm force directly operates the valve plug, and the relative positions of the valve body and plug affect the value of the controlled variable. Note that in this example the controlled medium and the control agent are the same.

The relay-operated controller is illustrated in Figs. 10-22 and 10-23. A relay-operated controller must have at least one source of energy other than that derived from the controlled medium through the primary element to operate the final control element. A relay-operated controller either may have a self-opperated measuring means and a relay-operated controlling means; a relay-operated measuring means and a self-operated controlling means; or a relay-operated measuring means and a relay-operated controlling means. Figure 10-22 shows a self-operated measuring means and a relay-operated controlling means. Figure 10-23 shows a relay-operated measuring means and a relay-operated controlling means.

The self-operated measuring means of Fig. 10-22 may be a millivoltmeter pyrometer or a pressure-actuated thermometer, in which case the energy derived from the controlled medium through the primary element is transformed before actuating the controlling means. However, this energy is neither supplemented nor amplified by energy from another source.

If in Fig. 10-22 the nomenclature should be reversed, *i.e.*, a relay-operated measuring means and a self-operated controlling means, the resultant combination also would be a relay-operated controller. An example of such a combination would be a potentiometer pyrometer with balancing motor operating a final control element by means of a direct mechanical connection. In this case, the final control element may be an electrical rheostat or a small needle valve. The relay-operated measuring means, in this example the potentiometer, employs additional energy for actuating the controlling means other than the energy transmitted through the primary element.

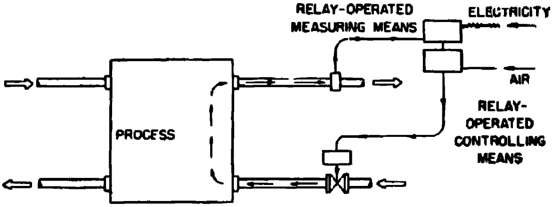

RELAY-OPERATED MEASURING MEANS
ELECTRICITY
AIR
RELAY-OPERATED CONTROLLING MEANS
PROCESS

Fig. 10-23. Schematic diagram of relay-operated controller, with relay-operated measuring means and relay-operated controlling means.

Automatic-control Characteristics. Closely associated with the characteristics of automatic controllers are the characteristics of automatic control itself. Some of these characteristics are defined (ASME) and described below.

405 *Cycling* is a periodic change of the controlled variable from one value to another. ("Oscillation" is a synonymous term.)

104 The *desired value* is the value of the controlled variable which it is desired to maintain.

105 The *set point* is the position to which the control-point setting mechanism is set.

106 The *control point* is the value of the controlled variable which, at any instant, the automatic controller operates to maintain.

401 *Error* is the difference between the instantaneous value and the desired value of the controlled variable.

402 *Deviation* is the difference between the instantaneous value of the controlled variable and the value of the controlled variable corresponding with the set point.

403 *Offset* is a sustained deviation due to an inherent characteristic of positioning-controller action and is the difference existing at any time between the control point and the value of the controlled variable corresponding with the set point.

Cycling of the controlled variable is illustrated in Fig. 10-24. It is a continuous change of the controlled variable alternately increasing and decreasing in value. The time interval between two successive peaks of the cycle remains substantially constant nd is termed the "period" of the cycle. The amplitude of cycling is the maximum departure of the controlled variable from the average value of the cycle.

Three types of cycling of the controlled variable are shown in Fig. 10-24. Cycling in which the amplitude gradually decreases may be obtained, for example, with proportional-speed floating controller action. Cycling in which the amplitude is constant may be obtained with two-position controller action. Cycling in which the amplitude gradually increases is not suitable for automatic control.

The desired value of the controlled variable is determined by the operating requirements of the process. Usually, the set point of the automatic controller is adjusted so that the control point and the desired value will coincide.

The set point of an automatic controller is the position of the control-point-setting mechanism translated into units of the controlled variable. Generally, a set-point scale is provided. The set point may be varied manually or by automatic means, such as in time-schedule or ratio control. The control point is the value of the controlled variable maintained by the controller.

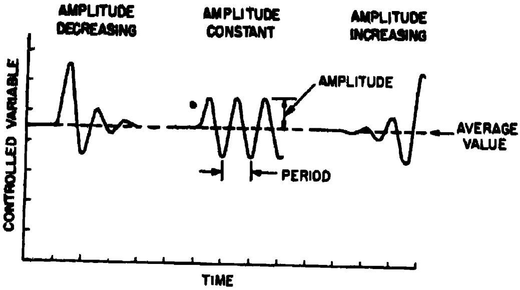


FIG. 10-24. Schematic diagram showing cycling of the controlled variable.

In some types of automatic controllers, *e.g.*, those with two-position differential gap or floating with neutral controller action, the control point becomes a control range of the controlled variable, and the set point is generally selected as the center of this range of values.

Figure 10-25 illustrates the condition at which the desired value, set point, and control point coincide. This type of controlled-variable record may be obtained by a proportional-plus-reset controller action correcting for a sustained change of process load. The set point was initially adjusted to coincide with the desired value; and the set point and control point coincide because of the floating-type controller action employed. In floating with neutral-zone controller action, the control point and the set point always coincide. The error and deviation coincide because of their respective associations with the desired value and set point.

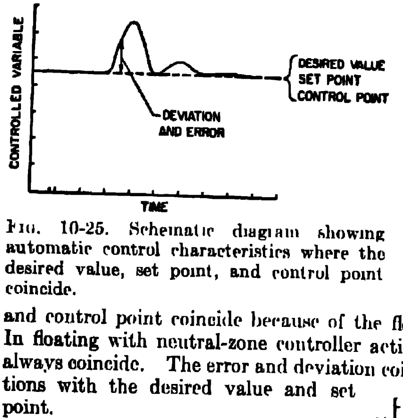


FIG. 10-25. Schematic diagram showing automatic control characteristics where the desired value, set point, and control point coincide.

Figure 10-26 illustrates the condition at which the desired value and set point coincide, and the control point differs from the set point by the amount of offset. This type of controlled-variable record may be obtained by a proportional-position controller action correcting for a sustained change of process load. The deviation and error coincide because the set point was initially adjusted to coincide with the desired value. In positioning-type controller action, the control point may lie anywhere within a predetermined range of values of the controlled variable. The control point may then differ from the set point by the amount of offset.

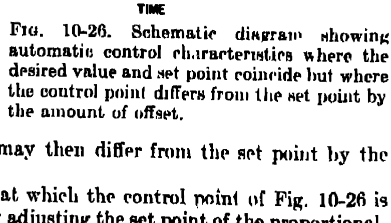


FIG. 10-26. Schematic diagram showing automatic control characteristics where the desired value and set point coincide but where the control point differs from the set point by the amount of offset.

Figure 10-27 illustrates the condition at which the control point of Fig. 10-26 is made to coincide with the desired value by adjusting the set point of the proportional-position controller of Fig. 10-26. The error which is associated with the desired value now differs from the deviation which is associated with the set point.

Modes of Automatic Control. In the following definitions, it is assumed that the automatic controller operates ideally, *i.e.*, it detects infinitesimal variations of the controlled variable and responds instantaneously in accordance with its predetermined action or mode. All numbered definitions are ASME.

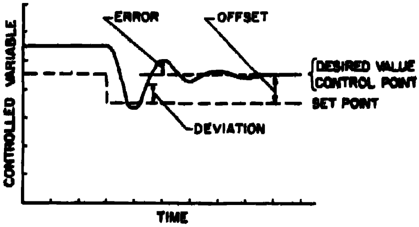


Fig. 10-27. Schematic diagram showing automatic control characteristics where the control point shown in Fig. 10-26 is made to coincide with the desired value by adjusting the set point of the proportional-position controller of Fig. 10-26.

501*ab* *Two-position single-point action* is that in which a final control element is moved from one of two fixed positions to the other at a single value of controlled variable.

501*aa* *Two-position differential-gap action* is that in which a final control element is moved from one of two fixed positions to the other when the controlled variable reaches a predetermined value from one direction, and subsequently is moved to the other position only after the variable has passed in the opposite direction through a range of values to a second predetermined value.

602 *Differential gap*, applying to two-position controller action, is the smallest range of values through which the controlled variable must pass in order to move the final control element in succession to both of its fixed positions.

501*b* *Multiposition action* is that in which a final control element is moved to one of three or more predetermined positions, each corresponding to a definite range of values of the controlled variable.

502*aa* *Single-speed floating action* is that in which a final control element is moved at a single rate.

601 *Neutral zone* is a predetermined range of values of the controlled variable in which no corrective action occurs.

601 *Floating speed*, applying to single or multispeed floating controller action, is the rate of motion of the final control element.

502*ab* *Multispeed floating action* is that in which a final control element is moved at two or more rates, each corresponding to a definite range of values of the controlled variable.

Two-position, single-point action is illustrated in Fig. 10-28. The two-position action, commonly termed, "on-off," is obtained at a point value of the controlled variable. The position of this point is adjusted within the range of the controlled variable by the set-point adjustment. The ON or OFF position of the final control element is determined by the direction of change of the controlled variable from the set point.

This type of controller action may be considered the equivalent of proportional-position action with zero proportional band; floating action with zero neutral zone and infinite floating speed; or two-position action with zero differential gap.

Two-position differential-gap action is shown in Fig. 10-29. The position of the final control element as the controlled variable passes the set point is determined by the direction of motion of the controlled variable as it passes through the differential gap. In this case, the control point becomes the differential gap, a fixed range of values of the controlled variable. The controlled variable must pass through the differential gap before a change of controller position takes place. When the controlled variable is within the differential gap, the controller maintains its previous position. The differential gap is commonly expressed in per cent of controller-scale range, and its position within the range is determined by the set-point adjustment.

Multiposition controller action is illustrated in Fig. 10-30. The three positions of the final control element correspond to three predetermined ranges of the controlled variable.

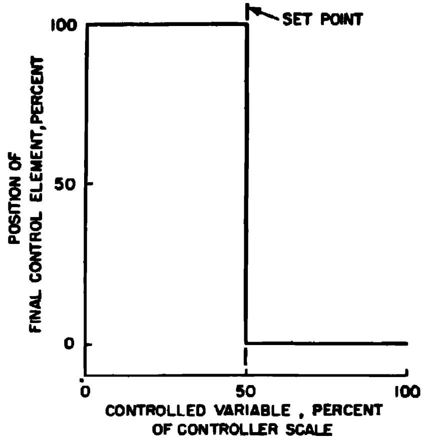


FIG. 10-28. Schematic diagram showing two-position single-point controller action

Single-speed floating action is illustrated in Fig. 10-31. The floating speed is the rate of motion of the final control element, which is a predetermined single value. It is generally expressed as the per cent of full-range motion of the final control element per minute. A neutral zone, in which no motion of the final control element occurs, is employed. The control point becomes the neutral zone. The direction of motion of the final control element is determined by the direction of change of the controlled variable from the neutral zone. The neutral zone is commonly expressed in per cent of controller-scale range, and its position within th range is determined by the set-point adjustment.

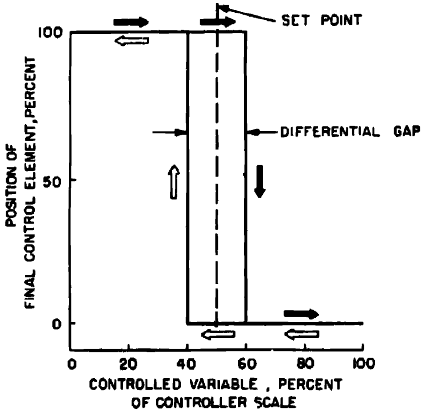


FIG. 10-29. Schematic diagram showing two-position differential-gap controller action.

Multispeed floating controller action is illustrated in Fig. 10-32. It differs from single-speed floating action in that there is more than one rate of motion of the final

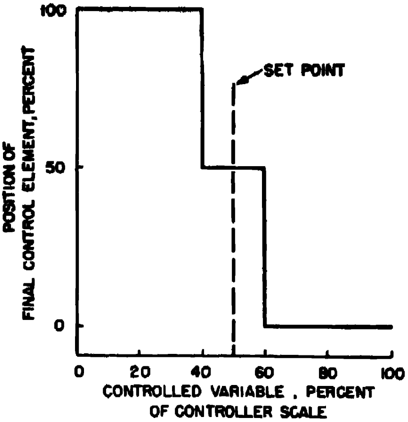


Fig. 10-30. Schematic diagram showing multiposition controller action.

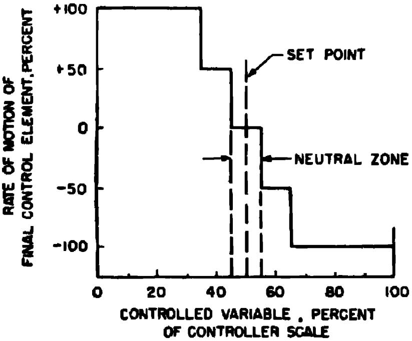


Fig. 10-31. Schematic diagram showing single-speed floating controller action.

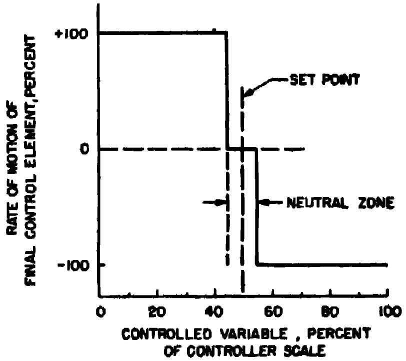


Fig. 10-32. Schematic diagram showing multispeed floating controller action.

control element. Each rate of motion corresponds to a predetermined range of the controlled variable.

501c *Proportional-position action* is that in which there is a continuous linear relation between value of the controlled variable and position of a final control element.

603 *Proportional band*, applying to proportional-position controlled action, is the range of values of the controlled variable which corresponds to the full operating range of the final control element.

501d *Average-position action* is that in which there is a predetermined relation between value of the controlled variable and the time-average position of a final control element which is moved periodically from one of two fixed positions to the other.

502ad *Floating average-position action* is that in which there is a predetermined relation between value of the controlled variable and rate of change of the time-average position of a final control element which is moved periodically from one of two fixed positions to the other.

Proportional-position controller action is illustrated in Fig. 10-33. The proportional band is adjustable and is shown equal to 40 per cent of the controlled-variable controller-scale range.

The proportional band is commonly expressed in per cent of controller-scale range or, particularly in the absence of a controller scale, in units of the controlled variable. The set point is also adjustable and is shown at 40 per cent of the controller-scale range. The intersection of the set point and the proportional-position action line determines the position of the final control element when the controlled variable is at the set point. In this illustration (Fig. 10-33), the position of the final control element is at 50 per cent. As the controlled variable deviates from the set point, the final control element takes a position directly proportional to the deviation, as indicated by the proportional-position action line.

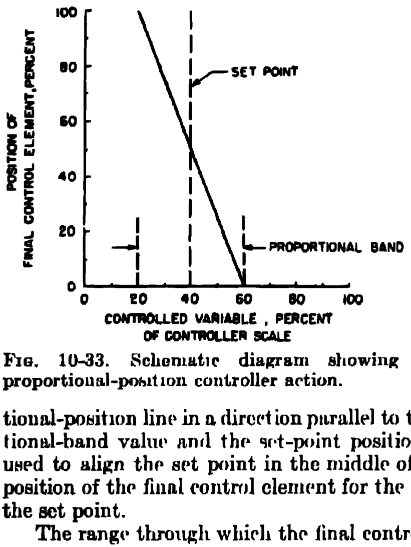


FIG. 10-33. Schematic diagram showing proportional-position controller action.

Proportional-position controllers are frequently equipped with a manual adjustment for shifting the proportional-position line in a direction parallel to the controlled-variable axis. The proportional-band value and the set-point position are not changed. This adjustment is used to align the set point in the middle of the proportional band and to adjust the position of the final control element for the purpose of aligning the control point with the set point.

The range through which the final control element may operate may be less than 100 per cent. It is dependent upon the relative values of the proportional band and the position of the band on the controller scale. As the value of the proportional band is increased above 100 per cent controller scale, the range of the final control-element position is decreased below 100 per cent for any position of the set point.

Average-position controller action is equivalent to a combination of positioning controller action and a two-position final control-element action operating in a periodic manner. The cyclic period of the two-position final control-element action is predetermined and fixed. The change from one position, "time on," to the other position, "time off," is nearly instantaneous.

The average-position action, illustrated in Fig. 10-34, is of the proportional type. The percentage "time on" or "time off" is proportional to the deviation of the controlled variable from the set point.

Floating average-position controller action is equivalent to a combination of floating controller action and a two-position final control-element action operating in a periodic manner.

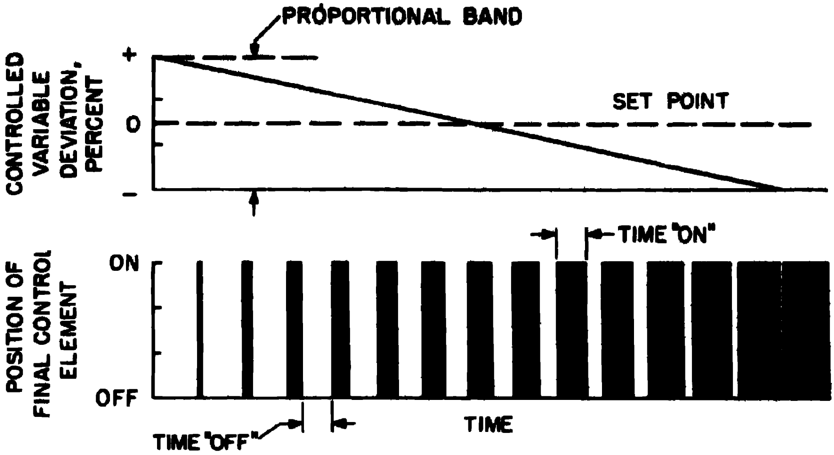

PROPORTIONAL BAND
CONTROLLED VARIABLE DEVIATION, PERCENT
+
0
−
SET POINT
POSITION OF FINAL CONTROL ELEMENT
ON
OFF
TIME "ON"
TIME "OFF"
TIME

FIG. 10-34. Schematic diagram showing average-position controller action of the proportional type.

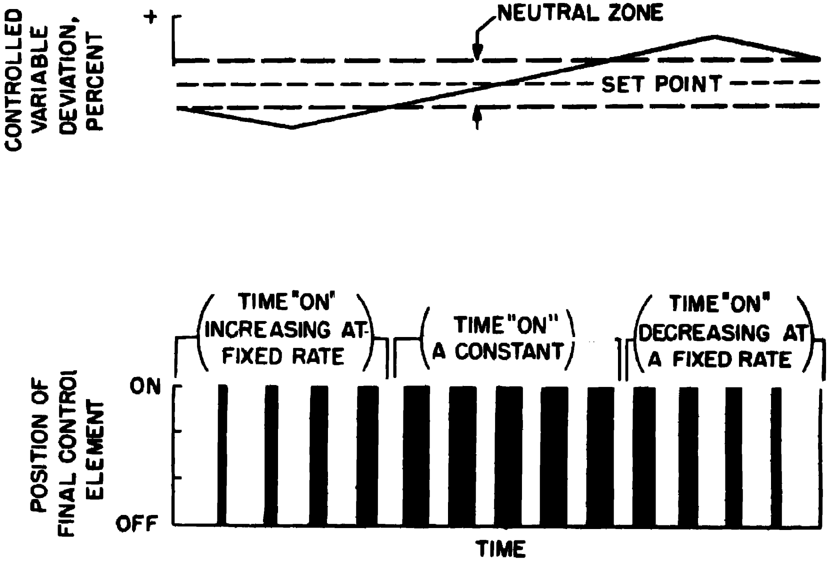

CONTROLLED VARIABLE DEVIATION, PERCENT
+
NEUTRAL ZONE
SET POINT
TIME "ON" INCREASING AT FIXED RATE
TIME "ON" A CONSTANT
TIME "ON" DECREASING AT A FIXED RATE
POSITION OF FINAL CONTROL ELEMENT
ON
OFF
TIME

FIG. 10-35. Schematic diagram showing floating average-position controller action of the single-speed type.

The floating average-position controller action, illustrated in Fig. 10-35, is of the single-speed type. The rate of change of the percentage "time on" or "time off" is a fixed value when the controlled variable within the neutral zone, the percentage "time on" or "time off," is constant.

502ac *Proportional-speed floating action* is that in which there is a continuous linear relation between value of the controlled variable and rate of motion of a final control element.

605 *Floating rate*, applying to proportional-speed floating controller action, is the rate of motion of the final control element corresponding to a specified deviation.

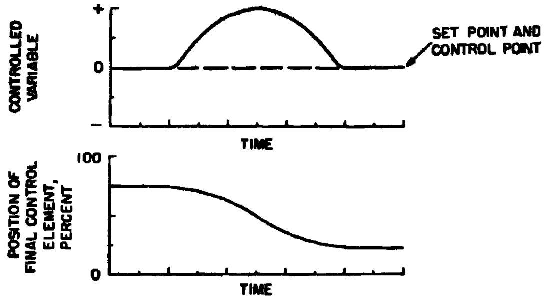


FIG. 10-36. Schematic diagram showing the response of a proportional-speed floating-action controller.

The response of a proportional-speed floating-action controller is illustrated in Fig. 10-36. The deviation of the controlled variable is caused by a sustained change in process load. The controlled variable always returns to the set point at the completion of the corrective action of the controller; this is a characteristic of all floating-type controller actions. The final control element takes a new position to correct for the process-load change. The manner in which the corrective action is accomplished is a function of the controlled-variable deviation, as well as the floating-rate adjustment of the controller action.

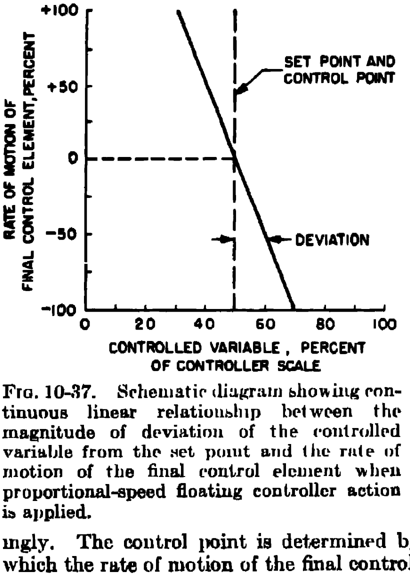


FIG. 10-37. Schematic diagram showing continuous linear relationship between the magnitude of deviation of the controlled variable from the set point and the rate of motion of the final control element when proportional-speed floating controller action is applied.

Figure 10-37 shows the continuous linear relationship between the magnitude of deviation of the controlled variable from the set point and the rate of motion of the final control element, when proportional-speed floating controller action is applied. The slope of the curve of Fig. 10-37 is determined by the value of the floating rate of the controller action. If the set point is shifted along the axis of the controlled variable, the controlled point moves correspondingly. The control point is determined by the value of the controlled variable at which the rate of motion of the final control element is zero.

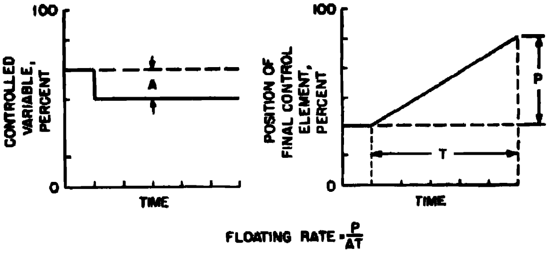


FIG. 10-38. Schematic diagram showing the floating rate of proportional-speed floating controller action.

The floating rate of proportional-speed floating controller action is illustrated in Fig. 10-38. It is the rate of change of position of the final control element divided by the controlled-variable deviation initiating the corrective action. Floating rate is commonly expressed in per cent of full range motion of the final control element per minute per per cent deviation.

504aa *Proportional-plus-reset action* is that in which proportional-position action and proportional-speed floating action are combined.

606 *Reset rate*, applying to proportional-plus-reset controller action and proportional-plus-reset-plus-rate controller action, is the number of times per minute that the effect of the proportional-position action upon the final control element is repeated by the proportional-speed floating action.

503a *Rate action* is that in which there is a continuous linear relation between rate of change of the controlled variable and position of a final control element.

504b *Proportional-plus-derivative action* is that in which proportional-position action and derivative action are combined.

607 *Rate time*, applying to proportional-plus-rate controller action and proportional-plus-reset-plus-rate controller action, is the time interval by which the rate action advances the effect of the proportional-position action upon the final control element.

504ca *Proportional-plus-reset-plus-rate action* is that in which proportional-position action, proportional-speed floating action, and rate action are combined.

The response of a proportional-plus-reset action controller is illustrated in Fig. 10-39. The deviation of the controlled variable is caused by a sustained change in process load. The controlled variable returns to the set point at the completion of the corrective action of the controller owing to the floating-type controller-action component employed. The final control element takes a new position corresponding to the new value of the process load. The manner in which the corrective action is accomplished is a function of the controlled-variable deviation, as well as the adjustments of the proportional band and the reset rate. The proportional-band adjustment simultaneously affects the proportional-speed floating action in such a manner that the reset remains constant at its set value.

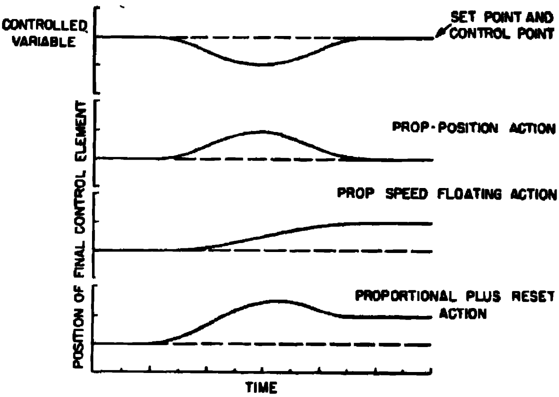

CONTROLLED VARIABLE
SET POINT AND CONTROL POINT
POSITION OF FINAL CONTROL ELEMENT
PROP-POSITION ACTION
PROP SPEED FLOATING ACTION
PROPORTIONAL PLUS RESET ACTION
TIME

FIG. 10-39. Schematic diagram showing the response of a proportional-plus-reset action controller.

When proportional-speed floating action is combined with proportional-position action, the proportional-speed floating action is termed "reset action."

Reset rate of a proportional-plus-reset controller is illustrated in Fig. 10-40. In this illustration, proportional-position action is obtained by an instantaneous change in set-point position with the other variables remaining constant. Reset rate is the rate of change of position of the final control element resulting from the proportional-speed floating-action component, divided by the change in position of the final control element resulting from the proportional-position action component, with the same deviation in both cases. The two controller actions are initiated at the same instant.

Reset rate is commonly expressed as the number of "repeats" per minute originating from a given deviation of the controlled variable. Reset rate is determined by dividing (1) the travel of the final control element in 1 min due to the effect of the proportional-speed floating action by (2) the travel of the final control element due to the effect of the proportional-position action.

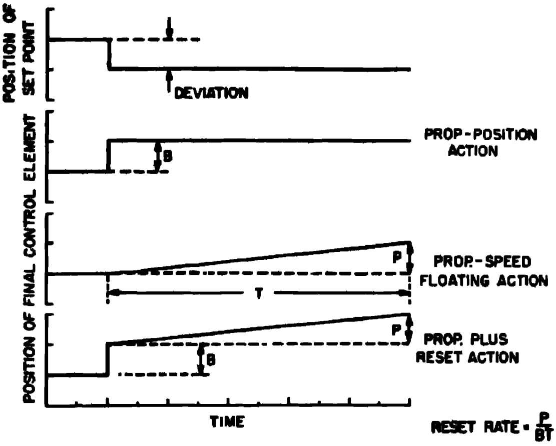


Fig. 10-40. Schematic diagram showing the reset rate of a proportional-plus-reset controller.

The reset rate is fixed by a manually set adjustment; it is independent of controlled-variable deviation and the proportional-band adjustment of the controller.

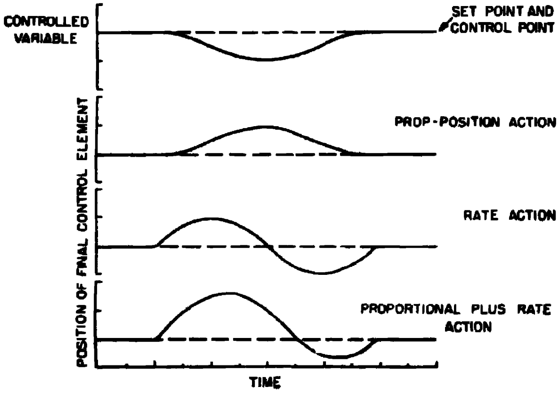

CONTROLLED VARIABLE
SET POINT AND CONTROL POINT
POSITION OF FINAL CONTROL ELEMENT
PROP-POSITION ACTION
RATE ACTION
PROPORTIONAL PLUS RATE ACTION
TIME

Fig. 10-41. Schematic diagram showing the response of a proportional-plus-rate action controller.

Proportional plus rate action is that in which proportional-position action and rate action are combined. The rate-action component employs a derivative of rate of change of the controlled variable and the position of the final control element.

The response of a proportional-plus-rate action controller is illustrated in Fig. 10-41. The deviation of the controlled variable is caused by a temporary process-load change of negligible duration. The controlled variable returns to the set point, and the final control element returns to its original position. If the process-load change were sustained, the controlled variable would line out at a new control point which would differ from the set point by the amount of offset. The manner in which the corrective action is accomplished is a function of the controlled-variable deviation, as well as the adjustments of the proportional band and rate time.

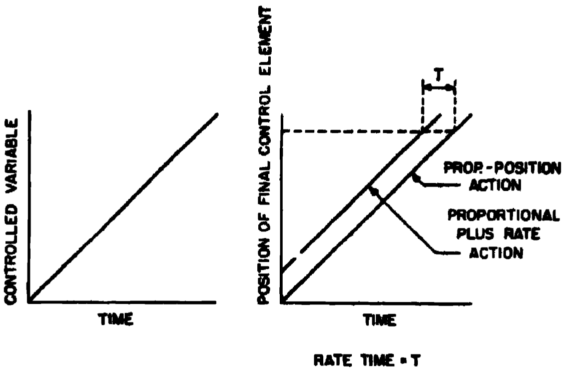

CONTROLLED VARIABLE
TIME
POSITION OF FINAL CONTROL ELEMENT
T
PROP.-POSITION ACTION
PROPORTIONAL PLUS RATE ACTION
TIME
RATE TIME = T

Fig. 10-42. Schematic diagram showing the rate time of a proportional-plus-rate action controller.

Rate time of a proportional-plus-rate action controller is illustrated in Fig. 10-42. It is the advance in position of the final control element resulting from the rate-action component. The advance is measured in units of time, generally minutes, and is equal to the time required for a selected motion of the final control element due to the effect of proportional-position action alone, minus the time required for the same motion due to the effect of proportional-position-plus-rate actions, with the same rate of change of the controlled variable in both cases. The rate action and proportional-position action are initiated at the same instant by the deviation of the controlled variable.

The response of a proportional-plus-reset-plus-rate action controller is illustrated in Fig. 10-43. The deviation of the controlled variable is caused by a sustained change of process load. The controlled variable returns to the set point at the completion of the corrective action of the controller because of the floating-type controller-action component employed. The final control element takes a new position corresponding to the new process-load value. The manner in which the corrective action is accomplished is a function of the controlled-variable deviation, as well as the adjustments of the proportional band, the reset rate, and the rate time.

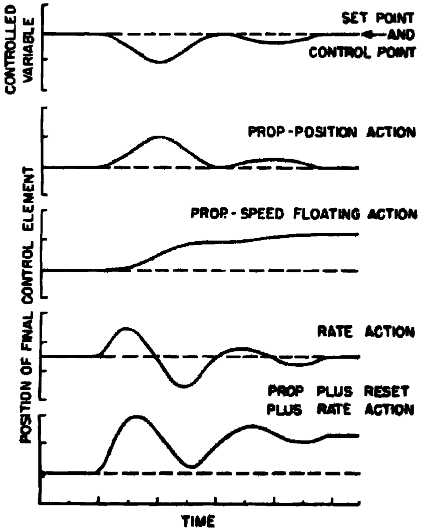

CONTROLLED VARIABLE
SET POINT AND CONTROL POINT
POSITION OF FINAL CONTROL ELEMENT
PROP-POSITION ACTION
PROP.-SPEED FLOATING ACTION
RATE ACTION
PROP PLUS RESET PLUS RATE ACTION
TIME

Fig. 10-43. Schematic diagram showing the response of a proportional-plus-reset-plus-rate action controller.

Proportional-plus-reset-plus-rate controller action is that in which there is a predetermined relation between the position of the final control element and (1) the deviation of the controlled variable, (2) the time integral of the deviation, and (3) the rate of change of the deviation.

The proportional-position reset and rate-controller action components occur simultaneously and are initiated by the controlled-variable deviation. The proportional-band value of the proportional-position-action component directly affects the magnitude of the floating and rate-controller actions. In automatic controllers having proportional-plus-reset-plus-rate action, a reset-rate adjustment and a rate-time adjustment, the proportional band may be adjusted without affecting the set value of the reset rate or the rate time.

The following groups of terms are of a general nature and define basic controller actions:

404 *Corrective action* is predetermined variation of the manipulated variable initiated by a deviation.

501 *Positioning action* is that in which there is a predetermined relation between value of the controlled variable and position of a final control element.

501*a* *Two-position action* is that in which a final control element is moved from one of two fixed positions to the other. ("Open and shut action" or "on-off" action are synonymous terms.)

502 *Integral action* is that in which there is a predetermined relation between an integral function of the controlled variable and position of a final control element.

502*a* *Floating action* is that in which there is a predetermined relation between value of the controlled variable and rate of motion of the final control element.

503 *Derivative action* is that in which there is a predetermined relation between a derivative function of the controlled variable and position of a final control element.

504 *Multiple action* is that in which two or more controller actions are combined.

504*a* *Proportional-plus-floating action* is that in which proportional-position action and floating action are combined.

504*c* *Proportional-plus-floating-plus-derivative action* is that in which proportional-position action, proportional-speed floating action, and derivative action are combined.

AUTOMATIC-CONTROL-SYSTEM APPLICATIONS

Previous parts of this section on instrumentation point out the many variations which enter into process control. Instrumentation engineering involves an evaluation of all these variations, including process characteristics and automatic-controller characteristics. Once these variations are evaluated, the engineer must select the instruments with characteristics which most appropriately meet the requirements of the process, not forgetting the cost of the installation. Some idea of the numerous ways in which instruments and controls can be applied to a process can be obtained from the examples which follow:

Figure 10-44 illustrates a simple *open-kettle temperature-control system*. The measuring means comprises a pressure-type thermometer bulb located through the side of the vessel. Internal pressure in the thermal system is transmitted to a measuring spiral within the instrument case by means of armored capillary tubing. Automatic control is accomplished pneumatically by positioning a diaphragm motor valve on the steam-supply line to the heating coils.

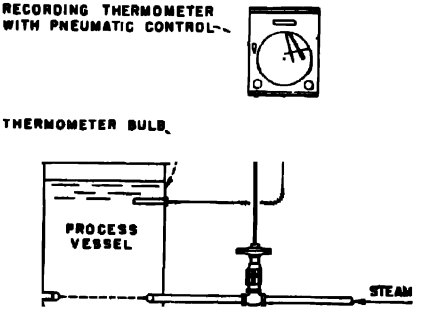


FIG. 10-44. Simple open-kettle temperature-control system.

In order to increase production, it may be desirable to heat the contents of the kettle as quickly as possible to a given temperature and then hold the kettle at that temperature. The heating time is a controllable factor which should be reduced to a minimum because it is unproductive time. This means that, during the heating time, a large heat input is required, and when the control temperature is reached, a much smaller heat input, sometimes only sufficient to supply radiation losses, is required. To meet the demands of such operations, two controlled valves, as illustrated in Fig. 10-45, may be used in parallel—a large valve and a smaller valve. If the controlled output air pressure from the instrument varies from 0 to 15 psi with 7.5 psi at the control point, the large valve will be designed to throttle from an open to a closed position when the diaphragm air pressure is varied from 0 to 7.5 psi, and the smaller valve will be designed to proportion over the entire range from 0 to 15 psi. Thus, both valves will be open in the early stages of heating. The large valve will move toward a closed position and will be closed when the set point is reached, and the smaller valve will be throttling in mid-position at the set point.

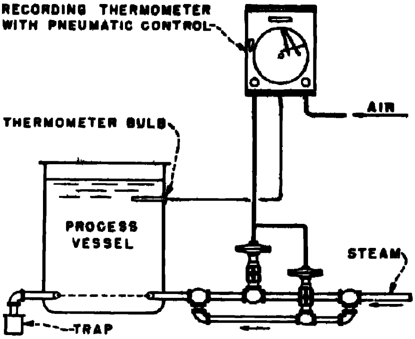


Fig. 10-45. Temperature-control system designed for rapid heating.

In some operations, it is desirable to employ a *jacketed kettle* and circulate the heating or cooling medium through the jacket. Figure 10-46 is a schematic diagram of a temperature controller applied to such an installation. In this case, a three-way valve is used to direct any part or all the flow of the heating medium through the jacket. That which does not go through the jacket is by-passed around it. Three-way valves in such applications permit a large quantity to pass through the jacket for quick heating and a small quantity for holding the temperature at the set point, similar to the two-valve application of Fig 10-15, and they do not materially restrict the total flow of heating medium being controlled.

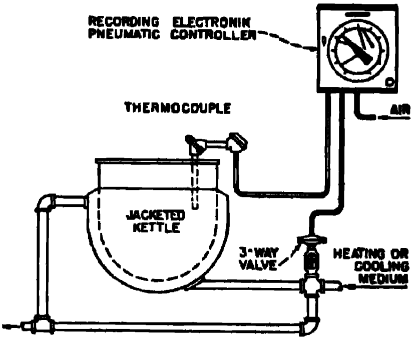

RECORDING ELECTRONIK PNEUMATIC CONTROLLER
THERMOCOUPLE
AIR
JACKETED KETTLE
3-WAY VALVE
HEATING OR COOLING MEDIUM

Fig. 10-46. System showing use of three-way diverting valve in temperature control.

Often it is desirable to bring an open kettle up to a given temperature, hold it at that temperature for a given length of time, and then shut off the heating medium and discharge the contents of the vessel. A process-control system to perform these operations is shown in Fig. 10-47. A pressure switch in the air line to the steam valve closes when the temperature reaches the set point. The pressure switch starts a timer, and at the end of the period for which the timer is set, the three-way solenoid valves in the air lines to the steam valve and to the discharge valve are operated. Both the steam valve and the discharge valve are normally closed so that they will move to a closed position upon air failure. Therefore, at the end of the timing period,

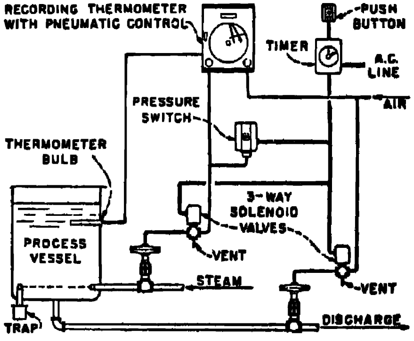


Fig. 10-47. Electric-pneumatic time temperature control system.

the three-way solenoid valve in the air line to the steam valve closes and bleeds the air from the diaphragm, and the steam valve closes; also the three-way valve in the air line to the discharge valve closes its exhaust port and admits air to the diaphragm from the supply line to open the discharge valve. The steam valve remains closed, and the discharge valve remains open. When the vessel is recharged, the operator pushes the start button which momentarily opens the electric-current supply. This automatically resets the timer to its starting position and opens the steam valve and closes the discharge valve. In the meantime, while the vessel was being emptied, the temperature dropped below the set point, and the pressure switch opened. The vessel now can be recharged to repeat the heating and holding cycle. In plants where a number of vessels are under the supervision of one operator, it may be desirable to incorporate signal lights with the automatic equipment to indicate when the vessels are being emptied.

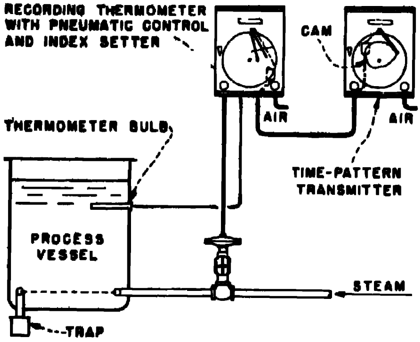


Fig. 10-48. Pneumatic time temperature control system.

In some operations, it is desirable that the *temperature follow a predetermined program*. Figure 10-48 is a schematic diagram of such an installation in which the control index of the recorder is moved by varying air pressure from the transmitter. The output air pressure from the transmitter varies in direct proportion to the radius of the cam mounted on it. Thus, the control instrument automatically controls to the time-temperature pattern of the cam. The cams are usually constructed of aluminum sheets. Graduations are printed on the aluminum disk similar to the graduations printed on a paper chart. This makes it possible to keep transmitter disks on hand and to cut new cams from them as required. The index of the control instrument can be set manually for operations when an automatic time-temperature program is not required.

Where there is energy dissipation within a vessel, as with *exothermic reactions*, it often is necessary to remove heat from the process to prevent the temperature from becoming excessive. Chromium plating is an example of a process in which energy is dissipated in the bath owing to the passage of electric current. Figure 10-49 illustrates the application of temperature control to a plating tank. The bath contains both heating (steam) and cooling (water) coils. The valves in these lines are connected to the output air line from the control instrument. One valve opens with an increase in air pressure, and the other valve closes with an increase in air pressure. The valve springs are so adjusted that one valve is open at 0 psi and closed at 8 psi, while the other valve is open at 15 psi and closed at 9 psi. Thus, when the air pressure on the diaphragms is between 8 and 9 psi, both valves will be closed. The instrument-control mechanism is adjusted to a narrow proportioning band with an output air pressure of 8.5 psi at the set point. When the temperature rises above the set point, the steam valve is closed, and the water valve opens. When the temperature drops below the set point, the water valve is closed, and the steam valve opens.

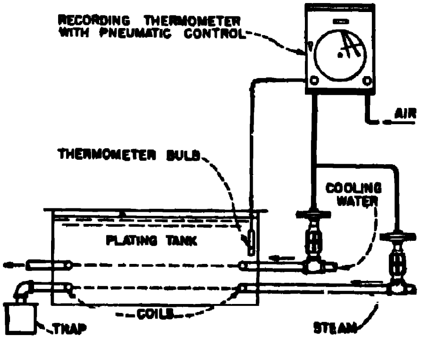


Fig. 10-49. Combined heating and cooling in one control system.

Another method of obtaining both *heating and cooling* in a control system is shown in Fig. 10-50. In this system, a tank of hot water is automatically held at the set point by a separate controller which regulates the flow of steam and water to the tank. The tank is provided with an overflow. A pump circulates hot water from the tank through a closed-coil heater in the process vessel or through a jacket surrounding the vessel. The process temperature controller operates a three-way valve to regulate the flow of hot water to the process or by-pass it back to the auxiliary tank. With a high rate of circulation, very close control can be obtained with this method, and it possesses the advantage of limiting the temperature of heating medium where high temperatures would cause decomposition. The steam- and water-valve arrangement on the heating tank is similar to Fig. 10-49.

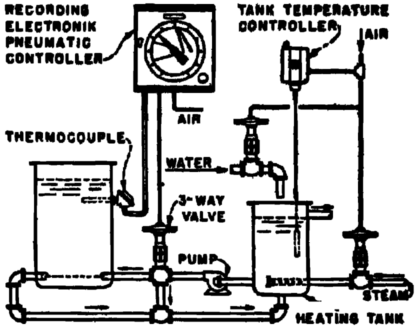

RECORDING ELECTRONIK PNEUMATIC CONTROLLER
TANK TEMPERATURE CONTROLLER
AIR
AIR
THERMOCOUPLE
WATER
3-WAY VALVE
PUMP
STEAM
HEATING TANK

FIG. 10-50. Combined heating and cooling control of process-control agent.

Figure 10-51 illustrates an open vessel in which the liquid contents are *heated by means of an electric immersion element*. The power supply to the heating element is turned on and off automatically by means of a mercury switch in the control instrument. Mercury switches such as this are designed to break at about thirty amperes at 115 volts. For greater power requirements, an auxiliary contactor is necessary.

A number of other systems have been devised to regulate the power input to electrically heated processes, including the following:

1. Automatic adjustment of resistances in series with the heating elements
2. Use of power input controllers which are a type of circuit interrupter in which the length of time that the power is supplied to the process is automatically varied in reference to departure from the set point
3. Automatically varying the voltage applied to the heating elements by means of multitap transformers or induction voltage regulators
4. Automatically varying the power supplied to the heating elements by means of a saturable reactor

Adjustable resistors, input controllers, multitap transformers, and induction regulators generally are operated by reversible motors from a single-pole double-throw mercury switch operated by the automatic-control instrument. When the temperature is at the set point, the position of the switch is such that the circuits to the reversing motor are open so that the power input to the process remains at that value.

The saturable-reactor type of electric control system receives impulse from the change in position of the sliding contact on a resistor.

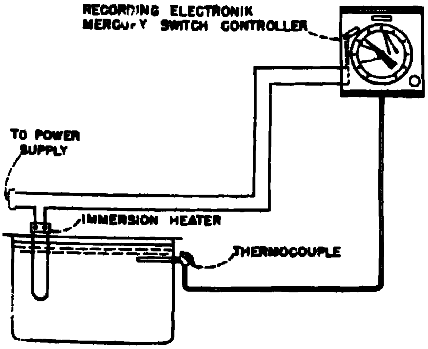


FIG. 10-51. On-off temperature-control system where electric immersion heater is used.

In Fig. 10-52, a thermocouple is located in the discharge of an electrically heated continuous-fluid heater. The output air pressure of the pneumatic control instrument is applied to a small air-operated slide-wire. The purpose of using the pneumatic control instrument to operate the slider is to take advantage of the adjustable proportional band and reset rate incorporated in its control system. These control characteristics are necessary for good control on continuous processes in which the throughput varies.

The potential unbalance caused by the change in position of the sliding contact is amplified electronically to regulate the amount of direct current in the saturating winding of a saturable-core reactor. The power supply is connected to the primary of the saturable reactor, and the heating element is connected to the secondary. As the direct current through the saturating winding varies, more or less power is transmitted from the primary to the secondary to which the heating element is connected.

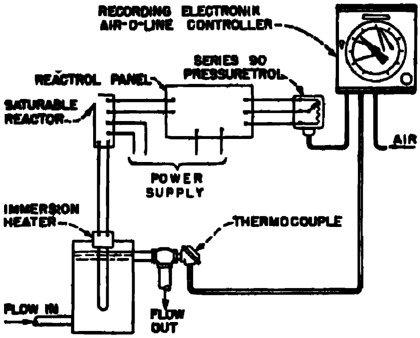


FIG. 10-52. Control system for saturable-reactor-type electric heater.

In most heating operations in *closed vessels*, such as autoclaves, the upper space in the vessel is filled with steam. In such cases, a pressure controller may be used for regulating the flow of the heating medium. The steam pressure is an indirect measure of the temperature. Thus, pressure controllers may be used in place of temperature controllers and have the advantages of lower cost, simplicity of operation and installation, and often better control because of less lag. However, when pressure control is used, it is necessary to vent entrapped air or other gases released during the heating-up period. Pressure control in lieu of temperature control may not be satisfactory for processes requiring the introduction of inert or other gases.

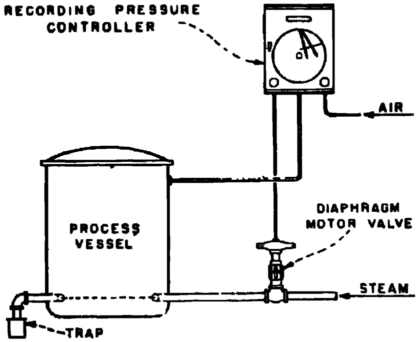


FIG. 10-53. Typical pressure-control system for a closed vessel.

Figure 10-53 shows a pressure controller applied to a closed vessel. If direct temperature control is required, owing to the presence of inert gases or other reasons, the system illustrated in Fig. 10-54 can be used. However, if either temperature or pressure is controlled, the uncontrolled temperature or pressure can be recorded on the same chart where desired.

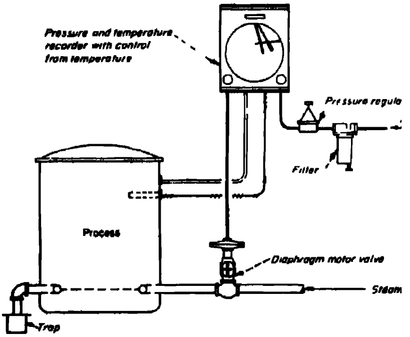

Pressure and temperature recorder with control from temperature
Pressure regula
Filter
Process
Diaphragm motor valve
Steam
Trap

FIG. 10-54. Temperature control of a closed vessel with pressure recorded.

Suppose that the upper space in a closed vessel is filled with steam and that a pressure recorder connected to it reads 100 psi. If a thermometer bulb partly filled with water is inserted in the same space and is connected to read on the same chart, it will also read 100 psi. This combination has been used to detect the presence of gases in closed vessels, such as paper digesters, by noting whether the pressure pen records the same value as the temperature pen. When other vapors are present, as in distillation columns, the thermometer bulb can be partly filled with the condensed vapors, and the difference between the temperature and pressure pens then will be a measure of the composition of the vapor in the column compared to the vapor in the thermometer bulb.

The pressure in closed vessels can be reduced *below atmospheric pressure* by means of a steam-jet ejector. As a rule, steam-jet ejectors will not operate satisfactorily over a wide range of steam flow. By setting the steam flow to the ejector at its optimum value, good control can be obtained by regulating a valve opening to atmosphere. Such an installation is illustrated in Fig. 10-55. In l-p operations, it is often necessary to use absolute-pressure controllers to obtain consistent results. Absolute-pressure controllers compensate for variations in atmospheric pressure. When centrifugal pumps are used to reduce the pressure, control may be obtained by regulating a valve located in the suction line of the pump, as in Fig. 10-56.

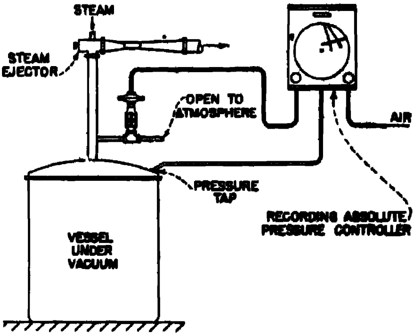


Fig. 10-55. Vacuum control of a steam-ejector system.

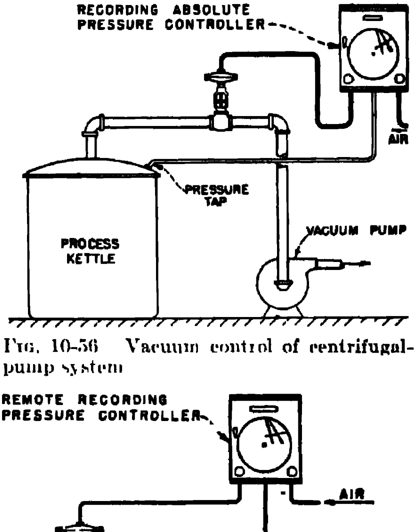


Fig. 10-56. Vacuum control of centrifugal-pump system.

Pneumatic transmission of flow, temperature, pressure, and liquid-level measurements from process locations to a central control board was originally developed to avoid the use of electricity for this purpose in plant areas where explosive atmospheres are possible. A further advantage is the complete isolation from the control board of all process piping containing high pressures or hazardous fluids. A pressure-control system using pneumatic transmission is shown in Fig. 10-57. The indicating-pressure transmitter is located near the h-p line on the processing unit and transmits the pressure reading to the recording control instrument which operates the control valve in the pipe line. To reduce lag, a transmitter equipped with control often is used.

Automatic control of flow and liquid level represents the largest field of application for *differential-pressure controllers* in the process industries. Figure 10-58 is a schematic diagram of a typical flow-control installation in which the controlled valve is located in the flow line A differential mercury manometer connected across the orifice transmits the movement of its mercury electrically to a recording control instrument which, in turn, operates the valve in the flow line to maintain a constant differential pressure across the orifice and consequently a uniform rate of flow in the pipe.

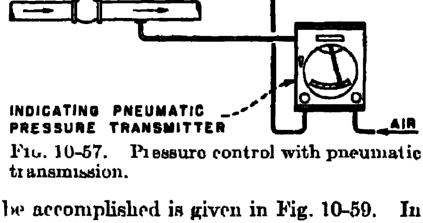

REMOTE RECORDING PRESSURE CONTROLLER
AIR
HIGH PRESSURE LINE
INDICATING PNEUMATIC PRESSURE TRANSMITTER
AIR

Fig. 10-57. Pressure control with pneumatic transmission.

Often it may be desirable automatically to control one *rate of flow in direct proportion to another rate of flow*. A schematic diagram showing how this can be accomplished is given in Fig. 10-59. In the upper diagram, a meter body or manometer connected across the orifice in the primary flow line operates a pneumatic mechanism which develops an air pressure in direct proportion to the differential pressure across the orifice. This air pressure is transmitted to the control instrument connected across the orifice in the secondary flow line where, by means of a bellows-operated mechanism, it moves the control index. Thus, the set point of the controller regulating the secondary flow is moved in proportion to the primary flow. The mechanism in the secondary-flow controller is adjustable so that the ratio of the two flows can be varied. Maximum and minimum limits also can be set for the secondary-flow rate.

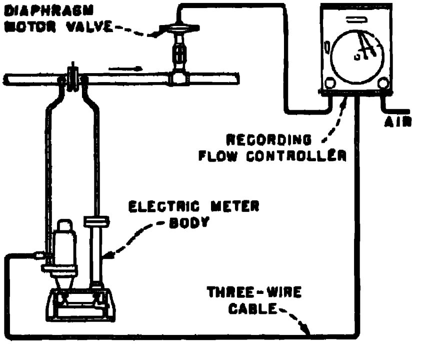


Fig. 10-58. Typical flow-control system with electrical transmission.

The lower diagram (Fig. 10-59) shows how pneumatic transmission can be used in a *ratio flow-control system* to permit locating the recording controller remotely with respect to the primary- and secondary-flow lines. It is possible by using equipment similar to that shown in Fig. 10-59 to reset the control index of a flow controller from variations in liquid level or to reset the control index of a pressure controller from variations in temperature or other combinations that may be desirable.

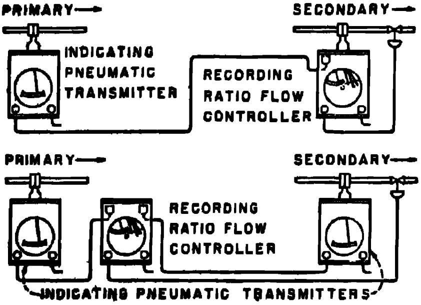

PRIMARY
SECONDARY
INDICATING PNEUMATIC TRANSMITTER
RECORDING RATIO FLOW CONTROLLER
PRIMARY
SECONDARY
RECORDING RATIO FLOW CONTROLLER
INDICATING PNEUMATIC TRANSMITTERS

Fig. 10-59. Two common arrangements for ratio flow control.

Differential-pressure recorders also are used in connection with continuous filters, screens, and heat exchangers to reveal undesirable operating conditions such as the accumulation of scale or foreign material which affect the efficiency of the equipment.

Figure 10-60 illustrates the application of a differential-pressure control to maintain automatically the *level in a closed vessel under pressure*. Instruments of this type can be adjusted to hold the level within close limits, but this is not always desirable. For example, the level in an accumulator tank is not important until the tank is quite full or almost empty. For such applications, the controller can be adjusted to permit the level to change between wide limits, and where these limits are reached, the discharge valve will move to make large changes in the rate of discharge to prevent the tank from filling or becoming empty.

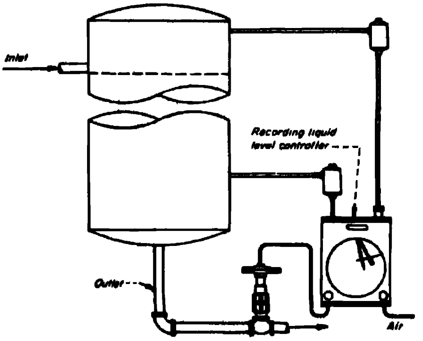


Fig. 10-60. Differential-pressure-type liquid-level control system.

There are occasions when it is desirable to *record or control the difference in temperature between two points* in a process or to control one temperature at a fixed difference with respect to another. This can be accomplished by using two thermocouples connected differentially to a potentiometer. The temperature difference across heat exchangers, compressors, centrifugal pumps, or between the heating jacket and column on adiabatic laboratory stills are but a few applications for this arrangement. A diagram of such an application is shown in Fig. 10-61.

Heat exchangers are used to heat or cool, with heat conservation as a possible dividend. The temperature of the fluid being heated or cooled can be controlled automatically by regulating the rate of flow of the fluid through the exchanger by means of a temperature controller responsive to the outlet temperature. If it is not feasible to put a control valve in this line, it may be placed in the line of the heating or cooling fluid, but the control may not be so close with the valve in this location because of the increased time required for the effect of a change in flow rate to be felt by the control instrument.

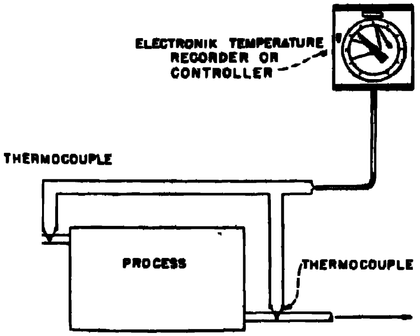


Fig. 10-61. Two thermocouples connected to measure the difference in temperature between two points.

Figure 10-62 shows a heat exchanger in which hot oil is being cooled to a constant lower temperature by water. For the purpose of water conservation and control stabilization, the outlet-water temperature is controlled by an inexpensive nonindicating controller which regulates the flow of water through the exchanger. The oil-temperature controller operates a three-way valve which divides the oil flow into two parts; one part goes through the heat exchanger, and the other part by-passes the heat exchanger. The part that goes through the exchanger is cooled below the desired final temperature. Upon leaving the exchanger, the cooled oil is mixed with the hot oil by-passed around the exchanger. Thus, the temperature controller blends hot oil and cold oil to obtain oil at the desired temperature. In this system, a change in valve position is felt very quickly by the temperature controller, and only a controller that will respond quickly should be used.

In some cases, a liquid is heated by steam in a heat exchanger. If the exchanger has excess capacity, a pressure controller can be used to regulate the flow of steam, and the final temperature of the liquid also can be recorded on the same chart.

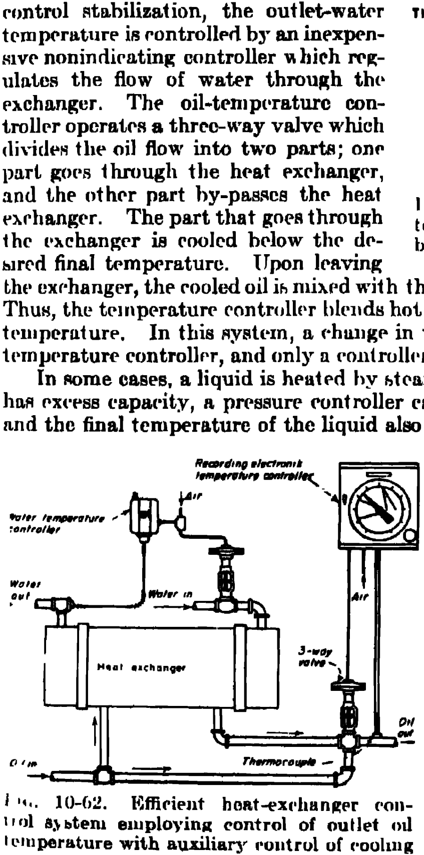


Fig. 10-62. Efficient heat-exchanger control system employing control of outlet oil temperature with auxiliary control of cooling water.

Dowtherm vapor is used as a high-temperature heating medium in many of the process industries. Figure 10-63 is a diagram of an automatic-control system applied to an oil-fired Dowtherm vaporizer. A pressure controller responsive to the outlet pressure of the vaporizer regulates the flow of oil and air to the burner. A photoelectric flame detector and pressure switches in the air and oil lines are safety devices to open the circuit of the solenoid valve to shut off the oil supply and sound an alarm when the air pressure is too low, when the oil pressure is too low, or if the flame goes out. On electrically heated Dowtherm vaporizers, the pressure controller can be used to operate a switch to open and close the circuit to the heating elements or to operate the slider on a potential divider, as shown previously for the control of electric heating elements by means of saturable reactors.

Kilns used for drying may be automatically controlled in response to the temperature of the exit material. The material dried in kilns is usually in the form of small particles so that a stationary thermocouple can be placed in the stream of material leaving the kiln. Care should be taken so that the thermocouple will respond to changes in temperature of the material and not be affected by air currents.

Figure 10-64 is a *kiln-control system* in which the oil burner is regulated from an instrument connected to a radiation pyrometer sighted on the material in the kiln or leaving the kiln.

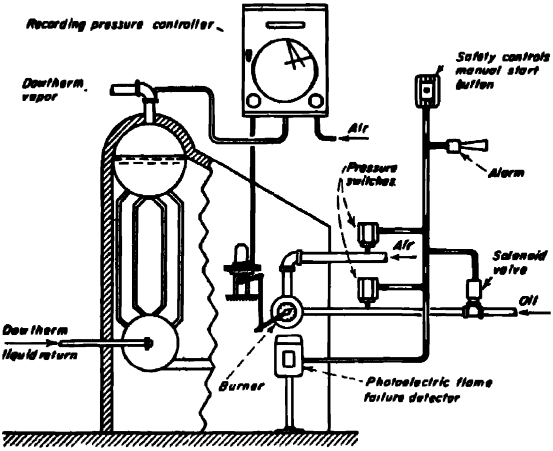

Recording pressure controller
Dowtherm vapor
Safety controls manual start button
Air
Pressure switches
Alarm
Air
Solenoid valve
Oil
Dowtherm liquid return
Burner
Photoelectric flame failure detector

FIG. 10-63. Typical control system for oil-fired Dowtherm vaporizer.

Dust is a problem around most kilns, and the air-cooled fitting not only cools the radiation heat, but it prevents dust from depositing on the lens. The location of the radiation pyrometer should be selected so that the line of sight does not include any part of the flame.

When the *rate of drying* is to be controlled, wet- and dry-bulb measurements of the air prior to its contact with the material are desirable. If the wet and dry bulbs are located in the air stream leaving the materials, a measure of the moisture from the material is obtained. The rate of drying may be accelerated by introduction of fresh air and retarded by introduction of water vapor. It should be borne in mind that, when applying control equipment to driers, moisture removed is carried away in the exit air, and the distribution of air in the drier is important from the standpoint of uniform drying. The air velocity across the wet bulb should be at least 10 ft per sec to ensure sufficient evaporation.

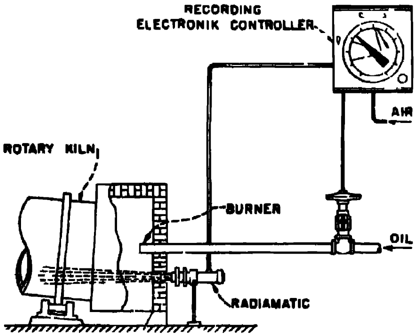


FIG. 10-64. Temperature-control system for oil-fired rotary kiln.

Direct measurement of relative humidity by means of hair elements is restricted to temperatures below 40°C, and they should be used only in clean atmospheres because grease and dust close the pores of the hair, and it will not respond rapidly to moisture changes.

Thermocouples, particularly pencil-type couples, can be used satisfactorily for wet-bulb measurements. The wet bulb should be sufficiently long to preclude erroneous readings due to heat conduction to the hot junction of the thermocouple. When relative humidities above 90 per cent are involved, careful consideration should be given to the sensitivity and accuracy of the instruments because the wet- and dry-bulb temperatures approach the same value at 100 per cent.

Figure 10-65 shows a wet- and dry-bulb-control installation on a *recirculating drier.* The sensitive elements are located in the air leaving the drier. The dry bulb controls dampers either to direct air through the heater or to by-pass it around the heater. The wet bulb controls admission of fresh and release of moisture-laden air by means of a three-way damper.

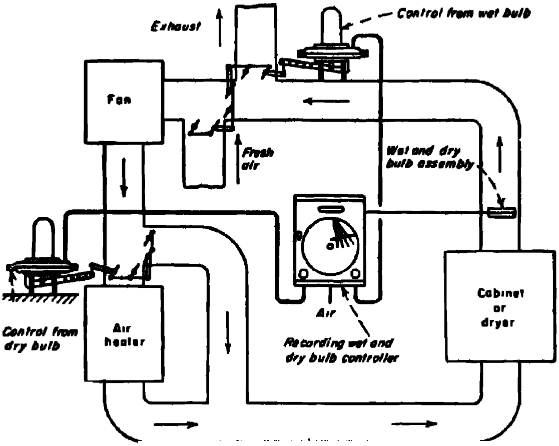


Fig. 10-65. Humidity-control system for recirculating drier.

At times it is desirable to have a valve close and stay closed after a process has reached a predetermined condition (*automatic-shutdown systems*). In Fig. 10-66, a pneumatic controller is shown with its control index set at a predetermined value. When the value is reached, the output air pressure of the controller drops below 15 psi, and the pressure switch opens the circuit to a three-way valve which shuts off the air to the valve and permits the air on the diaphragm to discharge to atmosphere. The diaphragm valve closes and stays closed until the three-way valve is opened again by means of the manual-reset push button.

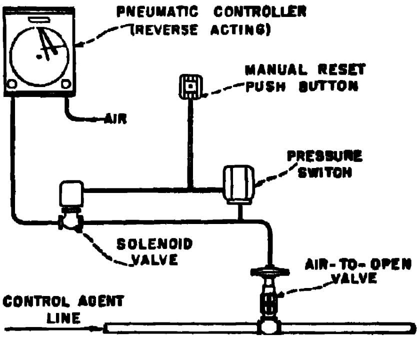


Fig. 10-66. Automatic safety shutdown system with combination electric and pneumatic controls.

Figure 10-67 shows an automatic-shutdown system which is completely pneumatic for use in locations where the use of electricity is considered hazardous. In this system, when the pen reaches the set point, the output air pressure from the instrument decreases. The controlled valve closes, and a small diaphragm valve in the air line from the instrument also closes. With the diaphragm connection of this valve on the downstream side of the valve, both valves will remain closed even though the pen on the instrument returns to a position below the set point. However, the valves can be reopened by manual operation of the reset valve which should be of the self-closing type. Similar systems using combinations of electrical control instruments and electrically operated valves have been used in nonhazardous locations.

Distillation equipment varies from small kettle stills to large continuous stills, and the instrumentation applied varies accordingly. Kettle-type stills require little control equipment. They are usually charged with a given quantity of material which then is heated until the volatile components are removed. Then the still is emptied. In such operations, the vapor temperature at the top of the still is the main guide in its operation. The vapor temperature may be simply recorded while the still operator manually controls the heating medium in accordance with his observations of the temperature record, or the temperature recorder may incorporate a control system automatically to control the heating medium or shut off the heating medium and open the discharge valve when a predetermined temperature is reached In some continuous-still operations, the feed is sufficiently uniform that efficient operation can be obtained with a flow-rate controller on the feed and a flow-rate controller on the steam supply, a temperature recorder on the vapor line, and a liquid-level controller regulating a valve in the discharge from the bottom.

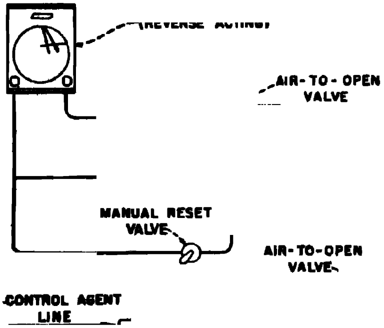


FIG. 10-67. Pneumatic safety shutdown system.

Figure 10-68 outlines the *instrumentation on a large fractionating column with a separate reboiler.* A flow-rate controller admits the feed to the column at a uniform rate. The feed could also be brought through a controlled heat exchanger so that it would enter the column at a constant temperature. A pressure controller regulates the release of vapor from the column to maintain a uniform back pressure. A temperature controller resets the control index of a flow controller on the steam line to the reboiler to maintain a uniform temperature of the vapor leaving the reboiler This method of controlling the steam eliminates variations due to fluctuations in steam pressure. A flow-rate controller on the reflux line regulates the return of condensate to the column. A level controller or the reflux accumulator governs t valve in the product line so that the demands for reflux will be met first. A mu tipoint instrument records the temperature of the vapor, the feed, the condensate, and the temperatures at several locations on the column. Variations of the above instrumentation of a fractionating column can be made to meet particular require

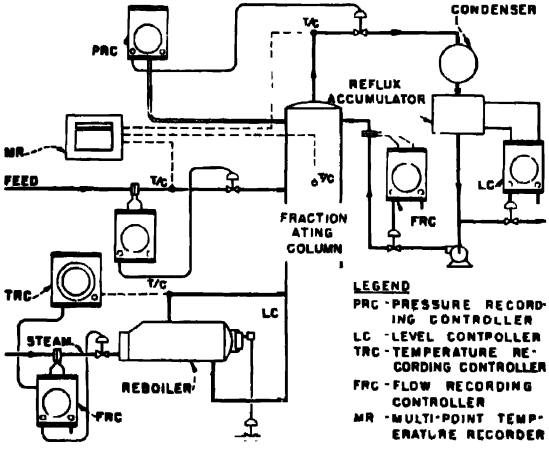

CONDENSER
T/C
PRC
REFLUX ACCUMULATOR
MR
FEED
T/C
T/C
FRACTIONATING COLUMN
FRC
LC
TRC
T/C
LC
STEAM
REBOILER
FRC
LEGEND
PRC - PRESSURE RECORDING CONTROLLER
LC - LEVEL CONTROLLER
TRC - TEMPERATURE RECORDING CONTROLLER
FRC - FLOW RECORDING CONTROLLER
MR - MULTI-POINT TEMPERATURE RECORDER

FIG. 10-68. Typical control system for continuous fractionating column.

ments. For example, the back pressure on a column can be controlled by regulating the flow of cooling water to the condenser. Also the steam pressure to the bottom of a column has been regulated from the pressure differential across the column to maintain a uniform rate of vapor up the column, and ratio controllers may be used to return a fixed percentage of the condensate to the column.

INSTRUMENTATION COSTS

Instrumentation of plant operations, especially where continuous processing is involved, has increased manyfold during the past decade or two. This increased usage of instruments and automatic controls is a result of several factors all intimately associated, directly or indirectly, with processing economies.

Instruments are rapidly becoming an integral consideration in the design of all processing plants, and the conception that indicating, recording, and controlling instruments are accessories "to be added later" is seldom encountered where new construction is involved. Consequently, instrumentation costs now represent a significant portion of the over-all investment in plant equipment, whereas in a past era, the expenditure for instrumentation was oftentimes placed in the realm of incidental costs.

It must be stressed that, while most plant designers and contractors recognize the factor of instrumentation at the outset of a project, they too often relegate the proposed expenditure for instrumentation to the tail end of their project budget thinking. Consequently, instrumentation costs often are underestimated, with resultant penalties to smooth and efficient process operation. Although only 5 per cent of the investment in a plant may be required for instrumentation, it is an extremely important 5 per cent and, if trimmed by only ½ per cent, the "apparent" savings will ultimately be reflected in one of several ways, including higher over-all costs and lower product quality.

A study of instrumentation costs not only is complicated by the great variety of plant operations and processes used, the basic instrumentation requirements of which vary widely, but also by the many methods of measurement and styles of instruments available to do a given job. A process temperature can be measured in one fashion, *e.g.*, by simple, low-cost glass-stem thermometers; on the other hand, where accurate, continuous, and automatic recording and controlling of temperatures are required, the instrument to do the job may logically be a self-balancing potentiometer incorporating carefully machined parts and electrical and electronic components, all mounted in a rugged case of attractive appearance for convenient mounting on a control panel. Obviously, instruments of the latter type, performing many more functions and assuming greater responsibilities, cost more. Over-all instrumentation costs have risen not only because more instruments are being used but also because the instruments themselves have become sophisticated and complex.

Tastes of the design engineer also greatly affect over-all instrumentation costs. There is a trend in the design of control rooms for large processing units toward more and more attractive appearance of instrument-panel boards and greater comfort and pleasing surroundings in the control room. Air conditioning, indirect lighting, soundproofing, and color dynamics in the control room are looked upon by some designers as luxury items of purely aesthetic value; other designers regard them as important contributing factors to long-term operating efficiency of the unit. Some designers are content to install pressure gauges on each pump or blower; other designers specify that these instruments all be incorporated on a central panel board in the pump room. The figures given in the table with this article are based upon average installations of the day. Where centralized control is not considered essential, possibly these figures will be a trifle high; where it is desired to take advantages of centralized control and an attractive and comfortable control room, the figures will be considerably lower. Thus, the factor of taste must be considered when making a cost estimate.

For use in making preconstruction instrumentation-cost estimates at a stage before the complete instrument and automatic-control layout is known, the percentage of total equipment cost required for instrumentation, as experienced by other similar processes, can be helpful. A very limited but recent survey of the chemical process industries divulges the figures presented in Table 10-2.

Instrumentation costs for the production of methanol intermediates, acetic acid recovery, polythene synthesis, manufacture of acrylic resin molding powder, and agricultural insecticide and fungicides are reported by one user to vary from 8 to 12 per cent of the investment in total installed equipment. Another user reports that the average instrumentation cost for 10 units making agricultural chemicals and plastics is 9.7 per cent. A progressive manufacturer of perfumes and cosmetics reports that instrument costs usually run between 5 and 10 per cent, but that on some processes, instrument costs run as high as 19 per cent. A textile manufacturer reports that the average mechanical operation in a textile mill, such as weaving and spinning, requires an investment in instrumentation of well under 1 per cent of the total plant-equipment investment. Large soap manufacturers report that instrumentation costs run between 2.2 and 3.2 per cent of the total investment in equipment. On a more complicated plant operated by a soap manufacturer, it is reported that the instrumentation costs are as high as 8.6 per cent. A processor of nonferrous ores and manufacturer of nonferrous salts reports that instrumentation costs of an electrolytic refining unit are approximately 1.02 per cent. A lime plant owned by this same processor has an investment of only 0.39 per cent in instrumentation.

Table 10-2. Installed Costs of Instruments as Percentages of Total Plant- and Process-equipment Costs

Process type	*Per cent of equipment cost*
Ethylene plants	4
Solvent-dewaxing units	3
Thermofor catalytic cracking units	3.4
Viscose making	2.9
Synthetic ammonia plants	3.2
Contact sulfuric acid plants	3
Nitric acid plants	2
Hydrochloric acid plants	1.9
Superphosphate plants	Negligible
Hydrofluoric acid alkylation units	5.7
Gas-recovery units	4.9
Styrene plants	4
Pharmaceutical plants	15
Soap plants	2 3 2
Electrolytic nonferrous metal refining	0.91
Lime plants	0.45
Manufacture of fluoride salts	1.1
Pulp mills	3
Textile mills	1.5

These wide variations in instrumentation-cost percentages are a result of many factors as discussed in the following paragraphs.

Numerous factors determine the percentage of the total investment in a process which must be allotted to instrumentation. Some of these factors are: (1) nature of unit operations making up process; (2) special items required because of corrosion, low flows, etc.; and (3) capacity vs. number of processing units. The fact that some industries are more instrument-minded and conscious of the many advantages which continuous measurements and automatic controls bring about also has an important bearing on comparative costs. Petroleum refiners and manufacturers of organic chemicals are outstanding in their appreciation of instrumentation, and hence they invest more capital in this direction. An important observation made over the past few years indicates that improvements and technical progress in processing go hand in hand with increased instrumentation.

Processes involving distillation are well known as heavy users of instruments. One fractionating tower, for example, may require a potentiometer to control column temperature by resetting the control index of a flow controller on the steam-supply line to the reboiler, a flow controller on the tower feed, a liquid-level controller or the tower bottoms, a flow controller on the reflux line, a liquid-level controller on the reflux accumulator, and integrating flowmeters on the product lines.

On the other hand, mixing, crushing, grinding, and filtering are operations which usually require only the simplest type of instrumentation. A superphosphate plant which involves principally crushing, grinding, leaching, mixing, and packing involves practically no instrumentation. Reflection on the figures given in Table 10-2 will bear out these conclusions.

Some operations require that more variables be controlled and more facts be known than do others. Logically, where temperature, pressure, pH, or other variables must be controlled within very narrow limits, the best in available instrumentation will be required. Where costs of steam and other mediums are extremely pertinent, numerous recording and indicating meters may be required.

It is not unusual for 50 per cent of the total investment in a pilot plant to be devoted to instruments and controls. Here temperatures and pressures and other variables are measured and recorded at a host of locations which normally would not be of interest in a large operating unit once it is developed.

Where corrosive materials are handled, equipment costs in general rise. Instruments and controls are no exception. The figure of 5.7 per cent for hydrofluoric acid alkylation given in Table 10-2, as compared with the other typical petroleum processes listed, is due in large part to special valves, protection for sensing elements, etc. Where small flows which cannot be handled in the smallest conventional control valves are encountered, more expensive valves of special construction must be used, upping the total instrumentation costs.

Although the capacity of one unit making a given substance may be three times the capacity of another unit making the same substance, the number of critical process variables which must be measured or controlled remains about the same. For example, about 4 per cent of the investment in a 20-ton-capacity contact sulfuric acid plant is for instrumentation. A 200-ton-capacity plant requires only 2 per cent investment in instrumentation. In increasing the capacity of a unit, the major increase in instrumentation costs is brought about by the greater cost for larger control valves, longer lengths of extension wire, connecting tubing, etc., and is not markedly affected by requirements for a larger number of individual instruments.

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SECTION 11

INDUSTRIAL PIPING

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PIPE, VALVES, AND FITTINGS FOR NORMAL AND HIGH PRESSURES

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Pipe, valves, and fittings are used in every industry to convey fluids from one point to another. These fluids may be in a gaseous state, a liquid form, semisolid, abrasive, and/or corrosive; they may be flowing under low-pressure conditions, high pressure, and at low or high velocities. Each of these and other manifold variables make the proper selection of the most suitable piping material, the valve which will perform most satisfactorily, or the fitting which will produce the utmost in economy and service a problem which can be dealt with only by one having sufficient information regarding the performance of pipe, valves, and fittings under the service conditions which are likely to be encountered.

In the design and selection of piping materials for service conditions ranging from the average home plumbing installation to the h-p power plant, careful study must be given to the individual requirements to provide a basis for the selection of the proper equipment.

The primary purpose of this subsection is to present information which can be used as a guide by the designer in selecting the product most suitable for the many service conditions encountered in the various industries. While it is true that there are many codes, standards, and manufacturers' catalogues which can be used to some degree for this purpose, the basic reasoning behind the adoption of certain types of products may not be readily available. It must be understood, however, that no product should be used which is definitely legislated against by these codes or standards and that manufacturers' recommendations should be followed.

PIPE

The most prevalent type of product used in all piping systems is "pipe." The selection of the size, material, and class of pipe for a particular service condition is dependent upon several factors among which are pressure, temperature, corrosion, availability, etc.

PIPE MATERIALS

The pipe material is generally selected on the basis of the requirements set up in various codes and standards which refer in the majority of cases to specifications established by the American Society of Testing Materials. Other widely used specifications are those published by the Navy Department, the American Petroleum Institute, and the United States Government. A partial list of piping materials and the corresponding specification reference is given in Table 11-1.

Some of the more important piping materials and their specific characteristics are as follows:

Carbon-steel Pipe. Carbon-steel pipe derives its distinctive properties chiefly from the amount of carbon which it contains. The basic material used in the production of carbon-steel pipe is generally produced either by the open-hearth, electric-furnace, or the acid-Bessemer process. In ordering the pipe for fabrication, open-hearth-process pipe is generally specified because of its superior forming qualities.

Butt-welded carbon-steel pipe is available in sizes from $\frac{1}{8}$ to 3 in. inclusive, lap-welded pipe is available in $1\frac{1}{4}$-in. sizes and larger. Both types are available in standard and extra-strong thicknesses. Carbon-steel pipe is manufactured to conform with numerous ASTM specifications, three of which are commonly used for fabricating.

Table 11-1*

Materials, pipe, and tubing	*Specification*
Brass and copper	
Copper pipe, standard sizes	**ASTM B42**
Red brass pipe, standard sizes	**ASTM B43**
Seamless copper tubing, bright annealed	**ASTM B68**
Seamless copper tubes	**ASTM B75**
Copper water tube	**ASTM B88**
Copper and copper-alloy seamless condenser tubes and ferrule stock	**ASTM B111**
Cast iron	
Cast-iron pit-cast pipe for water or other liquids	**ASA A21.2 and** **ASTM A44**
Pipe, water, cast iron (bell and spigot)	**FSB WW-P-421**
Mechanical joint cast iron pipe	**CIPRA**
Wrought iron	
Welded wrought iron pipe	**ASTM A72**
Spiral welded steel or iron pipe	**ASTM A211**
Casing, drill pipe, and tubing	**API 5-A**
Line pipe	**API-5-L**
Steel	
Welded and seamless steel pipe	**ASTM A53**
Lap welded and seamless steel and lap welded iron boiler tubes	**ASTM A83**
Forged or rolled steel pipe flanges, forged fittings, and valves and parts for high temperature service (material for forged and bored carbon steel pipe)	**ASTM A105**
Seamless carbon steel pipe for high temperature service	**ASTM A106**
Black and hot dipped zinc coated (galvanized) welded and seamless steel pipe for ordinary uses	**ASTM A120**
Electric fusion welded steel pipe (sizes 30 in. and over)	**ASTM A134**
Electric resistance welded steel pipe	**ASTM A135**
Electric fusion welded steel pipe (sizes 8 in. to but not including 30 in.)	**ASTM A139**
Electric fusion welded steel pipe for high temperature and h.p. service	**ASTM A155**
Seamless alloy steel pipe for high temperature service	**ASTM A158**
Electric resistance welded steel and open hearth iron boiler tubes	**ASTM A178**
Seamless cold drawn low carbon steel heat-exchanger and condenser tubes	**ASTM A179**
Forged or rolled alloy steel pipe flanges, forged fittings, and valves and parts for high temperature service (material for forged and bored alloy steel pipe)	**ASTM A182**
Seamless steel boiler tubes for h.p. service	**ASTM A192**
Seamless carbon molybdenum alloy steel pipe for high temperature service	**ASTM A206**
Medium carbon seamless steel boiler and superheater tubes	**ASTM A210**
Spiral welded steel or iron pipe	**ASTM A211**
Seamless alloy steel boiler and superheater tubes	**ASTM A213**
Electric resistance welded steel heat exchanger and condenser tubes	**ASTM A214**
Electric resistance welded steel boiler and superheater tubes for h.p. service	**ASTM A226**
Atomic hydrogen arc welded and electric resistance welded alloy steel boiler and superheater tubes	**ASTM A249**
Copper brazed steel tubing	**ASTM A254**
Casing, drill pipe, and tubing	**API 5 A**
Line pipe	**API 5-L**
Seamless and welded ferritic stainless steel tubing for general service	**ASTM A268**
Seamless and welded austenitic stainless steel tubing for general service	**ASTM A269**
Seamless austenitic chromium nickel steel still tubes for refinery service	**ASTM A271**
Seamless chrome molybdenum alloy steel pipe for service at high temperatures	**ASTM A280**

* Published in ASA Code for Pressure Piping, Supplement No. 2, April 1947.

ASTM A120 Specification for Black and Hot-dipped Zinc-coated (*Galvanized*) *Welded and Seamless Steel Pipe for Ordinary Uses.* ASTM A120 pipe might be called the "garden" variety of pipe since it is used extensively for general piping purposes. These ordinary uses include those for low-pressure domestic service, small-size low-pressure water installations, and gas lines. A120 is not suitable, however, for close bending or for high-temperature service.

The steel used for this pipe is made by the open-hearth, electric-furnace, or the acid-Bessemer process and is of soft weldable quality. If galvanized pipe is desired, the coating inside and outside is applied by the hot-dip process, and the weight of the coating is listed in ounces per square foot. Specimens of each lot of pipe are selected and subjected to tests that measure their coating weight, pipe weight, hydrostatic strength, weld strength, and diameter and wall thickness. A120 pipe lengths are usually cut in the 16- to 22-ft range for standard-weight pipe. Specifications are also listed for workmanship and finish required for this grade of pipe which is stamped or stenciled to show the manufacturer, the length, and the ASTM Specification A120.

ASTM A53 Specification for Welded and Seamless Steel Pipe. ASTM A53 pipe is manufactured as black or hot-dipped galvanized pipe and might be called a medium-grade pipe. Pipe complying with these specifications may be used for coiling, bending, flanging, or other specified special purposes. The steel for A53 is made by the same process as that used for A120 with the exception that electric-resistance-welded pipe for the smaller sizes of ⅛ and ¼ in. in diameter is made from open-hearth steel. The chemical content for this pipe steel is specified, and samples from each lot of 500 lengths are checked against the specifications. The tensile strength as well as the flattening and bending properties of the pipe are checked. Following the hydrostatic test, the pipe is checked for weight, workmanship, finish, marking, and final inspection.

The Code for Pressure Piping limits the use of A53 to a maximum steam pressure of 400 lb and a maximum temperature of 750°F. Other codes differ in their specified use for A53.

ASTM A106 Specification for Seamless Carbon-steel Pipe for High-temperature Service. The steel used for A106 pipe, which is specified as grade A or grade B, is killed steel made by the open-hearth or the electric-furnace process. The chemical composition and the physical properties are carefully specified and presented in table form. Test specimens are used to determine the bending and flattening properties of the pipe, and unless otherwise specified, each length of pipe is hydrostatically pressure-tested. After checking the weight, wall thickness, diameter, finish, and over-all workmanship, the pipe is marked and inspected. A106 pipe is used for high-pressure and high-temperature work and should be suitable for bending, flanging, lapping, and similar forming operations. The A106 specifications are rigid, and the manufacturing tolerances which are specified are closely followed by the producer.

Alloy-steel Pipe. The term "alloy-steel pipe" is meant to apply to any tubular steel product which differs from carbon steel by the addition of one or more alloying metals to which it owes its distinctive properties. The basic materials for the manufacture of alloy pipe are made by the open-hearth or electric-furnace process. Most of the alloy-steel pipe is made by the seamless process, although some high-alloy varieties are produced as welded pipe. In the sizes up to 1½ in. the pipe is cold-drawn; in sizes from 1½ to 10 in. it is either hot-finished or cold-drawn; and in sizes above 10 in. it is hot-finished. Alloy-steel pipe is normally classified by schedule number or wall thickness.

In recent years steam temperatures have exceeded the limits for which carbon-steel pipe is suitable. This has naturally resulted in the more extensive use of alloy-steel piping. In power-plant work involving temperatures up to 1000°F, carbon-molybdenum pipe has found wide application. Basically this material is a low-carbon (0.10 to 0.20 per cent) silicon-killed steel to which molybdenum (0.45 to 0.65) per cent is added. This small quantity of molybdenum gives the material superior creep resistance at temperatures up to 1000°F. At this writing certain specifications call for the addition of 0.5 to 1.0 per cent chromium to resist graphitization of carbon-molybdenum piping at welded joints which results from service under high temperature.

ASTM A206 Specification for Seamless Carbon-molybdenum Alloy-steel Pipe for High-temperature Service (Tentative). A206 seamless carbon-molybdenum alloy-steel pipe for high-temperature service is made of steel selected for its high-temperature properties, weldability, and general stability under severe working conditions. This steel is made either by the open-hearth or the electric-furnace process, with the grain size carefully checked to meet the specified limits. After the finishing process is completed, the pipe is annealed. The chemical composition and the tensile properties are tabulated for quick reference. When the purchaser so requests, a photomicrograph at 100 diameters, taken from a specimen cut, is furnished by the manufacturer. As specified by the code, the use of A206 is, at the present time, quite rigidly controlled.

ASTM A158 Specification for Seamless Alloy-steel Pipe for High-temperature Service

Table 11-2a. Standard Wrought-steel-pipe Data

(All dimensions and weights are nominal)

Size in	Diameter		Thickness in	Circumference		Transverse areas			Length of pipe/sq ft of		Length of pipe containing 1 cu ft ft	Weight/ft		Number of threads/in of screw	Moment of inertia	Wt. of water/ft lb
	External in	Internal in		External in	Internal in	External sq in	Internal sq in	Metal sq in	External surface ft	Internal surface ft		Plain ends lb	Threaded and coupled lb			
⅛	0.405	0.269	0.068	1.272	0.845	0.129	0.057	0.072	9.431	14.199	2533.775	0.244	0.245	27	0.00106	0.025
¼	0.540	0.364	0.088	1.696	1.144	0.229	0.104	0.125	7.073	10.493	1383.789	0.424	0.425	18	0.00331	0.045
⅜	0.675	0.493	0.091	2.121	1.549	0.358	0.191	0.167	5.658	7.747	754.360	0.567	0.568	18	0.00729	0.083
½	0.840	0.622	0.109	2.639	1.954	0.554	0.304	0.250	4.547	6.141	473.906	0.850	0.852	14	0.01709	0.132
¾	1.050	0.824	0.113	3.299	2.589	0.866	0.533	0.333	3.637	4.635	270.034	1.130	1.134	14	0.03704	0.231
1	1.315	1.049	0.133	4.131	3.296	1.358	0.864	0.494	2.904	3.641	166.618	1.678	1.684	11½	0.08734	0.375
1¼	1.660	1.380	0.140	5.215	4.335	2.164	1.495	0.669	2.301	2.767	96.275	2.272	2.281	11½	0.1947	0.65
1½	1.900	1.610	0.145	5.969	5.058	2.835	2.036	0.799	2.010	2.372	70.733	2.717	2.731	11½	0.3099	0.88
2	2.375	2.067	0.154	7.461	6.494	4.430	3.355	1.075	1.608	1.847	42.913	3.652	3.678	11½	0.6657	1.45
2½	2.875	2.469	0.203	9.032	7.757	6.492	4.788	1.704	1.328	1.547	30.077	5.793	5.819	8	1.530	2.07
3	3.500	3.068	0.216	10.996	9.638	9.621	7.393	2.228	1.091	1.245	19.479	7.575	7.616	8	3.017	3.20
3½	4.000	3.548	0.226	12.566	11.146	12.566	9.886	2.680	0.954	1.076	14.565	9.109	9.202	8	4.788	4.29
4	4.500	4.026	0.237	14.137	12.648	15.904	12.730	3.174	0.848	0.948	11.312	10.790	10.889	8	7.233	5.50
4½	5.000	4.506	0.247	15.708	14.156	19.635	15.947	3.688	0.763	0.847	9.030	12.538	12.642	8	10.44	6.91
5	5.563	5.047	0.258	17.477	15.856	24.306	20.006	4.300	0.686	0.756	7.198	14.617	14.810	8	15.16	8.67
6	6.625	6.065	0.280	20.813	19.054	34.472	28.891	5.581	0.576	0.629	4.984	18.974	19.185	8	28.14	12.51
7	7.625	7.023	0.301	23.955	22.063	45.664	38.738	6.926	0.500	0.543	3.717	23.544	23.769	8	46.52	16.80
8	8.625	8.071	0.277	27.096	25.356	58.426	51.161	7.265	0.442	0.473	2.815	24.696	25.000	8	63.35	22.18
8	8.625	7.981	0.322	27.096	25.073	58.426	50.027	8.399	0.442	0.478	2.878	28.554	28.809	8	72.49	21.70
9	9.625	8.941	0.342	30.238	28.089	72.760	62.786	9.974	0.396	0.427	2.294	33.907	34.188	8	107.6	27.20
10	10.750	10.192	0.279	33.772	32.019	90.763	81.585	9.178	0.355	0.374	1.765	31.201	32.000	8	125.9	35.37
10	10.750	10.136	0.307	33.772	31.843	90.763	80.691	10.072	0.355	0.376	1.785	34.240	35.000	8	137.4	34.95
10	10.750	10.020	0.365	33.772	31.479	90.763	78.855	11.908	0.355	0.381	1.826	40.483	41.132	8	160.7	34.20
11	11.750	11.000	0.375	36.914	34.558	108.434	95.033	13.401	0.325	0.347	1.515	45.557	46.247	8	217.0	41.20
12	12.750	12.090	0.330	40.055	37.982	127.676	114.800	12.876	0.299	0.315	1.254	43.773	45.000	8	248.5	49.70
12	12.750	12.000	0.375	40.055	37.699	127.676	113.097	14.579	0.299	0.318	1.273	49.562	50.706	8	279.3	49.00

Table 11-2b. Extra-strong Wrought-steel-pipe Data

(All dimensions and weights are nominal)

Size, in	Diameters		Thickness, in	Circumference		Transverse areas			Length of pipe/sq ft of		Length of pipe containing 1 cu ft, ft	Weight/ft plain ends, lb	Moment of inertia	Wt of water/ft, lb
	External, in	Internal, in		External, in	Internal, in	External, sq in	Internal, sq in	Metal, sq in	External surface, ft	Internal surface, ft				
1/8	0.405	0.215	0.095	1.272	0.675	0.129	0.036	0.093	9.431	17.766	3966.392	0.314	0.00122	0.016
1/4	0.540	0.302	0.119	1.696	0.949	0.229	0.072	0.157	7.073	12.648	2010.290	0.535	0.00377	0.031
3/8	0.675	0.423	0.126	2.121	1.329	0.358	0.141	0.217	5.658	9.030	1024.689	0.738	0.00862	0.061
1/2	0.840	0.546	0.147	2.639	1.715	0.554	0.234	0.320	4.547	6.995	615.017	1.087	0.02008	0.102
3/4	1.050	0.742	0.154	3.299	2.331	0.866	0.433	0.433	3.637	5.147	333.016	1.473	0.04479	0.188
1	1.315	0.957	0.179	4.131	3.007	1.358	0.719	0.639	2.904	3.991	200.193	2.171	0.1056	0.312
1¼	1.660	1.278	0.191	5.215	4.015	2.164	1.283	0.881	2.301	2.988	112.256	2.996	0.2418	0.56
1½	1.900	1.500	0.200	5.969	4.712	2.835	1.767	1.068	2.010	2.546	81.487	3.631	0.3912	0.77
2	2.375	1.939	0.218	7.461	6.092	4.430	2.953	1.477	1.608	1.969	48.766	5.022	0.8679	1.28
2½	2.875	2.323	0.276	9.032	7.298	6.492	4.238	2.254	1.328	1.644	33.976	7.661	1.924	1.87
3	3.500	2.900	0.300	10.996	9.111	9.621	6.605	3.016	1.091	1.317	21.801	10.252	3.894	2.86
3½	4.000	3.364	0.318	12.566	10.568	12.566	8.888	3.678	0.954	1.135	16.202	12.505	6.280	3.84
4	4.500	3.826	0.337	14.137	12.020	15.904	11.497	4.407	0.848	0.998	12.525	14.983	9.610	4.98
4½	5.000	4.290	0.355	15.708	13.477	19.635	14.455	5.180	0.763	0.890	9.962	17.611	14.05	6.27
5	5.563	4.813	0.375	17.477	15.120	24.306	18.194	6.112	0.686	0.793	7.915	20.778	20.67	7.88
6	6.625	5.761	0.432	20.813	18.099	34.472	26.067	8.405	0.576	0.663	5.524	28.573	40.49	11.29
7	7.625	6.625	0.500	23.955	20.813	45.664	34.472	11.192	0.500	0.576	4.177	38.048	71.37	14.95
8	8.625	7.625	0.500	27.096	23.955	58.426	45.663	12.763	0.442	0.500	3.154	43.388	105.7	19.78
9	9.625	8.625	0.500	30.238	27.096	72.760	58.426	14.334	0.396	0.442	2.464	48.728	149.6	25.30
10	10.750	9.750	0.500	33.772	30.631	90.763	74.662	16.101	0.355	0.391	1.929	54.735	212.0	32.35
11	11.750	10.750	0.500	36.914	33.772	108.434	90.763	17.671	0.325	0.355	1.587	60.075	280.1	39.40
12	12.750	11.750	0.500	40.055	36.914	127.676	108.434	19.242	0.299	0.325	1.328	65.415	361.5	46.92

Table 11-2c. Double-extra-strong Wrought-steel-pipe Data

(All dimensions and weights are nominal)

Size in	Diameter		Thickness in	Circumference		Transverse areas			Length of pipe/sq ft		Length of pipe containing 1 cu ft ft	Weight/ft plain ends lb	Moment of inertia	Wt of water/ft, lb
	External in	Internal in		External in	Internal in	External sq in	Internal sq in	Metal sq in	External surface ft	Internal surface ft				
½	0.840	0.252	0.294	2.639	0.792	0.554	0.050	0.504	4.547	15.157	2887.164	1.714	0.0242	0.022
¾	1.050	0.434	0.308	3.299	1.363	0.866	0.148	0.718	3.637	8.801	973.404	2.440	0.0579	0.064
1	1.315	0.599	0.358	4.131	1.882	1.358	0.282	1.076	2.904	6.376	510.998	3.659	0.1405	0.122
1¼	1.660	0.896	0.382	5.215	2.815	2.164	0.630	1.534	2.301	4.263	228.379	5.214	0.3411	0.273
1½	1.900	1.100	0.400	5.969	3.456	2.835	0.950	1.885	2.010	3.472	151.526	6.408	0.5678	0.42
2	2.375	1.503	0.436	7.461	4.722	4.430	1.774	2.656	1.608	2.541	81.162	9.029	1.311	0.77
2½	2.875	1.771	0.552	9.032	5.564	6.492	2.464	4.028	1.328	2.156	58.457	13.695	2.871	1.07
3	3.500	2.300	0.600	10.996	7.226	9.621	4.155	5.466	1.091	1.660	34.659	18.583	5.993	1.80
3½	4.000	2.728	0.636	12.566	8.570	12.566	5.845	6.721	0.954	1.400	24.637	22.850	9.848	2.53
4	4.500	3.152	0.674	14.137	9.902	15.904	7.803	8.101	0.848	1.211	18.454	27.541	15.28	3.38
4½	5.000	3.580	0.710	15.708	11.247	19.635	10.066	9.569	0.763	1.066	14.306	32.530	22.62	4.36
5	5.563	4.063	0.750	17.477	12.764	24.306	12.966	11.340	0.686	0.940	11.107	38.552	33.63	5.61
6	6.625	4.897	0.864	20.813	15.384	34.472	18.835	15.637	0.576	0.780	7.646	53.160	66.33	8.16
7	7.625	5.875	0.875	23.955	18.457	45.664	27.109	18.555	0.500	0.650	5.312	63.079	107.5	11.75
8	8.625	6.875	0.875	27.096	21.598	58.426	37.122	21.304	0.442	0.555	3.879	72.424	162.0	16.10

Table 11-3. Dimensions of Welded and Seamless Steel Pipe

Nominal pipe size	Outside diam	Nominal wall thicknesses for schedule numbers									
		Schedule 10	Schedule 20	Schedule 30	Schedule 40	Schedule 60	Schedule 80	Schedule 100	Schedule 120	Schedule 140	Schedule 160
⅛	0.405				**0.068**		**0.095**				
¼	0.540				**0.088**		**0.119**				
⅜	0.675				**0.091**		**0.126**				
½	0.840				**0.109**		**0.147**				0.187
¾	1.050				**0.113**		**0.154**				0.218
1	1.315				**0.133**		**0.179**				0.250
1¼	1.660				**0.140**		**0.191**				0.250
1½	1.900				**0.145**		**0.200**				0.281
2	2.375				**0.154**		**0.218**				0.343
2½	2.875				**0.203**		**0.276**				0.375
3	3.500				**0.216**		**0.300**				0.437
3½	4.00				**0.226**		**0.318**				
4	4.500				**0.237**		**0.337**		0.437		0.531
5	5.563				**0.258**		**0.375**		0.500		0.625
6	6.625				**0.280**		**0.432**		0.562		0.718
8	8.625		0.250	**0.277**	**0.322**	0.406	**0.500**	0.593	0.718	0.812	0.906
10	10.75		0.250	**0.307**	**0.365**	**0.500**	0.593	0.718	0.843	1.000	1.125
12	12.75		0.250	**0.330**	0.406	0.562	0.687	0.843	1.000	1.125	1.312
14 OD	14.00	0.250	0.312	0.375	0.437	0.593	0.750	0.937	1.062	1.250	1.406
16 OD	16.00	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.437	1.562
18 OD	18.00	0.250	0.312	0.437	0.562	0.718	0.937	1.156	1.343	1.562	1.750
20 OD	20.00	0.250	0.375	0.500	0.593	0.812	1.031	1.250	1.500	1.750	1.937
24 OD	24.00	0.250	0.375	0.562	0.687	0.937	1.218	1.500	1.750	2.062	2.312
30 OD	30.00	0.312	0.500	0.625							

All dimensions are given in inches

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions For tolerances on wall thicknesses see appropriate material specification

Thicknesses shown in boldface type in Schedules 30 and 40 are identical with thicknesses for standard weight pipe in former lists those in Schedules 60 and 80 are identical with thicknesses for extra strong pipe in former lists

The schedule numbers indicate approximate values of the expression $1\,000 \times P/S$

Owing to a necessary departure from the old standard weight and extra strong thicknesses in the 12-in size in Schedules 40 and 60 the new thicknesses are not as yet stocked by all manufacturers and jobbers Hence where agreeable to the purchaser and suitable for the service conditions the old standard weight 0.375-in wall pipe corresponding to a $1\,000\,P/S$ value of 37.7 is still available and can be substituted for the 0.406 in wall and the old extra strong 0.500-in. wall pipe corresponding to a $1{,}000\,P/S$ value of 55 can be substituted for the 0.562 in wall

(*Tentative*) ASTM A158 pipe covers several varieties of ferritic and austenitic seamless pipe and is applicable to services where the metal temperatures are high or where corrosion may be a factor These steels are characterized by their high-temperature properties, weldability, corrosive resistance, and general stability at the working temperatures They are made either by the open-hearth or electric-furnace process The heat-treatment to be used is carefully followed, and the tensile-strength limits are tabulated as a means of ready reference. Bending, flattening, and hydrostatic tests are made to check the quality of the product

The petroleum industry is a large user of alloy-steel piping Most of the low-alloy and 4 to 6 per cent chrome varieties of pipe were developed to combat the extremely corrosive conditions encountered in modern petroleum-refining processes Many of the cracking processes involve extreme temperatures, thus necessitating the use of higher alloy steels

The relatively high cost of alloy-steel pipe limits its application to those installations involving extreme temperatures and having severe corrosion conditions and where carbon steel is not found to be entirely satisfactory

Table 11-4. Nominal Weights of Welded and Seamless Steel Pipe*

Nominal pipe size, in.	Schedule 10, plain ends	Schedule 20, plain ends	Schedule 30		Schedule 40		Schedule 60, plain ends	Schedule 80, plain ends	Schedule 100, plain ends	Schedule 120, plain ends	Schedule 140, plain ends	Schedule 160, plain ends
			Plain ends	Threads† and couplings	Plain ends	Threads† and couplings						
⅛	...	...	....	....	**0.25**	**0.25**	....	**0.32**				
¼	...	....	....	....	**0.43**	**0.43**	....	**0.54**				
⅜	..	....	....	....	**0.57**	**0.57**	....	**0.74**				
½	..	....	....	....	**0.85**	**0.86**	....	**1.09**	....	....	....	1 31
¾	...	....	. ..	....	**1.14**	**1.14**	.. .	**1.48**	....	....	....	1.94
1	....	....	....	....	**1.68**	**1.69**	....	**2.18**	....	....	....	2.85
1¼	..	....	....	....	**2.28**	**2.29**	....	**3.00**	....	....	....	3.77
1½	...	....	....	....	**2.72**	**2.74**	....	**3.64**	....	....	....	4.86
2	...	....	...	....	**3.66**	**3.68**	....	**5.03**	....	..	....	7.45
2½	....	....	....	. .	**5.80**	**5.82**	....	**7.67**	...	....	.. .	10.0
3	....	....	....	....	**7.58**	**7.62**	....	**10.3**	....	....	. .	14.3
3½	....	....	....	....	**9.11**	**9.21**	....	**12.5**				
4	....	....	....	. .	**10.8**	**10.9**	....	**15.0**	....	19.0	. .	22.6
5	....	...	....	. .	**14.7**	**14.9**	....	**20.8**	...	27.1	. ..	33.0
6	....	....	.		**19.0**	**19.2**	....	**28.6**	.	36.4	....	45.3
8	....	22 4	**24.7**	**25.0**	**28.6**	**28.8**	35 7	**43.4**	50 9	60.7	67.8	74.7
10	....	28 1	**34.3**	**35.0**	**40.5**	**41.2**	**54.8**	64 4	77 0	89 2	105	116
12	..	33.4	**43.8**	**45.0**	53 6	55.0	73 2	88 6	108	126	140	161
14 OD	36 8	45.7	54.6	....	63 3	....	85 0	107	131	147	171	190
16 OD	42.1	52.3	62.6	....	82 8	...	108	137	165	193	224	241
18 OD	47.4	59.0	82.0	....	105	....	133	171	208	239	275	304
20 OD	52 8	78.6	105	....	123	....	167	209	251	297	342	374
24 OD	63 5	94.7	141	..	171	....	231	297	361	416	484	536
30 OD	99.0	158	197									

* Weights are given in pounds per linear foot and are for pipe with plain ends except for sizes which are commercially available with threads and couplings for which both weights are listed.

† Weight for line pipe with couplings are slightly greater than shown in Schedules 30 and 40 and may be found in API Specification 5-L. Weights shown in boldface type in Schedules 30 and 40 are identical with weights for standard-weight pipe in former lists; those in Schedules 60 and 80 are identical with weights for extra-strong pipe in former lists.

The schedule numbers indicate approximate values of the expression 1,000 × P/S.

Genuine Wrought-iron Pipe. The basic material used in the manufacture of genuine wrought-iron pipe is made from pig-puddled or processed wrought iron and is free from any admixture of iron scrap or steel. Wrought iron is defined as a ferrous material aggregated from a solidifying mass of pasty particles of highly refined iron with which, without subsequent fusion or chemical relationship, is incorporated a minutely and uniformly distributed quantity of slag. The durability of wrought iron when subject to general corrosive conditions is attributed to the siliceous slag fibers imbedded in the base metal. Wrought-iron pipe finds wide application in plumbing, water, and underground pipe lines where certain corrosive conditions exist.

This pipe is produced in butt-welded and lap-welded form. Sizes from ⅛ to 2 in., inclusive, are produced in the form of butt-welded pipe; sizes from 1¼ to 12 in. inclusive, are produced by the lap-welded process; sizes from 14 to 24 in. OD are either hammer-welded or fusion-welded. In the main genuine wrought-iron pipe is most used in those sizes 12 in. in diameter or smaller.

Genuine wrought-iron pipe may be obtained in standard, extra-strong, and double extra-strong thicknesses and is manufactured to conform to ASTM A72 specification for general usage, coiling, bending, flanging, and other special purposes. Although this material is more expensive than carbon steel, the producers claim that its resistance to corrosion offsets any difference in cost when the length of service is considered.

Table 11-5. Standard Pipe Sizes for Brass and Copper Pipe

Nominal size (iron pipe size), in.	Diameters		Wall thickness, in	Weight/lin ft		
	External, in.	Internal, in		Yellow brass, lb	Red brass, lb	Copper, lb
			Regular			
⅛	0.405	0.281	0.0620	0.246	0.253	0.259
¼	0.540	0.375	0.0825	0.437	0.450	0.460
⅜	0.675	0.494	0.0905	0.612	0.630	0.643
½	0.840	0.625	0.1075	0.911	0.938	0.957
¾	1.050	0.822	0.1140	1.24	1.27	1.30
1	1.315	1.062	0.1265	1.74	1.79	1.83
1¼	1.660	1.368	0.1460	2.56	2.63	2.69
1½	1.900	1.600	0.1500	3.04	3.13	3.20
2	2.375	2.062	0.1565	4.02	4.14	4.23
2½	2.875	2.500	0.1875	5.83	6.00	6.14
3	3.500		0.2190	8.31	8.56	8.75
3½	4.000	500	0.2500	10.85	11.17	11.41
4	4.500	000	0.2500	12.29	12.66	12.94
4½	5.000	500	0.2500	13.74	14.15	14.46
5	5.563	063	0.2500	15.40	15.85	16.21
6	6.625	6.125	0.2500	18.44	18.90	19.41
7	7.625	7.062	0.2815	23.92	24.63	25.17
8	8.625	8.000	0.3125	30.05	30.95	31.63
10	10.750	10.019	0.3655	43.91	45.20	46.22
12	12.750	12.000	0.3750	53.71	55.29	56.51
			Extra strong			
⅛	0.405	0.205	0.100	0.353	0.363	0.371
¼	0.540	0.294	0.123	0.593	0.611	0.624
⅜	0.675	0.421	0.127	0.805	0.829	0.847
½	0.840	0.542	0.149	1.19	1.23	1.25
¾	1.050	0.736	0.157	1.62	1.67	1.71
1	1.315	0.951	0.182	2.39	2.46	2.51
1¼	1.660	1.272	0.194	3.30	3.39	3.46
1½	1.900	1.494	0.203	3.99	4.10	4.19
2	2.375	1.933	0.221	5.51	5.67	5.79
2½	2.875	2.315	0.280	8.41	8.66	8.84
3	3.500	2.892	0.304	11.24	11.57	11.82
3½	4.000	3.358	0.321	13.67	14.07	14.37
4	4.500	3.818	0.341	16.41	16.89	17.25
4½	5.000	4.250	0.375	20.07	20.66	21.10
5	5.563	4.813	0.375	22.52	23.18	23.67
6	6.625	5.751	0.437	31.32	32.21	32.93
7	7.625	6.625	0.500	41.23	42.43	43.34
8	8.625	7.625	0.500	47.02	48.39	49.42
10	10.750	9.750	0.500	59.32	61.05	62.40

Toncan Iron Pipe; Copper-bearing Steel Pipe. In addition to carbon steel, occasional use is made of copper-bearing steel pipe or copper-molybdenum iron (Toncan) pipe. Copper-bearing steel is steel with an addition of 0.20 to 0.35 per cent copper. Toncan iron is open-hearth iron containing 0.10 per cent copper and 0.57 per cent molybdenum.

Brass and Copper Pipe. The three basic materials of this group consist of yellow brass, having an approximate composition of 67 per cent copper and 33 per cent zinc; red brass, which has an approximate composition of 85 per cent copper and 15 per cent zinc; and copper pipe, which is manufactured close to a 99.9 per cent copper

Table 11-6. Linear Expansion of Pipe

Temp, deg F	Inches/100 ft				Temp, deg F	Inches/100 ft				Temp, deg F	Inches/100 ft	
	Steel	Wrought iron	Brass	Copper		Steel	Wrought iron	Brass	Copper		Steel	Wrought iron
0	0 0	0 0	0 0	0 0	340	2 76	2 87	4 30	3 95	680	5 965	6 20
20	0 15	0 155	0 25	0 25	360	2 935	3 05	4 58	4 15	700	6 17	6.42
40	0 30	0 31	0 45	0 45	380	3 11	3 235	4 85	4 40	720	6 375	6 625
60	0 455	0 475	0 67	0 65	400	3 29	3 43	5 14	4 64	740	6 58	6 835
80	0 61	0 63	0 92	0 87	420	3 465	3 62	5 45	4 89	760	6 79	7 05
100	0 77	0 80	1 17	1 10	440	3 65	3 805	5 73	5 15	780	6 99	7 275
120	0 915	0 96	1 41	1 35	460	3 835	4 00	5 96	5 37	800	7 21	7.49
140	1 075	1 13	1 69	1 57	480	4 02	4 19	6 25	5.64	820	7 415	7 73
160	1 235	1 29	1 91	1 77	500	4 21	4 39	6 55	5 88	840	7 63	7 93
180	1 40	1 46	2 15	2 00	520	4 39	4 58			860	7 84	8 145
200	1 57	1 64	2 43	2 25	540	4 59	4 78			880	8 055	8 37
220	1 73	1 81	2 71	2 52	560	4 78	4 975			900	8 28	8 60
240	1 89	1 98	2 97	2 74	580	4 975	5 175			920	8 495	8 82
260	2 065	2 16	3 22	2 95	600	5 17	5 38			940	8 72	9 05
280	2 23	2 335	3 47	3 17	620	5 365	5 58			960	8 945	9.28
300	2 41	2 52	3 76	3 42	640	5 565	5 785			980	9 17	9 52
320	2 59	2 70	4 07	3 70	660	5 765	5 99			1 000	9 40	9 75

analysis. The pipe is seamless and is formed by cupping and drawing or by extrusion.

Yellow brass is suitable for most general water installations and is cheaper than either red brass or copper. Red brass, while more expensive than yellow brass, is easier to fabricate and, owing to its smaller zinc content, does not deteriorate so rapidly for severely corrosive water service. On the other hand, yellow brass has been found to be subject to corrosion due to dezincification.

Copper and brass are extensively used for handling waters, acids, alkalies, and other generally corrosive fluids.

Pipe Dimensions

Steel Pipe. Since steel pipe must be suitable for a wide range of services with respect to pressure, temperature, corrosion, and the like, it is produced in a variety of wall thicknesses and different weights for each of the various sizes. As a general rule, particularly in the case of carbon steel, pipe is classified by the terms standard, extra strong, and double extra strong. A more suitable method, which is being more universally applied particularly to alloy-steel pipe, has been developed by the ASA, *viz.*, the classification of pipe by schedule number (see Tables 11-2 to 11-5).

Pressure Rating for Steel Pipe. For pressure rating for steel pipe, refer to ASA Code for Pressure Piping, ASME Boiler Construction Code, or any code or standard applying to a specific application.

Expansion of Piping Materials. For expansion of piping materials, see Table 11-6.

VALVES

Valves are among the most important of all control instruments used in industry. Their function is to control and stop the flow of water, oil, gas, steam, air, chemicals, and in fact, any liquid or gas conveyed from place to place. Their selection for any particular service condition can be best accomplished by one who is thoroughly acquainted with their individual characteristics.

Classification of Valves by Material

There are three basic materials from which the principal parts of the great majority of valves are made. These are brass, iron, and steel. Each has its particular purpose in valve design. All these materials are made in a large variety of compositions having varying strengths and resistance to the effect of corrosion and temperature.

Brass. Brass is used extensively for bodies and bonnets of valves where resistance to corrosion by liquids or gases is desired. It is a satisfactory material to resist the corrosive action of most water and atmospheric conditions and can be used for most normal pressure and temperature conditions. It can be cast or forged and is reasonably easy to machine. Because it is much higher in cost than cast iron, its use usually has been confined to sizes 3 in. and smaller. Many large brass valves are used, however, to resist the corrosive action of sea water. Some of the specifications generally referred to for brass valves are

ASTM B62 Specification for Composition Brass or Ounce Metal Castings. ASTM B62 is widely used for valves rated as high as 150 lb steam pressure. It is limited by most codes to a maximum temperature of 400°F.

ASTM B61 Specification for Steam or Valve Bronze Castings. A brass used by the majority of manufacturers for valves having ratings higher than 150 lb steam is covered by the above specification. While some codes limit the use of this material to 500°F, most manufacturers recommend it for as high as 550°F.

Cast Iron. Cast iron is generally used for valve sizes 2 in. and larger except where a more ductile or more corrosion-resistant material is necessary. Although cast iron is used for reasonably high pressures, the metal sections must be made quite heavy because of its relatively low physical properties which consequently makes it more difficult to obtain sound pressure-tight castings. Its use, therefore, is generally restricted to pressures 500 lb and lower, particularly in sizes above 12 in. Codes in general limit the use of cast iron to a maximum temperature of 450°F. Cast iron for valves is covered by ASTM Specification A126.

Cast Steel and Forged Steel. Cast steel and forged steel are the accepted materials in valve bodies and bonnets for extremely high pressures, particularly when used in services where temperatures are in excess of 500°F. Its high strength, high ductility, and shock resistance, as well as resistance to the effect of high temperatures, make steel an extremely important material in the valve industry. The many different compositions and alloys, ranging from carbon steel to stainless and high-strength steels, make this material applicable for use in valves of all sizes, shapes, and designs.

Classification of Valves by Types

The basic types of valves are globe, gate, and check. There are many variations in the design of each type which make each one particularly adaptable to a specific use. In a piping system there are a multitude of requirements which must be considered when the type of valve is to be selected.

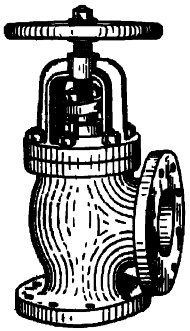

FIG. 11-1. Angle valve.

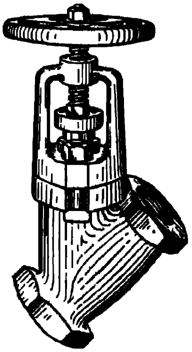

FIG. 11-2. Y-pattern globe valve.

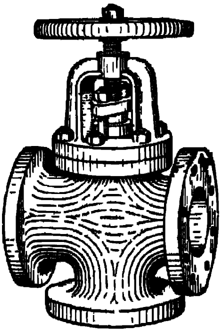

FIG. 11-3. Cross valve.

Globe Valves. The name "globe" probably originated because the common type of globe-valve body was globular in shape. The term globe, however, has for years been generally accepted as applied to the fundamental principle of plugging or stopping off a valve seat. For this reason, the term "globe type" has become quite common as it frequently applies to valves having shapes differing from the original globe valve, such as the angle, Y pattern, or cross.

The three globe-type valves illustrated in Figs. 11-1, 11-2, and 11-3 operate on the same principle as a regular globe valve. As shown in the cuts, any valve in which the closing and opening of the port through the valve seat is accomplished by forcing a "disc" down against the seat or into the seat opening to close it completely and where to open it the disc must be withdrawn is considered to be a "globe type." The force to actuate the disc is applied through the valve "stem" either by means of the stem threads or other actuating members such as levers, operating cylinders, toggle, etc.

Globe-type valves have always been preferred for throttling service, as they are easily controlled, but as explained hereinafter, the plug-type metal-disc design has given most satisfactory service.

Method of Operation. In globe valves, two common design principles are applied to the operating mechanism. One of these is the inside-screw rising stem (Fig. 11-4). In this type, the stem thread rotates in a corresponding or mating thread inside the bonnet so that, as the stem rises or lowers, it revolves as it moves the discs. Owing to the fact that the operating threads are exposed to the fluid within the valve, this design should not be used if the threads are likely to be seriously damaged by the fluid. For example, if the fluid contains an abrasive substance, excessive wear will quite likely occur in the stem threads and thus cause difficult operation.

In such cases a valve with an external operating mechanism such as the outside-screw and yoke (OS and Y) rising stem (Fig 11-5) should be used This type is generally specified in connection with larger sized valves. In this valve the stem thread rotates in a corresponding thread in the upper hub of the yoke. The stem itself also rises and lowers, thereby causing the disc to move up or down. It will be noted that the stem threads in this valve are not exposed to the fluid inside the valve.

There are also numerous special operating means such as the automatic-closing type of sliding stem as shown in Fig. 11-6. Globe-type valves are also made with

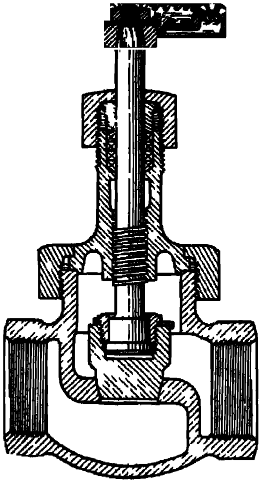

FIG. 11-4. Inside-screw, rising-stem globe valve.

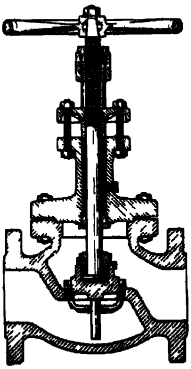

FIG. 11-5. Outside-screw and yoke (OS and Y) globe valve.

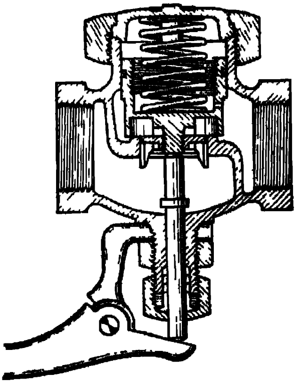

FIG. 11-6. Quick-opening sliding-stem globe valve.

special operating means such as gearing, cylinder operation, motor operation, etc. For all these the operating mechanism within the valve proper must be designed to suit the particular operating means.

Seat and Disc. The most important features of any valve, as far as service and tightness are concerned, are the material, design, and workmanship of the seat and disc. Briefly, the three designs of globe-type valves now in most common use are

1. Metal disc—narrow conical seat
2. Plug disc
3. Composition disc

There are many other special seat designs, but the general field is quite well covered by these three. Each has its advantages and each serves a useful purpose in industry.

Metal Disc—Narrow Conical Seat. The metal disc with a narrow conical seat usually has a ball-shaped surface on the disc, a conical flat seating surface in the body, and is generally referred to as the conventional design of globe valve. Figures 11-7 and 11-8 illustrate this design of seat and disc as applied to brass and steel globe valves.

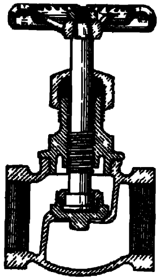
FIG. 11-7. Brass globe valve with ball disc and conical seat.

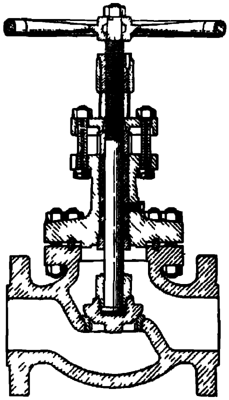
FIG. 11-8. Steel globe valve with ball disc and conical seat.

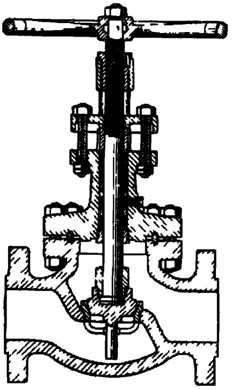
FIG. 11-9. Globe valve with plug-type disc

Because of the line bearing obtained in seating, this design is easily made tight, can be refaced or reground readily, and is adaptable to many types of services. Because operators have found that this type of seat provides an excellent means for removing coke, it is often specified for high-temperature oil service. Coke forms on metal surfaces in lines carrying high-temperature oil vapor. This hard brittle coating on the seating surfaces is generally not uniform in thickness and would soon lead to leaky valves. Because of the excessively high loading on the seating surface, the coke will crack when the operator closes the valve tightly. The valve is then opened slightly permitting a high-velocity flow across the seat, which action washes away the coke particles. This procedure is generally repeated several times when the valve is closed, particularly if the valve has been in service with the disc in the open position for a relatively long period.

The principal disadvantage of this type of seat is that very hard dirt or foreign matter may dent the seating surface, causing leakage. The result of such leakage is the inevitable wire-drawing action, which will destroy the seat in a very short time. Even though the dirt may not be hard enough to cut into the metal, if it holds the disc away from its seat and is allowed to remain in this position, the flow through the valve will cut the seat by wire-drawing action. When any valve fails to seat properly, it should be opened part way again and allowed to blow a short time. This, if repeated several times, will sometimes remove the obstruction and materially increase the life of the valve.

Plug Disc. The plug-type disc design has gained in popularity in recent years because of its ability to (1) throttle flow more effectively than other designs and (2) resist the wire-drawing action of high-velocity flow. This construction as illustrated in Fig. 11-9 comprises a plug-type disc and mating seat which produces

1. A gradual increase or decrease in the distance between the seat and disc surfaces when the valve is slowly opened or closed. This causes a gradual change in flow area thereby providing for close throttling control.

2. A wide contact area between disc and seat surfaces. In this manner the effect of wire drawing is greatly minimized as cutting always starts at outer edges of a bearing surface. In addition, any indentations in the seat caused by hard particles of foreign matter are unlikely to extend across the entire disc or seat surfaces and, therefore, are not apt to cause leakage.

If top-grade seating materials are used in this design, it is extremely difficult to surpass the service life of a plug-disc globe valve.

Composition Disc. The composition-disc design (Fig. 11-10) has been widely used for many services because of the ease with which a disc can be replaced. Its principle advantages are: (1) It requires less power to seat tight than a metal-disc valve. (2) Disc can be renewed with little effort. (3) It is less likely to be damaged by dirt or foreign matter than the metal disc with narrow seat. Dirt will imbed itself into the disc, and the valve will remain tight unless the dirt particle is exceptionally large. (4) By the use of different discs, the valve can be made suitable for different services such as steam, water, air, or gas.

For the most part, the use of composition discs has been confined to small sizes and for service conditions not to exceed 450°F because of the difficulties of obtaining suitable compositions for larger sizes and higher temperatures. A composition-disc valve should never be used for throttling as it may not give satisfactory service. Composition discs generally break down more quickly than metal discs when throttled, and on steam service a composition disc will expand enough to effect the steam flow if throttled closely.

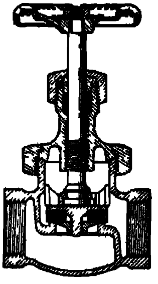

FIG. 11-10. Globe valve with composition disc.

Although a composition-disc valve is a great convenience when used in the proper service, it requires reasonable attention the same as any other mechanical device. If a disc has been in service long enough to cut out or leak for any reason, it must be replaced within a reasonable time or the valve seat will be damaged by wire drawing. It is a wise precaution to examine the seat before replacing the disc and, if it has been damaged, to take a cut off the seating surface with a reseating tool before placing the valve back in service.

Proper Installation of Globe-type Valves. There has always been a difference of opinion as to whether a globe-type valve should be installed in the line with the inlet pressure acting under the disc or above the disc. Obviously, if the pressure is under the disc, the greatest force is required at the end of the closing operation to make the seat tight. If the pressure is above the disc, the greatest force is required at the start of the opening operation to pull the disc away from the seat. In either of these operations the force must overcome the full pressure per square inch multiplied by the area of the disc.

Let us assume that steam pressure is acting under the disc when the valve is closed. At the time of closure the pressure on the outlet side will be dissipated through whatever equipment the line is feeding. As the pressure on the outlet decreases, the temperature will also decrease. The stem which has been partly within the steam flow will cool and contract. Condensation in the line is likely to flow back into the valve submerging the stem and further cooling and contracting it. Since it is the stem which holds the disc tight against the seat, the contraction will decrease that load. Under these conditions the pressure under the disc will tend to force the disc away from the seat and may cause leakage.

Let us now assume that the steam pressure is acting above the disc when the valve is closed. The stem is still subjected to full steam temperature, and there is no relaxation of load. It is, of course, possible that condensate may build up in the valve body and cool the stem. If that happens, the stem would contract as in the former case, but the pressure acting above the disc will tend to hold it against the seat. This is possible because there is a certain amount of lost motion between the stem thread and its mating thread and also in the disc-stem assembly.

On high pressures it is easier to make a globe valve tight with the pressure above the disc than with the pressure under the disc.

From the above it can be seen that the preferred position for a globe valve in a steam line is to have the pressure above the disc. This method of installation, however, may be undesirable in some specific cases. No matter how well a valve may be designed or built, there is always a possibility that the disc might become loose from the end of the stem. If that should occur in a boiler feed line, for example, and the pressure were above the disc, the disc would close the valve as in a check valve, and it would be impossible to get water into the boiler.

A good rule to follow is "If it would be dangerous to have the valve close (in the event of a disc becoming loose from the stem), the inlet pressure should be under the disc." Under most other conditions globe valves should be installed with pressure above the disc. One exception to this is the composition-disc globe valve. Experience has shown that a composition disc will last considerably longer with the pressure acting under the disc than with the pressure above the disc. The resiliency of the disc is in most cases sufficient to overcome any tendency for the stem to contract and cause leakage.

Gate Valves. The term "gate" as applied to valves refers to that type of valve in which the port is closed by a flat-faced disc sliding at right angles over the seat port to close it. Because of the nature of this means of closing, it is quite difficult to control flow by throttling, particularly if it is necessary to set the valve to deliver a definite volume of fluid. It is also objectionable to use a gate valve for throttling for other reasons. When the disc is in the partly open position, the seat faces are so close to each other that wire drawing is bound to result and cause damage to the seats. In the double-disc type, because the disc assembly consists of two or more parts, these parts are likely to vibrate and wear out rapidly. For these reasons a gate valve should preferably be kept either tight shut or wide open.

One of the principal advantages of the gate valve is the clear unobstructed flow opening which results in a very low pressure drop through the valve. This is the principal reason why gate valves are so generally used in liquid lines and in large h-p steam lines.

Methods of Operation. The four standard methods of operation used in gate valves are as follows:

1. Outside screw and yoke (OS and Y) (Fig. 11-11)
2. Inside screw, nonrising stem (Fig. 11-12)
3. Inside screw, rising stem (Fig. 11-13)
4. Sliding stem—quick opening (Fig. 11-14)

The outside-screw and yoke design shown in Fig. 11-11 is made in either the solid-wedge or double-disc design and in brass, iron, and steel. It has several important features.

1. The stem is nonturning, rises through the yoke sleeve which is revolved in the upper hub of the yoke by the wheel, and supplies a positive indication of the position of the disc.

2. Because the threads on the stem are outside the valve, they are not exposed to the wear often resulting from abrasives carried in the fluid. The stem threads can very easily be greased and their life materially increased. This practice is strongly recommended.

3. Many large valves are regularly made with ball-bearing yoke sleeves. This practically eliminates the excessive friction on the yoke-sleeve thrust collars.

The inside-screw nonrising-stem design shown in Fig. 11-12 is common to either brass or iron gate valves but is seldom used in steel. Its principal features are

1. Requires much less headroom than the OS and Y design.

2. The stem which revolves to open or close the valves does not rise since it is held in place by the collar enclosed in the top of the bonnet and will give less trouble from stuffing-box leakage than any other type of stem action.

3. The stem threads which screw into the disc hub, thus raising or lowering the disc, are always exposed to the fluid within the valve. This can be an advantage if the fluid has lubricating qualities and does not contain abrasive matter, but in the latter case the threads are subject to severe wear and possible jamming or sticking.

Valves of the inside-screw nonrising design should not be used on high-temperature service because of the danger of uneven expansion of the parts causing binding of the threads and resulting in a galling action. This is the principal reason why large iron nonrising-stem valves are not often used for steam service, and large steel valves are seldom made in this design.

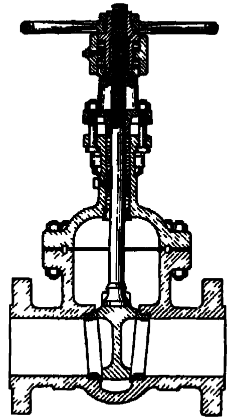

Fig. 11-11. Outside-screw and yoke (OS and Y) gate valve

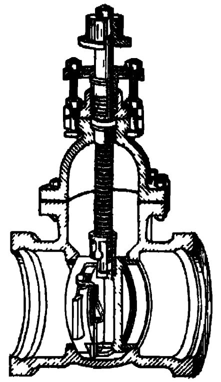

Fig. 11-12. Inside-screw, nonrising-stem gate valve.

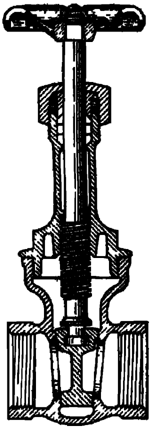

Fig. 11-13 Inside-screw, rising-stem gate valve.

The inside-screw rising-stem valve shown in Fig. 11-13 is seldom used except in small brass valves. In this design the stem rises and turns at the same time as the stem threads screw into a mating thread inside the bonnet. Its principal features are

1. The stem acts as an indicator of the position of the disc as it is raised or lowered.

2. It makes for a more compact small valve than the OS and Y as the need for a yoke is eliminated

3 Although these inside screw threads are subject to cutting action by any abrasive material in the line fluid, they give comparatively little trouble from this source on small valves.

The sliding-stem quick-opening design shown in Fig. 11-14 has the following features:

1. The stem slides up and down through the stuffing box and is actuated by a lever. It is intended only for special service where the flow must be cut off or on very quickly.

2. In the design of valve shown, the lever action is so designed that the maximum power necessary to close or open the disc is exerted at the point where the disc is in the closed position. As the valve is opened, the effective leverage is automatically reduced, and the speed of motion is increased, as it requires much less force to move the disc when it moves above the halfway-open point.

3. It should be remembered that no quick-opening valve should ever be used on water, oil, or other noncompressible fluid unless proper provision has been made in the line to prevent shock or water hammer. This is generally accomplished by the use of air chambers.

Disc Design. There are two basic designs of gate-valve discs in common use at the present time:

1. The wedge-disc design
2. The double-disc design

Wedge Disc. One of the best known disc designs is the solid wedge shown in Fig. 11-15. This design has been used in brass, iron, and steel gate valves for a great many years and with excellent results. Its principle of operation is quite obvious; the tapered wedge is driven in between the corresponding tapered seats, exerting an exceedingly high loading on the seat faces to hold tight against pressure. As the disc is withdrawn from the body seats, guided on the vertical center line of the body by a guide rib on each side, the disc faces are automatically withdrawn from the body seats because of the taper, and as a result the seat faces do not drag over each other after the fluid has started to flow. This prevents undue wear on the seat faces.

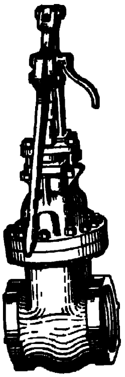

Fig. 11-14. Quick-opening, sliding-stem gate valve.

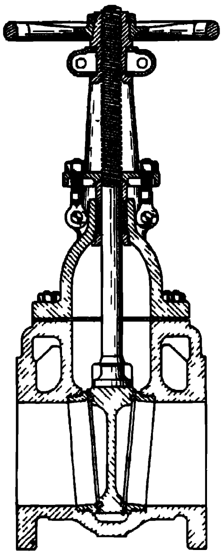

Fig. 11-15. Solid-wedge-disc gate valve.

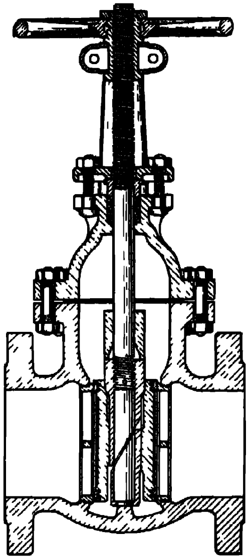

Fig. 11-16 Double-disc, parallel-seat gate valve.

The solid-wedge disc will perform satisfactorily in almost any service, such as water, air, gas, steam, or oil.

The advent of extremely high temperatures in power-plant service has produced a peculiar problem. When a solid-wedge disc is heated by superheated steam and then closed tightly and the valve allowed to cool, the disc may be held so tightly in the body that it is sometimes impossible to open the valve. Exhaustive testing has disclosed that, when the flow through an open valve causes heating of the parts, the disc which is in the bonnet is not heated to so high a temperature as the body of the valve and therefore does not expand so much as the body. This being the case, when the valve is closed and permitted to cool, the body contracts more than the disc thereby pinching it between the body seats. A steel disc of the flexible type has been developed which has sufficient spring to overcome this pinching action and therefore can be opened with comparative ease.

Double Disc. The double parallel disc, shown in Fig. 11-16, also is very popular. In this design the disc face drags over the seat face on the outlet side, and these two faces are held together by the pressure during the full travel of the valve This action has a tendency to wipe the seat clean, but in so doing, it accelerates wear and generally makes the valve harder to operate with high pressure against the disc. When the valve closes, the wedges between the two discs are stopped in the bottom of the body, causing the wedging surfaces to force the discs tightly against the body seats. There are various designs of wedging mechanisms for these discs, but they all operate on the same general principle.

Valves with double parallel seats are most commonly used in water-works lines and in pipe lines for cold oil and gas. Their use is insisted upon by many engineers owing probably to the fact that they are more easily repaired than the taper wedge-disc valves and also because their use in water works and pipe lines is traditional.

Check Valves. Check valves, which are designed to permit fluid flow in one direction only, fall into two general groups commonly known as swing check and lift check. Their purpose is extremely important to proper functioning of any piping system because of their quick automatic action and their sensitiveness to changes in flow conditions. Further, they will prevent dangerous backflow in a line when two or more fluids are being supplied to a common point at different pressures. Check valves might well be called "safety-first" valves.

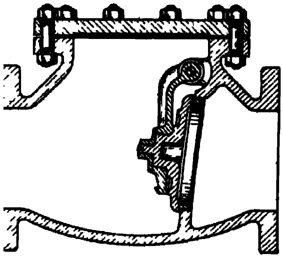
Fig. 11-17. Swing check valve.

Being automatic in action, check valves are subjected to greater abuse than are manually operated valves. Operators rely on the performance of their automatic features without giving any particular attention to periodic inspection. In addition, automatic action means that discs move back and forth with every fluctuation of flow and pressure and that this movement of the disc is not limited from one end of its travel to the other. It should be remembered that the only thing that makes a check valve tight when closed is the action of the pressure against the disc in the outlet end of the valve. The full line pressure is seldom available to accomplish this because the only effective holding power is the difference between the pressure on the outlet side and that on the inlet side. Although the slightest pressure differential will cause the valve to close, it requires a reasonable difference to hold the disc pressure-tight. As a result, a check valve is one of the most difficult valves to make and keep tight. The higher the pressure differential, the tighter the valve will be. These points are important and should be kept in mind when evaluating check valves in terms of service.

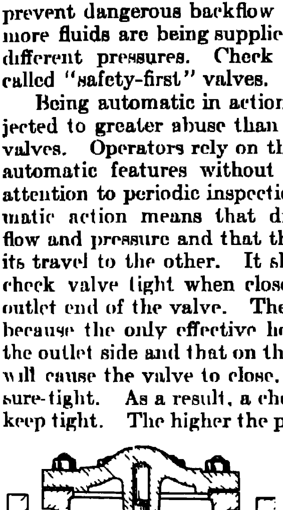
Fig. 11-18. Lift check valve.

Swing Check Valves. In the swing check design, flow through the body is very similar to that of a gate valve. The disc seats against a bridge wall (across the line of flow) and is hinged to a pin placed crosswise of the body above the bridge wall. The disc is free to swing through an arc between the bridge wall and the body or cap and far enough from the bridge wall to permit full flow through the valve when the disc is in wide-open position. The simple explanation is that the disc is held off its seat by flow in one direction and is returned to its seat by reverse flow or gravity (Fig. 11-17). The location of the bridge wall and the relationship in position between the hinge pin and the seat is such that, with no pressure or flow through the valve, the disc is held to its seat by gravity.

Lift Check Valves. The lift check (horizontal or globe type, Fig. 11-18) is designed so that flow through the valve lifts the disc away from its seat, and the force of gravity or back pressure holds the disc to its seat when closed. In this design it is common practice to use bodies similar to the regular globe or angle valves. In lift check valves having a metal disc, the disc is usually renewable and regrindable; in the composition-disc type; the disc is, of course, renewable. To ensure proper seating, the disc must be accurately guided.

FITTINGS

Fittings comprise a considerable part of any piping system and, in addition, constitute a good portion of the aggregate over-all cost. Further, and probably most important, is the fact that pipe itself is usually trouble-free if proper materials for the service have been selected, while joints are likely to be responsible for any unlooked-for maintenance costs which may occur. The working pressure and the nature of the service for which fittings are to be used have an important bearing on their design, and the wide variations in service conditions have necessitated the development of many different types and patterns as well as the use of different metals and alloys.

Groups of Fittings

The seven groups into which fittings (joints) generally fall are as follows:

1. **Screwed Ends.** Screwed ends are used more often owing to the fact that they make a substantially permanent connection—a very satisfactory metal-to-metal joint—are adequate for reasonable pressures and temperatures where piping is used, and, last but not least, are economical to use. It is well to note that they are most popular in the smaller sizes because the larger the pipe, the more difficult it is to make up a screwed joint.

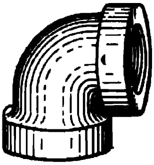

Fig. 11-19. Screwed-end elbow.

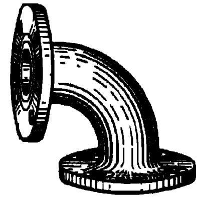

Fig. 11-20. Flanged-end elbow.

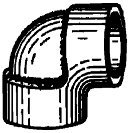

Fig. 11-21. Socket-welding elbow.

2. **Flanged Ends.** Flanged ends are used where conditions make screwed ends impractical and where bolting up the joint is advantageous. Although some flanged ends are made in sizes as small as ½ in., they are most generally used for lines 4 in. and larger because of their easier assembly and takedown. Further, flanged joints are made up with much smaller tools than are required for screwed connections of comparable size.

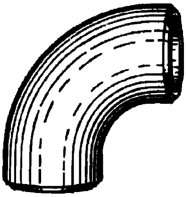

Fig. 11-22. Butt-welding elbow.

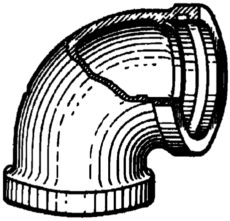

Fig. 11-23. Elbow with ends for brazing.

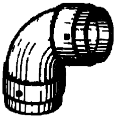

Fig. 11-24. Elbow with solder-joint ends.

3. **Welding Ends.** Welding ends are generally grouped under two headings: "socket-welding," primarily used in connection with small diameter piping; and "butt welding," for general all-around use and for any pressure or temperature to which the piping may be subjected. Also, welding ends are generally used on lines not requiring frequent dismantling.

4. **Brazed Ends.** Brazed ends are made with a brazing ring insert, having end joints similar to a socket weld or solder fitting as described herein. These ends are satisfactory for many installations such as steam and water lines, in process industries, refrigeration, etc. Like the solder end, they are often preferred as a means of obtaining a more permanently tight joint than is possible with a screwed end.

5. **Solder-joint Ends.** Solder-joint ends have been perfected for the purpose of forming a smooth, durable, and pressure-tight union between copper tubing and the

fitting. Fittings of this type are very often used in plumbing and heating lines, lubricating oil lines around machines, for services such as water, air, vacuum, brewery, or distillery liquors, and other nonhazardous and noncorrosive systems.

6. Hub and Spigot Ends. Hub and spigot ends are generally limited to valves and fittings specified for use with gas and water mains, sewage piping, or soil pipe, generally. The joint is assembled on the socket principle, with pipe inserted into hub end to valve or fitting, then caulked with oakum and sealed with molten lead. Hub-end fittings are usually made of cast iron and used with cast-iron pipe having diameters that differ from steel pipe. Variations of this joint for use with wrought-iron or steel pipe are known as the Matheson, Converse, etc., joint.

7. Flared Ends. Flared-end fittings are commonly used for copper and plastic tubing up to 2 in. in diameter. The end of the tubing is flared or skirted and a ring nut is used to make a union-type joint.

Common Types of Fittings

There are many patterns of fittings, each of which is designed for a particular and specific purpose, and the complete assortment is sufficient to care for any known joining problem. Pipe fitters can make up any simple or complex job with a minimum number of fittings and in strict accordance with specifications and codes.

Fittings are made in the same variety of metals as valves (brass, iron, steel, various alloys) and in corresponding pressure classes and end connections and in all standard pipe sizes. All types of fittings, however, are not made in all sizes and materials. Even the regular range of fittings includes a great multiplicity of items, and to show them all here would be practically impossible. An acquaintance with the most common types as illustrated and a general knowledge of the typical variations is advantageous. Generally speaking, these patterns are as follows:

Flanges
Elbows
Tees
Crosses
Reducers
Bushings
Couplings
Nipples
Unions
Return bends
Y bends or laterals
Caps and plugs

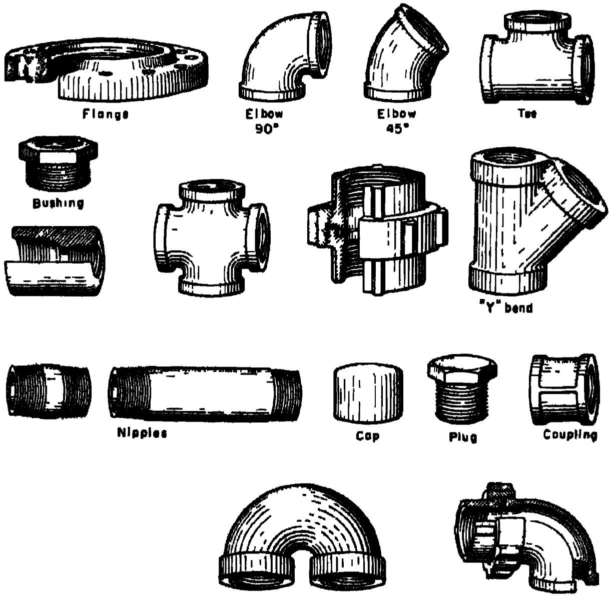

Fig. 11-25. Common types of fittings.

INDUSTRIAL PIPE INSULATION

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HIGH- AND MEDIUM-TEMPERATURE PIPE INSULATION

Proper thermal insulation is required on heated piping and related equipment, principally for the economy effected by the heat or fuel savings and the resultant reduction in production costs. However, it is frequently needed for other important purposes either alone or in conjunction with fuel savings. For example, insulation is used to ensure a minimum temperature drop in liquids and gases conveyed through pipes, ducts, and flues; to keep heat away from areas where it is not wanted in order to produce more efficient or comfortable working conditions; to protect against the hazards of fire or injury to personnel from contact with heated surfaces; to reduce strain on heated equipment by too rapid cooling, as in the case of exposed high-temperature pipe flanges and boiler brick settings; to maintain uniform temperatures in equipment in order to effect a desired chemical reaction; to prevent condensation of gases, *e.g.*, in maintaining the superheat in a steam line supplying a turbine or in flues to prevent corrosion of the metal flue; and for many other purposes.

Uninsulated Heated Surface Losses. Engineers generally recognize that it is good economy to insulate heated pipes, but other surfaces such as flanges and fittings often go unnoticed as they do not realize the magnitude of these losses.

That the area of these surfaces can total to a large quantity is readily seen from Table 11-7 which shows the areas of bare fittings and flanges. A single pair of 6-in. standard flanges has an area of 1.8 sq ft and is equivalent to a linear foot of 6-in. pipe. The loss of heat is not visible as in the case of steam leaks which would quickly be remedied by tightening and packing. The heat losses are nevertheless just as real and continuous.

When the magnitude of these losses is known, the need for a substantial saving of this wasted heat is readily understood. We all have difficulty in visualizing heat units (Btu), but when we talk of their equivalent actual cost in dollars or in pounds of coal, the extent of the losses is readily apparent. Figure 11-26 shows the equivalent losses in dollars per square foot per year (365 days of 24 hr) from bare heated surfaces. This is readily applied to different kinds of fuel of different costs and heating values burned at different efficiencies, as determined from Table 11-8 (see page 750). In considering the cost of heat, the cost delivered at the apparatus should be used. This includes not only the cost of fuel but may also include handling costs, depreciation, and other fixed charges on heat-generating and -transmitting equipment such as boilers, etc. Table 11-9 gives an example of the magnitude of these same losses in equivalent pounds of coal for a given set of operating conditions.

Table 11-9 shows the amount of heat energy lost, and coal consequently wasted, by the dissipation of heat through uninsulated surfaces. The calculations are based upon an available heat value per pound of coal of 10,000 Btu, which is equivalent to a boiler efficiency of 70 per cent using coal with a calorific value of approximately 14,000 Btu per lb. The figures are based also on continuous service, 24 hr per day, 365 days per year, with the average temperature of surrounding air at 70°F. A lesser boiler efficiency or inferior grade of coal would show even greater waste of fuel.

Monetary Value of Insulation. In referring to Fig. 11-26, it will be seen that bare-surface losses increase rapidly with increased temperature of the surface; hence the need for insulation is even more important the higher the temperature. A

Table 11-7. Areas of Flanged Fittings and Equivalent Pipe Lengths

Figures in columns under Area give surface areas in square feet.

Figures in columns under Pipe lengths give the number of feet of pipe which has an area equivalent to the area of the fitting.*

Nominal pipe size ID, in.	Standard flanged fittings, including accompanying flanges									
	Flanged coupling		90-deg ell		Long-radius ell		Tee		Cross	
	Area	Pipe lengths	Area	Pipe lengths	Area	Pipe lengths	Area	Pipe lengths	Area	Pipe lengths
1	0.320	0.93	0.795	2.31	0.892	2.59	1.235	3.59	1.622	4.72
1¼	0.383	0.88	0.957	2.20	1.084	2.49	1.481	3.40	1.943	4.47
1½	0.477	0.95	1.174	2.35	1.337	2.68	1.815	3.64	2.38	4.78
2	0.672	1.08	1.65	2.65	1.84	2.96	2.54	4.08	3.32	5.34
2½	0.841	1.12	2.09	2.78	2.32	3.08	3.21	4.26	4.19	5.56
3	0.945	1.03	2.38	2.60	2.68	2.93	3.66	3.99	4.77	5.70
3½	1.122	1.07	2.98	2.85	3.28	3.13	4.48	4.28	5.83	5.56
4	1.344	1.14	3.53	2.90	3.96	3.36	5.41	4.59	7.03	5.97
4½	1.474	1.13	3.95	3.01	4.43	3.38	6.07	4.63	7.87	6.01
5	1.622	1.11	4.44	3.049	5.00	3.43	6.81	4.67	8.82	6.06
6	1.82	1.049	5.13	2.95	5.99	3.45	7.84	4.53	10.08	5.81
7	2.17	1.097	6.17	3.09	7.38	3.697	9.37	4.69	12.00	6.01
8	2.41	1.067	6.98	3.09	8.56	3.79	10.55	4.67	13.44	5.96
9	3.00	1.19	8.71	3.457	10.57	4.20	13.18	5.23	16.78	6.66
10	3.43	1.22	10.18	3.61	12.35	4.38	15.41	4.47	19.58	6.95
12	4.41	1.32	13.08	3.92	16.35	4.90	19.67	5.89	24.87	7.45
14 OD	5.39	1.465	16.38	4.47	20.17	5.47	24.81	6.78	31.48	8.60
15 OD	6.18	1.572	18.50	4.72	22.92	5.83	27.91	7.10	35.48	9.04
16 OD	6.69	1.60	20.17	4.82	25.41	6.07	30.32	7.23	38.34	9.15

Nominal pipe size ID, in	Extra-heavy flanged fittings, including accompanying flanges									
	Flanged coupling		90-deg ell		Long-radius ell		Tee		Cross	
	Area	Pipe lengths	Area	Pipe lengths	Area	Pipe lengths	Area	Pipe lengths	Area	Pipe lengths
1	0.438	1.273	1.015	2.95	1.083	3.148	1.575	4.578	2.07	6.02
1¼	0.510	1.172	1.098	2.524	1.340	3.08	1.925	4.425	2.53	5.816
1½	0.727	1.459	1.332	2.674	1.874	3.762	2.68	5.381	3.54	7.108
2	0.848	1.363	2.01	3.23	2.16	3.473	3.09	4.968	4.06	6.528
2½	1.107	1.463	2.57	3.41	2.76	3.665	4.05	5.378	5.17	6.865
3	1.484	1.619	3.19	3.807	3.74	4.08	5.33	5.815	6.95	7.582
3½	1.644	1.57	3.96	3.782	4.28	4.087	6.04	5.768	7.89	7.535
4	1.914	1.624	4.64	3.938	4.99	4.236	7.07	6.001	9.24	7.843
4½	2.04	1.558	5.02	3.834	5.46	4.170	7.72	5.897	10.07	7.692
5	2.18	1.497	5.47	3.756	6.02	4.134	8.52	5.851	10.97	7.534
6	2.78	1.603	6.99	4.031	7.76	4.475	10.64	6.136	13.75	7.929
7	3.46	1.733	8.62	4.318	9.73	4.874	12.33	6.177	16.83	8.431
8	3.77	1.670	9.76	4.324	11.09	4.913	14.74	6.531	18.97	8.405
9	4.44	1.762	11.44	4.541	13.17	5.228	17.23	6.84	22.10	8.773
10	5.20	1.846	13.58	4.82	15.60	5.538	20.41	7.245	26.26	9.322
12	6.71	2.01	17.73	5.31	18.76	5.622	26.65	7.987	34.11	10.222
14 OD	8.30	2.26	22.31	6.08	25.70	7.02	33.63	9.18	43.15	11.75
15 OD	9.52	2.43	25.28	6.43	29.34	7.47	38.04	9.68	48.79	12.4
16 OD	10.05	2.4	27.18	6.475	31.73	7.575	40.94	9.775	52.35	12.5

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* Equivalent pipe lengths give a ready method of surveying a plant for heat losses by adding the fittings in pipe lengths to other bare pipe in the same run.

Table 11-8. Heat Cost of Fuels per Million Btu at Various Thermal Efficiencies

Fuels	Btu content	Cost/std unit	Thermal efficiency									
			100%	90%	80%	70%	60%	50%	40%	30%	20%	10%
Coal	13,500/lb	$6 00/ton	$ 222	$ 247	$ 278	$ 318	$ 370	$ 444	$ 555	$ 740	$1 11	$2 22
		5 00	185	206	232	265	309	370	462	617	925	1 85
		4 00	148	165	185	212	247	296	370	444	740	1 48
		3 00	111	124	139	159	185	222	278	370	556	1 11
		2 00	074	082	092	106	123	148	185	247	370	74
Fuel oil	150,000/gal	0 07/gal	466	.519	583	666	778	933	1 17	1 55	2 33	4 66
		0 06	400	445	500	572	666	800	1 00	1 33	2 00	4 00
		0 05	333	370	417	476	555	666	836	1 11	1 66	3 33
		0 04	267	297	334	381	445	533	666	89	1 33	2 67
		0 03	200	222	250	286	333	400	50	666	1 00	2 00
Coal tar	150,000/gal	0 05/gal	333	370	417	476	555	666	834	1 11	1 66	3 33
		0 04	267	207	334	381	445	533	666	89	1 33	2 67
		0 03	200	222	250	286	333	400	500	666	1 00	2 00
Natural gas	1 000/cu ft	0 70/Mcf	700	778	875	1 00	1 16	1 40	1 75	2 33	3 50	7 00
		0 60	600	667	750	857	1 00	1 20	1 50	2 00	3 00	6 00
		0 50	500	555	625	715	834	1 00	1 25	1 67	2 50	5 00
		0 40	400	445	500	572	667	80	1 00	1 33	2 00	4 00
		0 30	300	333	375	428	50	60	75	1 00	1 50	3 00
Artificial gas	500/cu ft	0 80 Mcf	1 60	1 78	2 00	2 29	2 67	3 20	4 00	5 33	8 00	16 00
		0 70	1 40	1 56	1 75	2 00	2 33	2 80	3 50	4 66	7 00	14 00
		0 60	1 20	1 33	1 50	1 71	2 00	2 40	3 00	1 00	6 00	12 00
		0 50	1 00	1 11	1 25	1 43	1 67	2 00	2 50	3 33	5 00	10 00
		0 40	80	89	1 00	1 14	1 33	1 60	2 00	2 67	4 00	8 00
Coke-oven gas	450/cu ft	0 10/Mcf	222	247	278	318	370	444	555	740	1 11	2 22
		0 09	200	222	250	286	333	400	500	666	1 00	2 00
		0 08	178	198	222	254	296	356	445	593	89	1 78
		0 07	156	173	195	222	259	311	389	518	777	1 56
		0 06	133	148	167	191	222	267	334	445	666	1 33
Producer gas	130 cu ft	0 06 Mcf	462	513	577	660	770	924	1 15	1 54	2 31	1 62
		0 05	385	427	481	550	641	770	962	1 28	1 92	3 85
		0 01	308	342	384	440	512	615	769	1 02	1 54	3 08
		0 03	231	257	289	330	385	462	577	77	1 15	2 31
		0 02	154	171	192	220	256	308	485	513	77	1 54
Blast-furnace gas	90/cu ft	0 05/Mcf	555	.619	695	794	925	1 11	1 39	1 85	2 78	5 55
		0 04	445	495	556	635	741	89	1 11	1 48	2 22	4 15
		0 03	334	370	417	476	555	666	833	1 11	1 66	3 34
		0 02	222	247	278	318	370	444	555	74	1 11	2 22
		0 01	111	124	139	159	185	222	278	37	555	1 11
Electricity	3,415/kwhr	0 05/kwhr	14 65									
		0 04	11 72									
		0 03	8 79									
		0 02	5 86									
		0 015	4 39									
		0 01	2 93									
		0 0075	2 20									

specific example is a boiler drum operating 7,200 hr per year at approximately 150 psi gauge pressure (366°F) with boiler-room air temperature of 75°F, or a temperature difference between boiler and air of 291°F and heat cost of 30 cents per million available Btu. The loss per square foot per year is

$$\$2.46 \times [7,200/(365 \times 24)] = \$2.02$$

Proper type of insulation for this service, saving over 90 per cent of this loss, can be applied at far less than half of 1 year's saving. This shows that insulation is an

Table 11-9. Example of Coal Waste Due to Heat Loss from Uninsulated Surfaces

Steam pressure psig	Temperature, deg F	Temperature difference, deg F	Btu loss /sq ft/hr	Coal wasted, lb/sq ft/year	Area, sq ft wasting 1 ton coal in 1 year
	100	30	56 6	49 6	40 3
	120	50	97 5	85 4	23 4
	140	70	142	124 3	16 1
	160	90	190	166 3	12 03
	180	110	242	212	9 44
	200	130	298 5	261 5	7 65
0	212	142	334	293	6 82
10	240	170	425	372	5 38
25	267	197	522 5	458	4 37
50	298	228	644	564	3 55
75	320	250	737 5	646	3 10
100	338	268	820	718	2 79
150	366	296	960	840	2 38
200	388	318	1 079	945	2 12
250	406	336	1 184	1 036	1 93

excellent investment and illustrates the need for insulating such surfaces as boiler heads, flanges, and fittings, which are often left bare even when piping is adequately insulated

Insulation is paid for whether it is bought or not because the cost of the heat loss without insulation in 1 year is generally considerably greater than the insulation cost.

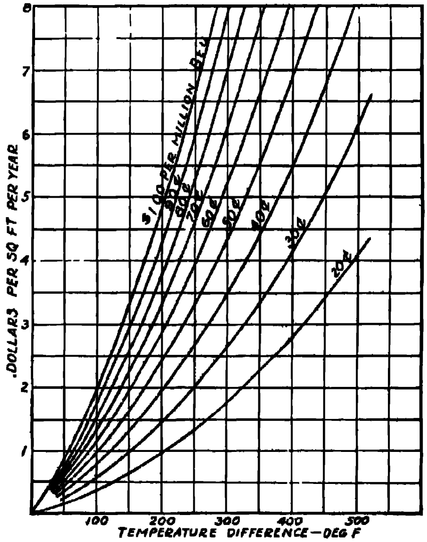

FIG. 11-26. Cost of heat losses per year at various values of heat.

Moreover, this heat-loss cost goes on year after year, so in effect the equivalent cost of insulation is paid out many times.

Theory of Losses from Heated Surfaces. Before discussing savings further, consider the source of the data on bare losses from heated surfaces. The loss from any heated surface exposed to air is made up of two components, one designated "radiation" and the other "convection." The radiation component has been expressed by Langmuir,[1] according to Stefan-Boltzmann in Eq. (1),

$$q_r = 0.174\epsilon[(T_1/100)^4 - (T_2/100)^4] \tag{1}$$

where q_r = rate of heat transfer from surface by radiation, Btu/(hr)/(sq ft)
ϵ = emissivity coefficient
T_1 = absolute temperature of hot surface, deg F
T_2 = absolute temperature of surrounding objects, deg F

showing that the loss is dependent principally upon the difference in the fourth powers of the absolute temperatures of the radiating surface and of the surroundings and also upon the nature of the surface and surroundings, whether "bright," "dull" (in reference to heat rays, not to light rays), etc., and to some small extent upon the geometrical arrangement. It should be understood that the radiation portion of the loss is the only part that is affected by "brightness" of the surface. It has been shown by test that, other than polished metal surfaces, all ordinary materials such as iron and steel, brick, wood, concrete, plaster, as well as nonmetallic paints irrespective of their color, have practically the same radiation loss.

Copper, brass, and aluminum have been shown to have lower radiation loss when new and brightly polished; but these losses increase in service owing to oxidation and by soiling from oil, grease, dust, and dirt. Therefore, it is an unnecessary refinement, not warranted in practice, to differentiate between the types of metal used, particularly as ferrous pipe is by far the most generally used for heated piping. However, it can be shown that, even when applied on new commercially finished copper and brass pipe, insulation pays a satisfactory return on its investment.[2]

The loss by convection, according to Eq. (2) developed by Langmuir,[1] for still air

$$q_c = 0.296(T_1 - T_a)^{5/4} \tag{2}$$

where q_c = rate of heat transfer, from surface by convection, Btu/(hr)/(sq ft)
T_a = absolute temperature of air, deg F

under average conditions, is dependent principally upon the five-fourths power of the difference in the absolute temperatures of the heated surface and the air. It is also known that the shape and the position of the surface with respect to the natural air currents set up by the convection loss have some effect and that curved surfaces of small diameter lose heat at a greater unit rate than large diameters. It should be noted here that the term "still air" is used to denote only the condition where natural air movement due to the heat dissipated from the pipe exists, and no forced or artificial circulation is present. However, the greatest affect on convection is that due to currents such as would be occasioned by forced air circulation indoors or wind blowing on outdoor equipment. Equation (3) (Langmuir's equation[1]) shows that the increase in convection by air circulation is dependent upon the velocity.

$$q_{cv} = q_c \sqrt{(v + 68.9)/68.9} \tag{3}$$

where q_{cv} = rate of heat transfer from surface by convection at any velocity, Btu/(hr)/(sq ft)
v = air velocity, fpm

The total heat transfer from any surface can thus be calculated by adding the radiation loss to the convection loss for the given air velocity.

How Test Data on Bare-surface Losses and Theory Agree. A number of tests have been made on heated surfaces, mainly on pipes, some by accurate electrical measurement of heat loss. These test results are shown in Fig. 11-27 compared with values calculated for combined radiation and convection under still-air conditions using Langmuir's equations (Eqs. (1) and (2).

It will be noted that in Fig. 11-27 there is little variation due to pipe size except for the very small diameters and that the McMillan results are representative of all tests. For this reason all bare-surface losses in still air used in this subsection (Table 11-10) have been based on these test results together with subsequent tests in the McMillan laboratories extending the data to 1000°F.

Test results on surfaces subjected to various air velocities have not been as numerous as still-air tests, probably owing to the larger number of variables which would have to be studied. Tests by Taylor[3] and Fulton and Parlett[4] have shown large increases in heat loss with small increase in air velocity and values even larger than those shown by Langmuir's equation. Langmuir's data have been generally used because of their completeness. Figure 11-28, based on these data, shows how the total loss increases compared with still-air conditions. It shows that, for an air velocity of only 10 mph (less than the average yearly wind velocity in New York City), the loss is 140 to 210 per cent of that under still-air conditions for the temperature range shown. Since Langmuir's figures are conservative, we can readily see the added importance of insulation, as many surfaces in the plant are exposed to circulating air.

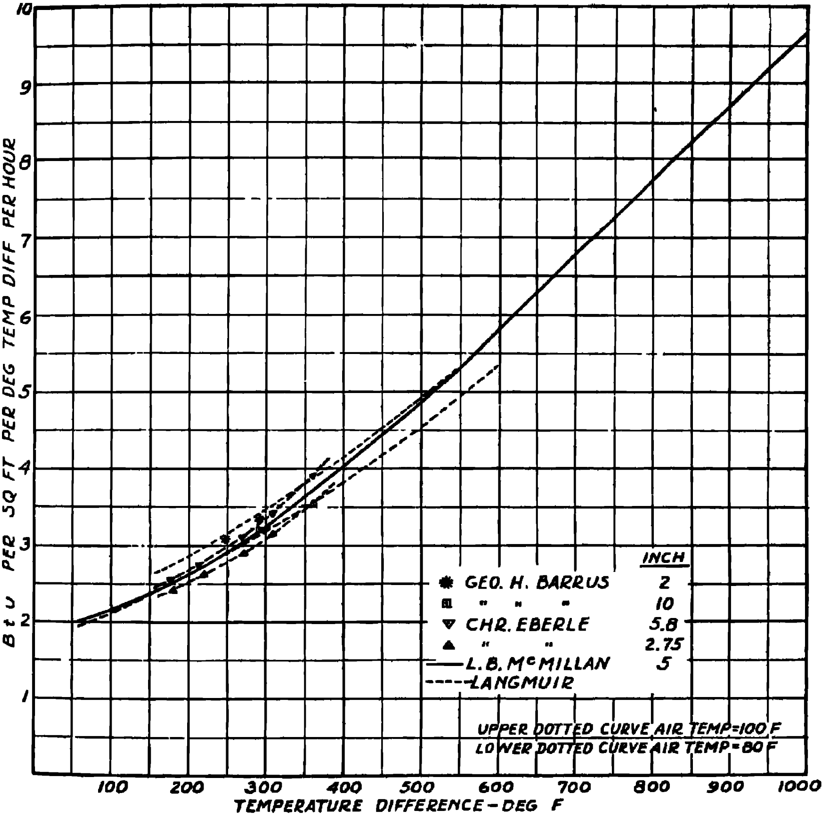

FIG. 11-27. Rate of heat transfer from metal surface to air. (*Barrus, Eberle, McMillan, and Langmuir.*)

Even at a velocity of only a few hundred feet a minute, the increase in loss may be as much as 50 per cent. Most indoor heated surfaces are subject to circulating air a greater portion of the time, and on outdoor apparatus and piping the effect is even greater.

Savings with Insulation. The large savings or reduction in heat losses from bare surfaces that can be made by proper insulation are definitely known from many tests. All standard insulating materials have been tested in some laboratory, and some manufacturers regularly and periodically test their products in their own laboratories to determine insulating value and to ensure uniformity in the product. These tests are run on electrically heated pipes and plates where the heat loss can be measured electrically within an accuracy of 1 per cent.(5, 6)

Pipe and insulation outer surface temperatures are accurately measured with thermocouples; and the tests are made in a closed room under still-air conditions with accurate air-temperature control.

The "conductivity" of an insulation is the *rate* of heat transfer through a homogeneous material in one direction (perpendicular to an area) per unit time per unit area per unit temperature gradient and is expressed as k = Btu/(hr)/(sq ft)/(deg F temperature difference between surfaces per 1 in. thick).

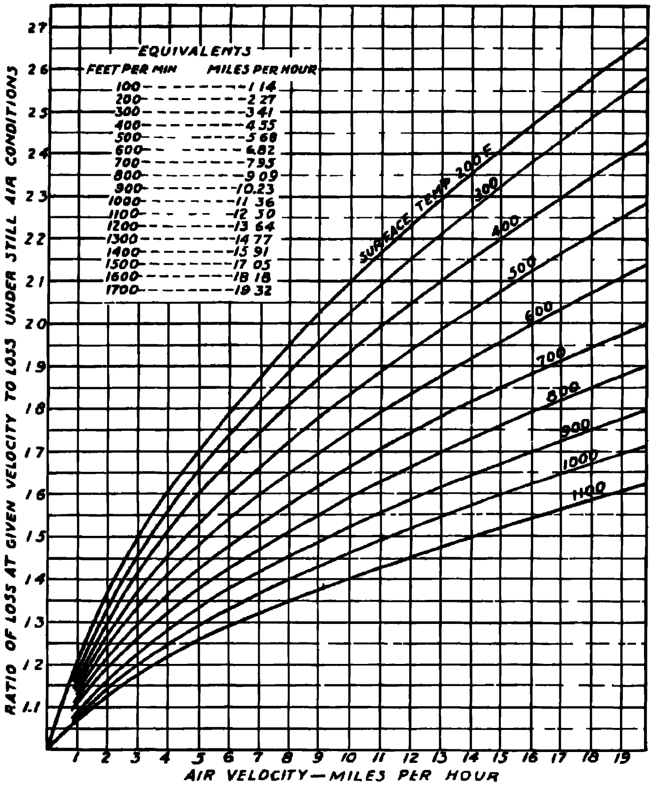

FIG. 11-28. Effect of air velocity on bare surface loss (temperature of air, 80°F).

The conductivity can be calculated from the measured Btu loss from the pipe, using the well-known rational equation for flow of heat, Eq. (4).

$$C_2 = \frac{(t_1 - t_2)}{[r_2 \log_e (r_2/r_1)]/k} \tag{4}$$

where C_2 = rate of heat transfer per sq ft of outer surface of insulation, Btu/(hr)(sq ft)
t_1 = temperature of warmer surface of insulation, deg F
t_2 = temperature of cooler surface of insulation, deg F
r_2 = outer radius of insulation, in.
r_1 = inner radius of insulation, in.
k = conductivity, Btu/(hr)/(sq ft)/(deg F temp diff between surfaces per 1 in. thick)

Table 11-10. Heat Loss from Bare Surfaces

Losses given in Btu per hour per linear foot of bare pipe at various temperature differences (pipe-air). (For finding losses at temperatures between those shown, the Btu increments per degree are given in lightface type between the main columns.)

Nominal pipe size, in.	Sq ft of pipe surface/lin ft	Loss, Btu/(hr)/(lin ft of bare pipe)										
		50°F	Btu increment /degree	100°F	Btu increment /degree	150°F	Btu increment /degree	200°F	Btu increment /degree	250°F	Btu increment /degree	300°F	Btu increment /degree
½	0.220	21.5	0.52	47.3	0.64	79.2	0.76	117.3	0.90	162.3	1.06	215.2	1.28
¾	0.275	26.8	0.65	59.2	0.80	99.0	0.95	146.6	1.13	202.9	1.32	269.0	1.60
1	0.344	33.6	0.81	74.0	1.00	123.8	1.19	183.4	1.41	253.8	1.65	336.4	2.01
1¼	0.435	42.4	1.02	93.6	1.26	156.6	1.51	231.9	1.78	320.9	2.09	425.4	2.54
1½	0.498	48.6	1.17	107.2	1.44	179.3	1.72	265.4	2.04	367.4	2.39	487.0	2.90
2	0.622	60.7	1.46	133.9	1.80	223.9	2.15	331.5	2.55*	458.9	2.99	608.3	3.63
2½	0.753	73.4	1.77	162.1	2.18	271.1	2.61	401.4	3.08	555.6	3.62	736.4	4.39
3	0.917	89.4	2.16	197.3	2.66	330.1	3.17	488.8	3.76	676.6	4.40	896.5	5.34
3½	1.047	102.1	2.46	225.3	3.03	376.9	3.62	558.1	4.29	772.5	5.03	1024.0	6.10
4	1.178	114.9	2.77	253.6	3.41	424.1	4.08	627.9	4.82	869.1	5.66	1152.0	6.88
4½	1.309	127.6	3.08	281.7	3.79	471.2	4.53	697.7	5.35	965.1	6.30	1280.0	7.64
5	1.456	142.0	3.43	313.3	4.22	524.2	5.04	776.1	5.96	1074.0	7.00	1424.0	8.48
6	1.734	169.1	4.08	373.2	5.02	624.2	6.00	924.2	7.10	1279.0	8.34	1696.0	10.10
7	1.996	194.6	4.70	429.5	5.78	718.6	6.91	1064.0	8.18	1473.0	9.58	1952.0	11.64
8	2.257	220.1	5.31	485.7	6.54	812.5	7.81	1203.0	9.24	1665.0	10.84	2207.0	13.16
9	2.519	245.6	5.93	542.1	7.29	906.5	8.72	1343.0	10.32	1859.0	12.10	2464.0	14.68
10	2.817	274.7	6.63	606.2	8.16	1014.0	9.76	1502.0	11.52	2078.0	13.54	2755.0	16.42
11	3.073	299.6	7.23	661.3	8.89	1106.0	10.64	1638.0	12.58	2267.0	14.76	3005.0	17.92
12	3.338	325.5	7.86	718.3	9.67	1202.0	11.54	1779.0	13.68	2463.0	16.04	3265.0	19.46
14 OD	3.665	357.1	8.62	788.3	10.61	1319.0	12.66	1952.0	15.02	2703.0	17.58	3582.0	21.36
16 OD	4.188	408.3	9.86	901.3	12.13	1508.0	14.48	2232.0	17.16	3090.0	20.12	4096.0	24.42
18 OD	4.716	459.8	11.10	1015.0	13.66	1698.0	16.32	2514.0	19.32	3480.0	22.64	4612.0	27.50
20 OD	5.236	510.4	12.33	1127.0	15.16	1885.0	18.10	2790.0	21.44	3862.0	25.16	5120.0	30.52
24 OD	6.286	612.9	14.80	1353.0	18.20	2263.0	21.74	3350.0	25.76	4638.0	30.20	6148.0	36.64
30 OD	7.854	765.8	18.48	1690.0	22.74	2827.0	27.18	4186.0	32.18	5795.0	37.72	7681.0	45.80

* *Example:* 2-in pipe, 235° temperature difference; 235° − 200° = 35°; 35° × 2.55 (Btu per deg) = 89.3 Btu; 331.5 + 89.3 = 420.8 Btu loss at 235° temperature difference.

Table 11-10. Heat Loss from Bare Surfaces (*Cont.*)

Nominal pipe size, in.	Sq ft of pipe surface/lin ft	Loss, Btu/(hr)/(lin ft of bare pipe)											
		350°F	Btu increment/degree	400°F	Btu increment/degree	450°F	Btu increment/degree	500°F	Btu increment/degree	550°F	Btu increment/degree	600°F	Btu increment/degree
½	0.220	279.3	1.52	355.1	1.73	441.7	1.99	541.2	2.17	649.6	2.45	772.2	2.63
¾	0.275	349.1	1.90	443.9	2.16	552.1	2.49	676.5	2.71	812.2	3.06	965.3	3.29
1	0.344	436.7	2.37	555.2	2.71	690.6	3.11	846.2	3.40	1016.0	3.82	1207.0	4.12
1¼	0.435	552.2	3.00	702.1	3.42	873.3	3.93	1070.0	4.30	1285.0	4.84	1527.0	5.20
1½	0.498	632.2	3.43	803.6	3.92	999.7	4.51	1225.0	4.92	1471.0	5.54	1748.0	5.96
2	0.622	789.6	4.29	1004.0	4.90	1249.0	5.62	1530.0	6.14	1837.0	6.92	2183.0	7.44
2½	0.753	955.9	5.18	1215.0	5.94	1512.0	6.80	1852.0	7.44	2224.0	8.38	2643.0	9.00
3	0.917	1164.0	6.32	1480.0	7.22	1841.0	8.30	2256.0	9.04	2708.0	10.22	3219.0	10.96
3½	1.047	1329.0	7.22	1690.0	8.24	2102.0	9.48	2576.0	10.32	3092.0	11.66	3675.0	12.52
4	1.178	1496.0	8.10	1901.0	9.28	2365.0	10.66	2898.0	11.62	3479.0	13.12	4135.0	14.08
4½	1.309	1662.0	9.02	2113.0	10.30	2628.0	11.84	3220.0	12.92	3866.0	14.58	4595.0	15.64
5	1.456	1848.0	10.04	2350.0	11.46	2923.0	13.18	3582.0	14.36	4300.0	16.22	5111.0	17.40
6	1.734	2201.0	11.96	2799.0	13.64	3481.0	15.70	4266.0	17.10	5121.0	19.30	6086.0	20.74
7	1.996	2534.0	13.76	3222.0	15.70	4007.0	18.06	4910.0	19.68	5894.0	22.24	7006.0	23.88
8	2.257	2865.0	15.56	3643.0	17.76	4531.0	20.42	5552.0	22.28	6666.0	25.12	7922.0	27.00
9	2.519	3198.0	17.36	4066.0	19.82	5057.0	22.80	6197.0	24.86	7440.0	28.04	8842.0	30.20
10	2.817	3576.0	19.42	4547.0	22.16	5655.0	25.50	6930.0	27.80	8320.0	31.36	9888.0	33.60
11	3.073	3901.0	21.18	4960.0	24.18	6169.0	27.82	7560.0	30.32	9076.0	34.30	10790.0	36.60
12	3.338	4238.0	23.00	5388.0	26.26	6701.0	30.22	8212.0	32.94	9859.0	37.20	11720.0	39.80
14 OD	3.663	4650.0	25.24	5912.0	28.84	7354.0	33.14	9011.0	36.20	10820.0	40.80	12860.0	43.80
16 OD	4.188	5317.0	28.84	6759.0	32.96	8407.0	37.90	10300.0	41.40	12370.0	46.60	14700.0	50.00
18 OD	4.716	5987.0	32.50	7612.0	37.10	9467.0	40.70	11600.0	46.60	13930.0	52.40	16550.0	56.40
20 OD	5.235	6646.0	36.06	8449.0	41.20	10510.0	47.40	12880.0	51.60	15460.0	58.40	18380.0	62.60
24 OD	6.286	7980.0	43.40	10150.0	49.40	12620.0	56.80	15460.0	62.20	18570.0	69.80	22060.0	75.20
30 OD	7.854	9971.0	54.18	12680.0	61.80	15770.0	71.00	19320.0	77.60	23200.0	87.40	27570.0	93.80

Table 11-10. Heat Loss from Bare Surfaces (*Cont.*)

Nominal pipe size, in	Sq ft of pipe surface/lin ft	Loss, Btu (hr)/(lin ft of bare pipe) 650°F	Btu increment degree	700°F	Btu increment degree	750°F	Btu increment /degree	800°F	Btu increment /degree	850°F	Btu increment /degree	900°F
½	0.220	903.8	2 86	1047	3 06	1200	3 28	1364	3 48	1538	3 70	1723
¾	0.275	1130.0	3 58	1309	3 84	1501	4 08	1705	4 36	1923	4 60	2153
1	0.344	1413.0	4 48	1637	4 80	1877	5 12	2133	5 44	2405	5 78	2694
1¼	0.435	1787.0	5 68	2071	6 06	2374	6 46	2697	6 88	3041	7 30	3406
1½	0.498	2046.0	6 50	2371	6 92	2717	7 42	3088	7 88	3482	8 34	3899
2	0.622	2555.0	8 12	2961	8 66	3394	9 24	3856	9 86	4349	10 42	4870
2½	0.753	3093.0	9 82	3584	10 50	4109	11 20	4669	11 90	5264	12 64	5896
3	0.917	3767.0	11 96	4365	12 76	5003	13 64	5685	14 52	6411	15 38	7180
3½	1.047	4301.0	13 66	4984	14 58	5713	15 56	6491	16 58	7320	17 56	8198
4	1.178	4839.0	15 36	5607	16 42	6428	17 52	7304	18 64	8236	19 76	9224
4½	1.309	5377.0	17 08	6231	18 22	7142	19 48	8116	20 72	9152	21 96	10250
5	1.456	5981.0	19 00	6931	20 26	7944	21 66	9027	23 10	10180	24 40	11400
6	1.734	7123.0	22 62	8254	24 14	9461	25 80	10750	27 40	12120	29 20	13580
7	1.996	8200.0	26 02	9501	27 80	10890	29 80	12380	31 60	13960	33 40	15630
8	2.257	9272.0	29 40	10740	31 60	12320	33 40	13990	35 80	15780	37 80	17670
9	2.519	10350.0	32 80	11990	35 00	13740	37 60	15620	39 80	17610	42 20	19720
10	2.817	11570.0	36 80	13410	39 20	15370	42 00	17470	44 40	19690	47 40	22060
11	3.072	12620.0	40 20	14630	42 80	16770	45 60	19050	48 60	21480	51 60	24060
12	3.338	13710.0	43.60	15890	46 40	18210	49 80	20700	52 80	23340	56 00	26140
14 OD	3.665	15050.0	47.80	17440	51 00	19990	54 40	22710	58 00	25610	61 40	28680
16 OD	4.188	17200.0	54.80	19940	58 20	22850	62 40	25970	66 20	29280	70 20	32790
18 OD	4.716	19370.0	61.60	22450	65 60	25730	70 20	29240	74 60	32970	79 20	36930
20 OD	5.236	21510.0	68.20	24920	72 80	28560	78 00	32460	82 80	36600	87 80	40990
24 OD	6.286	25820.0	82.00	29920	87 60	34300	93 40	38970	99 60	43950	105 40	49220
30 OD	7.854	32260.c	102.60	37390	109.20	42850	117.00	48700	124 20	54910	131 80	61500

Table 11-10. Heat Loss from Bare Surfaces (*Cont.*)

Nominal pipe size, in.	Sq ft of pipe surface /lin ft	Loss, Btu/(hr)/(lin ft of bare pipe)			
		Btu increment /degree	950°F	Btu increment /degree	1000°F
½	0.220	3.90	1918	4.10	2123
¾	0.275	4.88	2397	5.14	2654
1	0.344	6.08	2998	6.44	3320
1¼	0.435	7.72	3792	8.12	4198
1½	0.498	8.84	4341	9.30	4806
2	0.622	11.04	5422	11.60	6002
2½	0.753	13.34	6563	14.08	7267
3	0.917	16.26	7993	17.12	8849
3½	1.047	18.56	9126	19.50	10100
4	1.178	20.90	10270	22.00	11370
4½	1.309	23.20	11410	24.40	12630
5	1.456	25.80	12690	27.20	14050
6	1.734	30.60	15110	32.40	16730
7	1.996	35.40	17400	37.20	19260
8	2.257	40.00	19670	42.20	21780
9	2.519	44.80	21960	47.00	24310
10	2.817	49.80	24550	52.60	27180
11	3.073	54.60	26790	57.40	29660
12	3.338	59.20	29100	62.20	32210
14 OD	3.665	65.00	31930	68.40	35350
16 OD	4.188	74.20	36500	78.20	40410
18 OD	4.716	83.60	41110	88.00	45510
20 OD	5.235	92.80	45630	97.80	50520
24 OD	6.286	111.40	54790	117.40	60660
30 OD	7.854	139.20	68460	146.60	75790

Heat Loss from Flat, Curved, and Cylindrical Surfaces

Heat losses given in Btu per hour per square foot of bare surface at various temperature differences.

Temp difference, deg F	Total heat loss at temp indicated	Increments /degree for total losses	Loss/degree temp difference
50	97.5	2.35	1.950
100	215.2	2.90	2.152
150	360.0	3.46	2.400
200	533.0	4.10	2.665
250	737.8	4.80	2.951
300	978.0	5.83	3.260
350	1269.5	6.89	3.627
400	1614.0	7.87	4.035
450	2007.5	9.05	4.461
500	2460.0	9.87	4.920
550	2953.5	11.10	5.370
600	3510.0	11.96	5.850
650	4108.0	13.04	6.320
700	4760.0	13.93	6.800
750	5456.3	14.87	7.275
800	6200.0	15.83	7.750
850	6991.3	16.77	8.225
900	7830.0	17.73	8.700
950	8716.3	18.67	9.175
1000	9650.0		9.650

The conductivity, calculated from such tests can then be applied in the same type of equation [Eq. (5)] to determine the loss on a pipe or other curved surface of different

$$U_2 = (t_1 - t_a) \Big/ \left[\frac{r_2 \log_e (r_2/r_1)}{k} + \frac{1}{h}\right] \qquad (5)$$

where U_2 = over-all rate of heat transfer per sq ft of outer surface of insulation, Btu/(hr)/(sq ft)

t_a = temperature of air on cooler side, deg F
h = rate of heat transfer from outer surface of insulation to air, Btu/(hr)/(sq ft)/deg F

diameter, which makes it unnecessary to test insulation in every pipe size in which it is manufactured in order to determine the losses. The conductivity may also be used in the same type of rational heat-flow Eq. (6) for calculating the heat loss from a flat

$$U = \frac{(t_1 - t_a)}{(L/k) + (1/h)} \tag{6}$$

where U = over-all rate of heat transfer, Btu/(hr)/(sq ft)
L = thickness of insulation, in.

surface where the insulation would be applied in the form of flat blocks.

The heat losses determined in this manner are shown in Table 11-11 for 85% magnesia block, the universal standard insulation for temperatures up to 600°F. Table 11-11 applies only to blocks when used on flat surfaces or curved surfaces of very large diameter and not to pipes. Similar tables of heat losses for all pipe sizes and thicknesses of insulation are furnished by the insulation manufacturers for all types of insulation.

The losses from bare surfaces are also given, and the magnitude of the savings is shown. As an example, take the loss from the boiler drum, previously referred to, which operates at 150 psi (366°F). This is found to be 59.2 Btu/(sq ft)/(hr) when the drum is insulated with 2-in.-thick 85% magnesia. The loss from the same drum bare is 934.8/Btu/(sq ft)/(hr), or over 15 times as large. Hence a saving of 875 Btu/(sq ft)/(hr) is made which is over 14 times as much as the loss from the insulated pipe. The percentage of the bare loss that is saved is 93.6 per cent.

Table 11-11. Heat Losses and Savings, 85% Magnesia Blocks
[Insulation losses and savings, Btu/(hr)/(sq ft)]

Insulation thickness	Temperature of hot surface, deg F									
	125	175	225	275	325	375	425	475	525	575
	Temperature difference between hot surface and air, deg F									
	50	100	150	200	250	300	350	400	450	500
No insulation										
Bare-surface loss	97.5	215.2	360.0	533.0	737.8	978.0	1269.5	1614.0	2007.5	2460.0
1 in.										
Insulation loss	17.0	34.6	53.0	72.0	92.0	113.0	134.0	157.0	180.0	204.0
Saving..........	80.5	180.6	307.0	461.0	645.7	865.0	1135.5	1457.0	1827.5	2256.0
1½ in.										
Insulation loss..	12.1	24.6	37.7	51.2	65.3	79.8	94.9	110.0	126.0	143.0
Saving	85.4	190.6	322.3	481.8	672.5	898.2	1174.6	1504.0	1880.5	2317.0
2 in.										
Insulation loss.	9.35	19.0	29.1	39.6	50.5	61.8	73.5	85.6	98.1	111.0
Saving	88.1	196.2	330.9	493.4	687.3	916.2	1196.0	1528.4	1909.4	2349.0
2½ in.										
Insulation loss .	7.65	15.5	23.7	32.2	41.0	50.1	59.5	69.6	79.7	90.5
Saving.............	89.8	199.7	336.3	500.8	696.8	927.9	1210.0	1544.4	1927.8	2369.5
3 in.										
Insulation loss....	6.45	13.2	20.1	27.4	34.8	42.6	50.4	58.8	67.5	76.0
Saving	91.0	202.0	339.9	505.6	703.0	935.4	1219.1	1555.2	1940.0	2384.0
3½ in.										
Insulation loss......	5.00	11.4	17.4	23.6	30.0	36.6	43.4	50.4	58.1	66.0
Saving....	91.9	203.8	342.6	509.4	707.8	941.4	1226.1	1563.6	1949.4	2394.0
4 in.										
Insulation loss..	4.90	10.0	15.3	20.8	26.5	32.4	38.5	44.8	51.3	58.0
Saving	92.6	205.2	344.7	512.2	711.3	945.6	1231.0	1569.2	1956.2	2402.0

The Btu losses and savings under still-air conditions shown in Table 11-11 can be converted into equivalent pounds of steam or coal or other fuel, or dollars, by direct calculation. In order to determine the equivalent pounds of steam, it is necessary only to divide the Btu by the heat in a pound of steam above feed-water temperature. To convert Btu into equivalent pounds of coal or other fuel, divide by the Btu available per pound of fuel (Btu calorific value of fuel multiplied by heat-plant thermal efficiency). To convert Btu to equivalent dollars, the value of the heat per million available Btu must be known. As previously stated, this figure should be the delivered cost of the heat at the point where it is dissipated and may include not only the cost of fuel and labor of handling it but the fixed charges on the steam or other heat-generating and -transmitting equipment. The heat cost of the fuel alone can be determined directly from Table 11-8 when the thermal efficiency is known.

The magnitude of the losses and savings effected by proper insulation in a plant can be seen by referring again to a simple item such as the two heads with an area of 15 sq ft on a single boiler drum. The boiler operates at 150 psi for 7,200 hr per year, generating steam at a total cost equal to 30 cents per million available Btu, or approximately per 1,000 lb steam. The yearly bare loss from the two boiler drum heads amounts to 100,958 lb steam, or $30.28. The yearly savings when insulated with 2-in. 85% magnesia block are 94,414 lb steam, or $28.32. Other individual items similar to this one will be multiplied many times in some plants. Steam-generating or boiler-room equipment, such as boilers, turbines, pumps, heaters, and piping, is generally conceded to require proper insulation, although it is not always properly insulated. Besides these, there are savings to be effected throughout the plant by the use of proper insulation on such heated equipment as process tanks, heat exchangers, furnaces, waste-heat boilers, etc. Surveys made in each plant to determine the loss on all heated equipment generally show much equipment where the possibilities of savings have never before been explored.

The heat-loss table for 85% magnesia block is based on still-air conditions. It has already been shown that bare losses increase rapidly with increase in air velocity over the surface. For example, 40 to 110 per cent increase with a wind or air velocity of only 10 mph. The effect on well-insulated surfaces, however, is very small by comparison; the maximum increase due to a wind velocity of 30 mph being only about 15 per cent for 1-in.-thick insulation, 10 per cent for 2-in.-thick insulation, and 5 per cent for 3-in.-thick insulation; and at lower velocities there is even less effect. This small increase for well-insulated surfaces is due to the fact that air motion over the surface of the insulation reduces only the resistance to heat flow offered by the film of air at the surface and does not affect the internal resistance to heat flow offered by the insulation proper, which is many times as large as the surface resistance. This is at once apparent if we consider an example using the heat-flow equation for flat materials. Consider a particular case of a $1\frac{1}{2}$-in.-thick insulation, having a conductivity of 0.5 Btu/(hr)/(sq ft)/(deg F temperature difference between surfaces per 1 in thick) and a surface-film coefficient of heat transfer of 2.0 Btu/(hr)/(sq ft)/(deg F temperature difference, insulation surface to air) under still-air conditions. Then

Insulation resistance = thickness/conductivity = 1.5/0.5 = 3.0
Surface resistance = 1/heat-transfer coefficient = 1/2.0 = 0.5
Total resistance = 3.5
Rate of heat transfer = 1/total resistance = 1/3.5 = 0.28 Btu/(hr)/(sq ft)/(deg F temp diff)

If the surface resistance is reduced to such an infinitesimal quantity as to be practically eliminated, as from the effects of wind and rain on an outdoor installation, we find that the rate of heat transfer becomes

1/3.0 = 0.33 Btu/(hr)/(sq ft)/(deg F temp diff)

in which case the maximum possible increase in heat transfer is

$$[(0.33 - 0.28)/0.28] \times 100 \text{ per cent} = 18 \text{ per cent}$$

Thus it is seen that, since air motion increases insulated losses only slightly compared with the increase in bare losses, the saving when air motion is present is much larger than the saving under still-air conditions. Therefore, all analyses of savings based on still-air conditions are extremely conservative.

It should be noted that the example of the effect of air flow in increasing heat losses from insulation, together with the figures given, is based on the assumption that the insulation is tightly sealed. However, if joints are left open or the insulation is cracked to the extent that air can circulate through them, the increases will be far greater than those illustrated. Therefore, insulation should always be applied snugly to the pipe with joints sealed as tightly as possible. On outdoor equipment this is particularly important, in which case weatherproof jackets should also be well lapped and sealed. Any plant-insulation survey should take note of these points since no insulation that contains open joints or cracks or is loosely hanging in place is effecting the maximum savings possible.

Surface Temperatures Not a Reliable Measure of Heat Loss. In making plant surveys to determine surfaces losing heat, it should be stressed that no attempt be made to judge the need by the feel of the surface or even by measuring the surface temperature. Surface temperature alone is absolutely no measure of heat loss. The surface temperature will depend upon the temperature of the surrounding air, the proximity of other hot or cold objects, the velocity of the air adjacent to the surface, and the nature of the surface, whether polished, dull, etc. It is impossible to evaluate all the variables affecting a particular surface-temperature reading and thus to determine from it the actual heat loss. As an example, it is obvious that, with a temperature of 125°F on the side walls of two adjacent boilers a few feet apart and the confined air between at approximately the same temperature, the wall may be losing practically no heat. On the other hand, the rear boiler wall at only 100°F radiating heat to a cold exterior building wall at 50°F a few feet away is losing a relatively large quantity of heat.

Air circulation, which exists in all installations, causes such large variations in heat losses with the same temperature differences that, with an air velocity of only a few hundred feet a minute, the surface temperature becomes meaningless. As previously shown in the resistance example, the effect of air motion over insulation is to lower the surface resistance, or, stated in another way, to increase the rate of heat transfer from the surface. This cools the surface to a lower temperature than it would have under still-air conditions, and hence with a lower surface temperature, more heat is actually being lost than under still-air conditions. It was also shown in the resistance example that the effect of moving air on the over-all heat loss through insulation is small, but we can see from the above example that surface temperatures are not affected in the same proportion for a number of reasons. Hence they are valueless for measuring heat losses.

Insulation Types Available. There are a number of types of insulations available, the more common being described in the following material. The properties of weight, temperature-use range, and typical conductivity values are all shown in Table 11-12. These pipe insulations are, in general, furnished to fit all standard sizes of steel pipe and copper tubing in 3-ft-long cylindrical or semicylindrical sections or curved segments in thicknesses starting at 1 in. and increasing in thickness by $\frac{1}{2}$-in. increments. Some types are also furnished in "standard" thickness ($\frac{7}{8}$ in. for pipe sizes $\frac{1}{2}$ to $1\frac{1}{2}$ in.; $1\frac{1}{32}$ in. for 2 to $3\frac{1}{2}$ in.; $1\frac{1}{8}$ in. for 4 to 6 in.; $1\frac{1}{4}$ in. for 7 to 10 in.; $1\frac{1}{2}$ in. for 12 in. and larger) and "double-standard" thickness (two-layer) ($1\frac{15}{16}$ in. for $\frac{1}{2}$- to $1\frac{1}{2}$-in. pipe sizes; $2\frac{5}{32}$ in. for 2 to 3 in.; $2\frac{1}{4}$ in. for $3\frac{1}{2}$ and 4 in.; $2\frac{5}{16}$ in. for $4\frac{1}{2}$ to 6 in.; $2\frac{1}{2}$ in. for 7 to 10 in.; 3 in. for 12 in. and larger). Pipe insulations are regularly furnished with light canvas pasted on the outer surface. The materials are also furnished in flat or curved blocks of such standard sizes as 3 by 18 in., 6 by 36 in., and 12 by 36 in. in regular thicknesses, increasing by $\frac{1}{2}$-in. increments from 1 in., for application to flat or large-diameter curved surfaces. Cement or dry-powder insulation, to be mixed with water, is furnished for application to small fittings and surfaces too irregular to apply blocks. Both block and sectional-pipe insulation have lower thermal conductivity than cement and hence are to be preferred wherever applicable. Sectional-pipe and curved segmental insulation are wired or metal-banded to the pipe. Blocks are usually wired in place and finished with ½ in. of asbestos cement applied in two coats and reinforced with hexagonal wire mesh. This gives a smooth, hard, continuous surface.

Pipe insulation has been applied to steel pipe or metal flues and ducts on the outer or room side for pipe temperatures reaching from 950 to 1050°F. For higher temperatures, as in cylindrical steel flues carrying waste-heat gases operating up to 1200°F, internal insulation has been used, the lining consisting of hard-surface premolded special diatomaceous-silica blocks. Recent operating tests have been successfully made by the United States Navy using internal insulation on gas-turbine piping where gases were 1450°F. This 3,000-hp plant is the first large gas turbine reported ever to have operated continuously at peak efficiency functioning successfully at such high temperature. The insulation used was calcined diatomaceous silica 1900°F preformed segmental insulation applied in three layers. The insulation is finished on the inside with a high-temperature special alloy-steel liner while the regular carbon-steel outer pipe is kept at a safe temperature by means of the insulation between.

Table 11-12. Thermal Conductivity k of Different Types of Industrial Pipe Insulations for High- and Medium-Temperature Piping

Material	Density, pcf	General usage limit	Mean temperature, deg F							
			100	200	300	400	500	600	700	800
			k,* Btu/(hr)/(sq ft)/(deg F/in.)							
85% magnesia type	11–15	Up to 600 F	0.41	0.45	0.48	0.52				
Asbestos felt, laminated type	30–35	Up to 700 F	0.39	0.44	0.49	0.54				
Mineral-wool type (wire reinforced)	10–15	Up to 800 F	0.40	0.45	0.50	0.55				
Molded-asbestos type	13–18	Up to 750 F	0.34	0.39	0.44	0.49	0.54			
Diatomaceous-silica type	23–25	Up to 1900 F	..	0.65	0.67	0.69	0.71	0.73	0.75	0.77
Expanded-vermiculite type	17–20	Up to 1000 F	0.81	0.87	0.93	1.00	1.07			

* Average of values from one laboratory on materials of various manufacturers.

85% Magnesia Type. This is the most widely used of the various types for application on hot surfaces up to 600°F. This is a lightweight molded material composed of approximately 85% hydrated basic magnesium carbonate reinforced with asbestos fiber. Furnished in cylindrical half sections or segments 3 ft long for pipes, in thicknesses of "standard" and 1½ in.; 2 in.; 2½ in. in single-layer and "double-standard" (two layers) and 3 in. (double layer); also in flat blocks 3 in., 6 in., 9 in., and 12 in. wide by 18 and 36 in. long and curved blocks approximately 6 in. wide by 36 in. long. Blocks are regularly furnished in thicknesses from 1 to 4 in. by ½-in. increments. They are readily cut with knife or saw for fitting to bent pipe, fittings, and flanges. 85% magnesia is also furnished in powdered-cement form. It is sometimes used as a second layer over molded diatomaceous silica on high-temperature piping to obtain the most economical and thermally efficiency construction.

Asbestos Felt, Laminated Type. This material consists of asbestos felt, approximately 35 to 40 laminations per in. of thickness, cemented together at intervals with heat-resisting adhesive. It is furnished in cylindrical pipe sections 3 ft long split through one side for springing over the pipe in thicknesses of 1 in., 1½ in., 2 in., and 2½ in. single layer and in 2 in., 2½ in., and 3 in. double layer. It is also furnished in sheet or block form up to 36 by 36 in. in thicknesses from ½ to 4 in. This material, like 85% magnesia, is sometimes used as an outer layer over molded diatomaceous silica for high-temperature piping where the extra ruggedness of this type is required. It is recommended for temperatures up to 700°F.

Mineral-wool Type. This material is composed of synthetic mineral fibers made from rock, slag, or glass by melting and blowing with steam or air. The fibers are held between metal fabrics which are wired to each other by tie wires. It is shipped flat in pipe lengths of 2 ft and widths sufficient to encircle the pipe. This flexible type of pipe covering is used mainly where material must be occasionally removed for pipe inspections, etc., as in oil refineries. It is furnished for pipes and flat surfaces in thicknesses from 1 to 4 in. It is generally used for temperatures up to 800°F where removed frequently.

Molded-asbestos Type. This material is composed principally of natural asbestos mineral fiber, usually brown amosite, bonded with a small quantity of heat-resisting adhesive and molded into pipe sections and blocks. It is furnished in the same general forms and sizes as 85% magnesia and in split half sections up to 30-in. pipe size and in single layers up to 5 in. thick. Blocks are made up to 3 ft by 3 ft by 5 in. thick. It is used for temperatures up to 750°F. It is also combined with high-temperature ingredients and furnished for temperatures up to 1200°F either in one layer or integrally combined with an outer layer of 750° material.

Diatomaceous-silica Type. This is a product made of calcined diatomaceous silica combined with magnesium carbonate, bonding clay, and asbestos. It is also molded like 85% magnesia and is furnished in the same general forms and sizes and also in special thicknesses, nominally 1½ in. and 2 in., but actually of proper thickness to fit snugly inside the outer layers of 85% magnesia or laminated-asbestos felt of standard pipe sizes. It is suitable for 1900°F and is used as the inner temperature-reducing layer in combination with 85% magnesia or laminated-asbestos felt.

Expanded-vermiculite Type. This is made of vermiculite (mica) expanded by heat, which is bonded with asbestos fiber and gypsum. It is furnished in the same general form and sizes as 85% magnesia.

Asbestos-blanket Type. This is a mattress-form insulation composed of woven asbestos-cloth covers filled with amosite asbestos fiber, tufted with wire at close intervals and employing sewed-on metal hooks for wiring to heated equipment. It is used for quick accessibility to bolt heads of turbine flanges, pipe fittings, etc., where insulation must conform to irregular surfaces and be readily removable and replaceable. It is made to order for special sizes required usually in thickness of 1½, 2, and 2½ in. single layer and double layer for greater thicknesses. It is used for temperatures up to 850°F depending upon the type of asbestos cloth used.

Insulating Cements. There are several general classes of insulating cements, the first being a bulky fibrous type such as those composed principally of mineral wool, glass wool, or expanded vermiculite combined with some asbestos fiber and bonding clay. These cements are designed for mixing with water and applying in plastic form to irregular contour surfaces where more efficient block materials are not applicable. They readily adhere to metal surfaces and can be applied in thick layers—about one inch. Another type, such as 85% magnesia cement and diatomaceous-silica cement, is of the same composition as these block materials but is furnished in powdered form. These are also used on irregular surfaces, applied in layers about one-half inch thick. A third type, used principally as a finish over other insulations such as block, is composed principally of asbestos fiber and produces a semihard surface. This type is usually applied in two layers, ½ in. total thickness, over the block insulation and sometimes mixed with portland cement to give still harder surfaces. A fourth type, semirefractory, is composed of asbestos fiber and refractory materials and is used in a ½-in. layer for protecting block insulation when applied as a lining in flues.

Factors Governing Selection of Insulation. The proper selection of insulating material for a particular service is an engineering problem and cannot be left to rule-of-thumb methods. It must be determined that the material has

1. Suitable heat-resisting qualities to withstand the maximum working temperature of the equipment. Laboratory and service heat-resistance tests are the proper guides in this respect. It is not possible, although often attempted, to deduce accurately from the chemical composition what will happen to a particular material under temperature.

2. Low thermal conductivity. Conductivity must be known from accurate laboratory tests over the entire temperature range at which the equipment will operate.

It is not sufficient to know the conductivity at only one temperature since conductivity increases with temperature.

3. Sufficient durability and structural strength to assure long life under the particular service conditions and sufficient internal strength to maintain its original shape and particularly its thickness on which its ultimate insulating value depends.

4. Adaptability. Furnished in a form readily applied to the particular surface.

5. Resistance to moisture or to chemical fumes in some particular installations.

6. Proper thickness to effect the most economical heat saving or to accomplish some particular purpose such as preventing excessive temperature drop or condensation in steam lines or to effect comfortable working conditions, etc.

7. Proper method of application to ensure retention of its original strength and insulating value when in place.

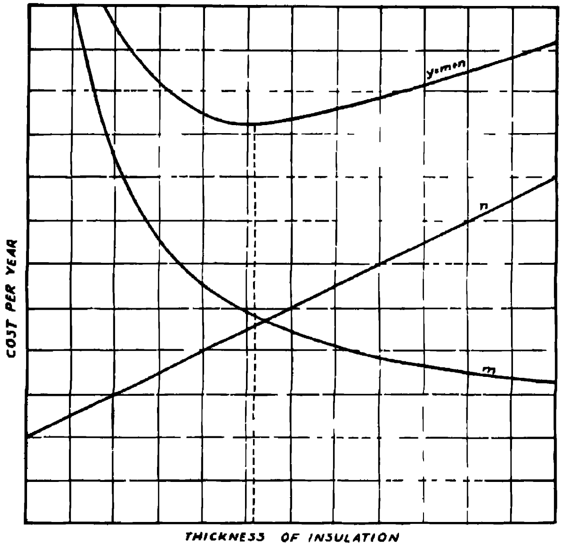

FIG. 11-29a. Economical thickness of insulation.

Since the selection of proper thickness is of primary importance in any installation, both because it affects directly the initial investment and the final heat-saving or other heat-retarding effect, we will consider this in detail.

Selection of Insulation Thickness for Heat Saving. If the principal reason for which insulation is being used is to effect a heat saving, the proper or economic thickness is that thickness the last increment of which pays an adequate return on the investment in that increment.

Figure 11-29a shows that, as the insulation thickness increases, the cost per year n (first cost multiplied by percentage fixed charges) increases and that also with increase in thickness the heat-loss cost per year m decreases. Then the total cost per year y is the sum of the yearly insulation cost and the yearly heat-loss cost or the cost at which a given temperature is maintained. The point on curve y at which the total cost is a minimum is the most economical thickness. Greater thickness would mean that the total cost is too high because of the increased cost of insulation; a lesser thickness, that the total cost is too high because of excessive heat loss due to the insulation's being too thin.

It is possible to determine the economic thickness for any given set of conditions by calculating the yearly cost of heat loss and the yearly cost of insulation for a number of different commercial thicknesses, increasing, for example, by ½-in steps, and then select the thickness which gives the minimum total cost per year. However, this would be laborious if there were a number of such calculations to be made.

To eliminate the necessity for this, McMillan[6] determined rational formulas for the economic thickness applying to both flat surfaces and pipes. He set up equations for the cost per year of heat loss m and cost per year of insulation n and adding the two obtained the equation for the total cost y. Then, to determine the thickness which gave the lowest total cost or the minimum value of y, he calculated this by the mathematical method of differentiating the equation and equating the first derivative to zero. This gave the economic-thickness equation. Since these equations are rather involved mathematically, the results of substituting values in the equations have been plotted in the alignment chart (Fig. 11-29*b*) from which the economic thickness can be accurately determined for all conditions without the necessity of any calculations. To use the chart proceed in the direction shown by the arrows after first determining the following factors:

1. *Hours of Operation per Year.* This should be the actual number of hours the equipment operates.

2. *Value of Heat.* As previously pointed out this should be the total value at the point where the insulation is applied in dollars per million available Btu. Sometimes this value will be higher than the average cost of heat in the plant, owing to the increased production or other useful results produced by the saving of a particular portion of the heat.

3. *Temperature Difference* (heated surface to be insulated minus air temperature). It is assumed that the heated surface temperature stays constant regardless of the thickness of insulation placed on it. This is true for saturated-vapor lines such as steam and can be assumed to be the case for practical purposes for other pipe lines or metal surfaces heated by liquids. Therefore, the temperature difference used in the chart can be the temperature of the vapor or liquid minus the air temperature.

4. *Conductivity of the Insulation.* This is determined from values of the material under consideration, such as shown in Table 11-12, furnished by the insulation manufacturer The actual value of this factor is necessarily dependent on the arithmetic mean temperature of the two surfaces of a material of undetermined thickness. However, since conductivity generally does not change rapidly with small increase in temperature, it will be satisfactory for use with this chart to use the conductivity at the arithmetic mean temperature of the heated surface and the air temperature.

5. *Cost of Insulation.* This is the applied cost which includes material plus labor. The method of determining is explained in detail in the notes on Fig. 11-29*b* and later in the text.

6. *Per Cent Annual Fixed Charges.* This factor includes the return desired on the investment, depreciation, insurance, etc., or the total annual fixed charges applied against the total applied cost of insulation expressed as a percentage.

7. *The Pipe Size.* For which the insulation is required.

In using the chart (Fig. 11-29*b*) start with hours of operation per year and follow through as indicated by the dotted line. The choice of proper economical thickness scale is determined by the choice of heat-value and temperature-difference scales. Example: with scale B heat value and scale A temperature difference, or vice versa, use scale AB economical thickness, etc.

Cost of insulation refers to applied cost. Therefore, to obtain proper discount, first use discount for material alone to determine approximate thickness on which to base material and application costs. Then use the following formula to determine the correct discount:

$$\text{Applied cost in per cent discount from list} = 100[1 - (X + Y)/Z]$$

where X = cost of material; Y = cost of application; Z = list price applying to material.

The thickness is again determined, and if it differs from the first thickness, the procedure is repeated until the thickness does not change.

For pipe insulation, where list prices do not apply, use the following formula to convert from applied cost per linear foot to applied cost per board foot:

$$\text{Applied cost per board foot} = 0.382A/dx$$

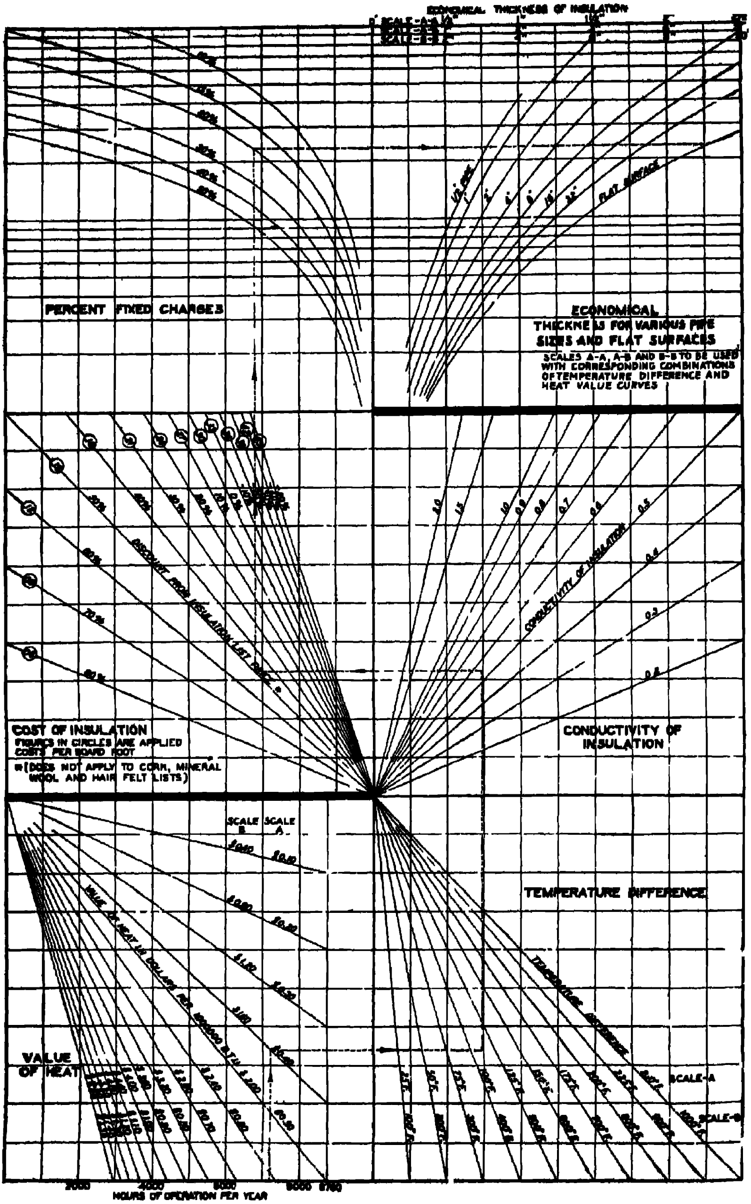

FIG. 11-29*b*. Chart for determining economical thickness of insulation.

where A = applied cost per linear foot; d = outside diameter of insulation in inches; x = thickness of insulation in inches.

Example of Typical Economic Insulation Thicknesses. Typical economic thicknesses determined by means of Fig. 11-29*b* are plotted in Fig. 11-30. This shows how the economic thickness increases with increasing pipe size and with increasing pipe temperature. The particular operating conditions assumed are such as might be found in an average large power or manufacturing plant, *viz.*, heat cost 30 cents per million available Btu, piping kept heated 8,760 hr per year, applied cost of insulation 85 to 135 per cent of list price of insulation, depending upon pipe size, fixed charges 10 per cent, air temperature 75°F. The curves for 800, 1000, and 1200°F pipe temperatures are based on using calcined diatomaceous silica—85% magnesia combination insulation while the curves for 200, 400, and 600°F are based on using 85% magnesia alone. A similar series of curves could be plotted for any other set of conditions existing in a particular plant.

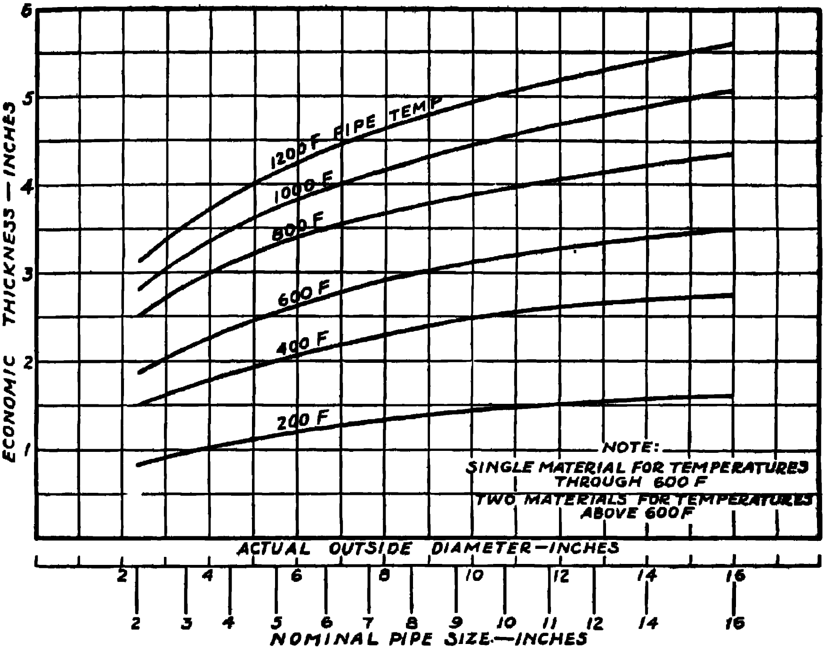

FIG. 11-30. Example of typical economic insulation thickness.

After the total insulation thickness is determined, the nearest commercial thickness, usually nearest ½ in., is selected. Where high-temperature combination insulation is required, a calculation must be made to determine the proportion of the total economic thickness which must be high-temperature material in order to reduce the temperature on the outer-layer material to its safe limit. Here also the nearest commercial thickness must be used and also one which will closely fit a standard pipe size outer layer. Manufacturers furnish special thickness high-temperature inner layers for this purpose.

Where insulation is used for some particular purpose, other than heat saving, such as temperature control, personnel protection, comfort, fire protection, etc., the thickness must be designed to accomplish the particular purpose. For example, if the insulation must limit the temperature drop in a long line carrying superheated steam, it is necessary to calculate the heat loss from the line per hour per pound of steam flowing through the line in the same time and then deduct this loss per pound of steam from the total heat in a pound of steam at the inlet steam conditions to find the total heat in a pound of steam at discharge end of the line. Having previously determined the pressure drop in the line from conventional friction flow formulas, the steam pressure and total heat of the steam at the end of the line are now known, and from the superheated-steam tables the corresponding temperature of the steam for this final pressure and total-heat condition can be found. If the insulation must have a maximum outer surface temperature to protect personnel, a calculation of the temperature gradient through the insulation is made using the resistance conception previously illustrated and the surface temperature determined.

The Importance of Proper Method of Application and Maintenance. The selection of the proper material with full consideration for the temperature and other conditions and the determination of proper thickness are of little importance if the material is applied improperly. For, if it is not properly applied, the actual heat loss may be considerably higher than the calculated value based on laboratory tests. Improper application may cause the insulation to deteriorate, and hence it may not give the years of service for which it was originally planned. It is not possible to give here detailed application specifications for applying insulation to various types of equipment, but there are general factors which apply to the satisfactory application of all insulation. Among these factors are

1. Insulation should be applied securely to and tightly against the pipe or other heated surface. An air space left between the insulation and pipe may result in heat being short-circuited through an open joint or crack. Furthermore, loose insulation is more easily damaged by impact. Some types will sag under vibration if not securely wired or banded in place.
2. Insulation should be applied with joints closely fitted, and multiple-layer insulation should be applied with staggered joints, particularly on high-temperature equipment. Otherwise, pipe expansion and vibration tend to open up through-joints.
3. Insulation should be protected against abuse such as abrasion, knocks, and crushing. Insulation is not a structural material. Hence, where it is liable to be walked upon or have ladders placed against it, runways or scaffold should be provided. This is particularly important on outdoor installations to avoid entry of rain or snow into the material. Where insulation may be subject to impact or abrasion, it should be protected with a suitable sheet-metal jacket or bumper.
4. A moisture-resisting finish should be provided on insulation outdoors, or indoors where it is liable to be splashed from process liquids or otherwise subjected to wetting. This may consist of an asbestos weatherproof jacket over the pipe insulation and asphaltic cement finish over insulation on fittings and irregular surfaces. Weatherproof jackets should be securely wired or banded in place, lapped, and sealed with asphaltic lap cement to prevent deterioration of insulation by wind, rain, and snow. Special provision must be made to protect insulation exposed to strong alkali or acid fumes. Underground pipe insulation requires tile protection specifically designed for the service.
5. All indoor insulation not otherwise protected should be finished with heavy canvas pasted or sewed in place, painted with two coats of lead and oil paint.
6. Sturdily constructed removable and replaceable flange insulation covers should be provided where flanges must be inspected frequently.
7. Last but by no means least important is that insulation should be installed by experienced mechanics.

An insulation properly selected for the conditions often outlasts the useful life of pipe to which it is applied; but in order to ensure its maintaining its initial insulating value throughout its life, it should be periodically inspected and maintained in good condition. This requires attention to such items as prompt replacing of torn weatherproof jackets and canvas finish; replacing of broken wires and sections of insulation; tightening of wires, bands, and loose insulation; repainting of canvas. Any cracks in cement or weatherproof finish should be sealed with the original material since wetting causes loss of insulating efficiency and deterioration of the insulation.

In addition to the foregoing, periodic maintainance inspections should check such items as whether improper pipe insulation is the cause of excessive condensation discharged from traps or whether steam temperatures have been raised beyond the safe limit of the insulating material used. It is also necessary to eliminate water hammer or other causes of excessive vibration which may cause joints to open in the insulation; to stop leaks in steam or hot-water lines as soon as possible since impinging steam causes rapid deterioration, and water from a leak may follow along a pipe, saturate the insulation, destroy its insulating value, and at times cause the insulation to fall from the pipe. Quite often when pipe repairs are made, insulation sections or flange and fitting insulation covers are removed. These should be promptly reapplied as carefully as though it were a new installation since improperly applied insulation is often thermally inefficient and is more liable to failure. Such inspections will pay dividends, not only in heat savings, but in saving of materials and labor.

Table 11-13. Data on Freezing of Water in Pipes*

Steel pipe size, in.	Hair felt, No of layers each 1 in thick	Btu/(deg temp diff)/(hr)/(lin ft)	Hr to cool from 42 to 32°F, no flow	Lb water flow/(hr)/(lin ft) to prevent freezing
½	2	0.0895	0.417	0.537
	3	0.0747	0.500	0.448
	4	0.0660	0.565	0.396
1	2	0.1125	0.825	0.675
	3	0.0911	1.020	0.548
	4	0.0798	1.160	0.480
1½	2	0.1400	1.400	0.840
	3	0.1126	1.740	0.676
	4	0.0972	2.020	0.583
2	2	0.1586	1.940	0.952
	3	0.1244	2.480	0.747
	4	0.1063	2.900	0.638
3	2	0.2062	3.250	1.237
	3	0.1572	4.270	0.943
	4	0.1322	5.080	0.793
4	2	0.2450	4.550	1.470
	3	0.1850	6.020	1.110
	4	0.1548	7.200	0.929
5	2	0.2887	5.920	1.733
	3	0.2146	7.960	1.289
	4	0.1764	9.690	1.059
6	2	0.3302	7.350	1.981
	3	0.2434	9.880	1.460
	4	0.1984	12.200	1.191
8	2	0.4100	10.050	2.460
	3	0.2960	13.900	1.776
	4	0.2390	17.250	1.434
10	2	0.4930	13.000	2.960
	3	0.3536	18.100	2.122
	4	0.2830	22.700	1.698
12	2	0.5720	15.800	3.432
	3	0.4090	22.200	2.454
	4	0.3222	28.100	1.933

* Figures are based on an air temperature of −20°F. This is an approximate 60° temperature difference. If air is to be taken at 0°F, the time to cool 10° will be approximately 45 per cent greater. The last column shows minimum amount of water at 42°F to prevent freezing.

These rates of flow include no safety factor to provide for temporary reductions due to lower pressure, etc. Weights of water are per foot, so the minimum flow in a line 100 ft long must be at least 100 times the tabular value.

LOW-TEMPERATURE PIPE INSULATION

The reasons for insulating pipes carrying refrigerants and other low-temperature fluids are similar to those for insulating hot lines. Heat flow is reversible, *i.e.*, a bare pipe 100°F below atmospheric temperature absorbs heat at about the same rate as one 100°F above ambient loses heat.

There are other considerations in addition. Bare, cold pipes drip or become covered with frost and then drip. This promotes corrosion and makes for dirty and unsanitary conditions as well as making space beneath or adjacent to the pipes unsuitable for use. On the economic side it is generally conceded that it costs more to remove heat than it does to supply it; consequently, greater thicknesses of insulation are used on refrigerated lines than on heated ones for equal temperature differences.

In addition to minimizing heat entry into cold piping, there is often the problem of preventing the freezing of water in pipes exposed to winter weather. No insulation, regardless of thickness or low conductivity, will prevent the freezing of still water in pipes exposed to below-freezing temperatures. Insulation retards heat flow; it does not eliminate it. If there is no movement of water in a pipe, it is merely a matter of time until the pipe and its contents reach the same temperature as the surrounding air. Table 11-13 can be used to determine the number of hours to cool water to the freezing point and the minimum flow necessary to prevent freezing in the pipe. It was developed for hair felt but can be used for any blanket-type insulation having a similar conductivity, *i.e.*, 0.30 Btu/(hr)/(sq ft)/(deg F per in.).

Conductivities of hair-felt and mineral-wool blankets are usually stated to be below 0.30. However, an appreciable degree of compression results from the wrapping and securing on a pipe, so a less than nominal thickness or a higher unit heat transfer must be assumed.

Table 11-14 is similar and can be used for any sectional-type pipe insulation made in the conventional three thicknesses and having a conductivity k of approximately 0.30 Btu/(hr)/(sq ft)/(deg F per in.).

Table 11-15 gives rates of heat transfer into cold pipes through sectional insulation of the type just described having a conductivity of approximately 0.30. This table is useful when it is desired to know how much heat will be absorbed by a fluid in traversing a given length of pipe, but it does not allow for the differences in rate of heat transfer into the fluid due to film effect on the inside of the pipe. It is assumed the fluid flow will be fast enough to produce turbulence which minimizes film resistance and that all the heat which can traverse the insulation will be absorbed by the fluid in the pipe.

Effective insulating of pipes to prevent sweating or frosting is based on using a thickness of insulation that will keep the outer surface of the insulation at a temperature above the dew point of the surrounding air. This cannot be done for all conditions. As the air nears saturation, the dew-point temperature approaches the dry-bulb or ordinary thermometer reading, the two becoming equal at complete saturation, sometimes called 100 per cent relative humidity. To prevent condensation under such a condition would require an infinite thickness of insulation. Figures 11-31, 11-32, and 11-33 illustrate this. To use these charts, find out what the climatic conditions are likely to be, and then from a psychrometric chart or table find the dew-point depression or difference between the air temperature (dry bulb) and the dew point. Start from that figure in the column at the left edge of chart, and draw a line horizontally until it intersects the curve representing the over-all temperature difference expected. From the intersection drop vertically to the bottom of the chart, and read the insulation thickness required. If the climatic data are not thought to be very accurate, the column of relative humidities at the right will be accurate enough for most cases. Choose the highest that is likely to obtain for any considerable period, and intersect the over-all temperature difference curve as before. It may be noted that it is rarely worth while to design against humidities over 85 per cent, as there are few areas where higher humidities last for any length of time, and the thicknesses required are unduly expensive.

There are only two types of pipe insulation generally used for low temperatures. One consists of flexible blankets which are wrapped around the pipe, tied on, and a

Table 11-14. Data on Freezing of Water in Pipes*

Steel pipe size, in.	Insulation thickness	Btu /(deg temp diff) /(hr) /(lin ft)	Hr to cool from 42 to 32°F, no flow	Lb water flow/(hr) /(lin ft) to prevent freezing
½	Ice water	0.110	0.43	0.63
	Brine	0.098	0.52	0.56
	Heavy brine	0.087	0.73	0.50
1	Ice water	0.139	0.78	0.79
	Brine	0.124	0.90	0.71
	Heavy brine	0.104	1.20	0.59
1½	Ice water	0.174	1.27	0.98
	Brine	0.134	1.72	0.76
	Heavy brine	0.118	2.04	0.67
2	Ice water	0.200	1.69	1.14
	Brine	0.151	2.31	0.86
	Heavy brine	0.134	2.71	0.76
3	Ice water	0.269	2.73	1.53
	Brine	0.186	4.02	1.06
	Heavy brine	0.162	4.68	0.93
4	Ice water	0.294	4.08	1.68
	Brine	0.209	5.84	1.19
	Heavy brine	0.182	6.77	1.04
6	Ice water	0.404	6.45	2.30
	Brine	0.259	10.01	1.48
	Heavy brine	0.228	11.61	1.30
8	Ice water	0.455	9.75	2.60
	Brine	0.318	14.00	1.82
	Heavy brine	0.263	17.00	1.50
10	Ice water	0.559	12.22	3.20
	Brine	0.383	17.95	2.20
	Heavy brine	0.309	22.32	1.76
12	Ice water	0.648	14.80	3.70
	Brine	0.438	22.12	2.50
	Heavy brine	0.364	26.80	2.08

* Figures are based on an air temperature of −20°F. This is an approximate 60° temperature difference. If air is to be taken at 0°F, the time to cool 10° will be approximately 45 per cent greater. The last column shows minimum amount of water at 42°F to prevent freezing.

These rates of flow include no safety factor to provide for temporary reductions due to lower pressure, etc. Weights of water are per foot, so the minimum flow in a line 100 ft long must be at least 100 times the tabular value.

sealing membrane applied to exclude moisture. On this type, hair felt has by far the longest history. Certain mineral-wool blankets are also offered for the same purpose. Blanket-type insulation has the advantage of versatility. Given enough material on the job, any size of pipe and any combination of fittings may be covered. It has the disadvantage of requiring exceptionally good and careful workmanship, particularly in applying the sealing membrane which protects it. No insulation job can be better than the workmanship of the applicator, but there is more inherent difficulty in sealing over a relatively soft and yielding surface. Also, the chance of the seal being broken by a blow or rough treatment is greater. This type of insulation is the only one that can be employed on hangers that are affixed directly to the pipe.

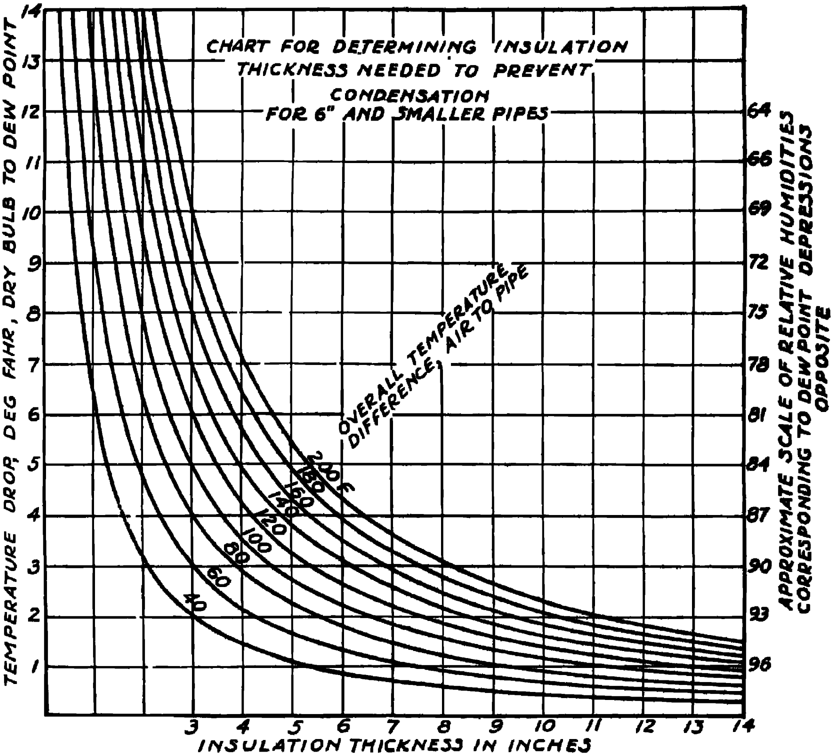

Fig. 11-31.

The second major type of pipe insulation consists of molded or formed sectional overing. Comparatively solid, it is furnished in either half sections 36 in. long or in half sections joined by an outer wrapper providing both a seal and a hinge joining the half sections. Fittings are insulated either with molded jackets or by use of a wrap-on-type insulation as in the type previously described. The sectional-type insulation is easier and quicker to apply. It is also decidedly easier to seal effectively against moisture.

Sealing of low-temperature insulation is of major importance as there is always a pressure, resulting from temperature differences, tending to force water vapor into the insulation, where it will condense. If the temperature is low enough, freezing follows, not only seriously impairing the insulating efficiency but frequently destroying the insulation and corroding the pipe.

Table 11-15. Rates of Heat Transmission Sectional Insulation

The rates of heat transmission given below are expressed in Btu per square foot (and also per linear foot), per hour, per degree temperature difference between fluid in the pipe and air surrounding the pipe. In the case of screwed fittings, where the proportion of fittings to pipe is not unusually high, no great error will be introduced by figuring small fittings, such as ells and tees, as 1 linear ft of pipe; and larger fittings, valves, crosses, etc., as 1½ lin ft. Refer to Table 11-7, Areas of Flanged Fittings, which appears on page 749.

Pipe size, in.	Ice-water thickness			Brine thickness			Heavy-brine thickness		
	Btu /lin ft	Btu/sq ft pipe surface	Btu/sq ft outer surface	Btu /lin ft	Btu/sq ft pipe surface	Btu/sq ft outer surface	Btu /lin ft	Btu/sq ft pipe surface	Btu/sq ft outer surface
½	0.110	0.502	0.105	0.098	0.446	0.075	0.087	0.394	0.051
¾	0.119	0.431	0.101	0.111	0.405	0.085	0.094	0.340	0.051
1	0.139	0.403	0.118	0.124	0.352	0.086	0.104	0.294	0.053
1¼	0.155	0.357	0.118	0.131	0.300	0.077	0.113	0.260	0.054
1½	0.174	0.351	0.133	0.134	0.270	0.073	0.118	0.238	0.053
2	0.200	0.322	0.139	0.151	0.244	0.077	0.134	0.214	0.057
2½	0.228	0.303	0.145	0.170	0.226	0.081	0.147	0.197	0.059
3	0.269	0.293	0.158	0.186	0.202	0.079	0.162	0.176	0.059
3½	0.295	0.282	0.161	0.191	0.183	0.073	0.176	0.167	0.061
4	0.294	0.248	0.140	0.209	0.176	0.076	0.182	0.154	0.058
5	0.349	0.239	0.148	0.241	0.165	0.080	0.202	0.138	0.057
6	0.404	0.233	0.154	0.259	0.150	0.076	0.228	0.130	0.060
8	0.455	0.201	0.139	0.318	0.140	0.081	0.263	0.116	0.059
10	0.559	0.198	0.147	0.383	0.135	0.086	0.309	0.110	0.062
12	0.648	0.194	0.150	0.438	0.131	0.088	0.364	0.108	0.067
14	0.670	0.182	0.142	0.487	0.132	0.093	0.392	0.107	0.068
16	0.750	0.179	0.143	0.542	0.130	0.094	0.434	0.104	0.069
18	0.833	0.176	0.145	0.604	0.127	0.096	0.476	0.102	0.070
20	0.919	0.175	0.146	0.661	0.126	0.097	0.520	0.100	0.071

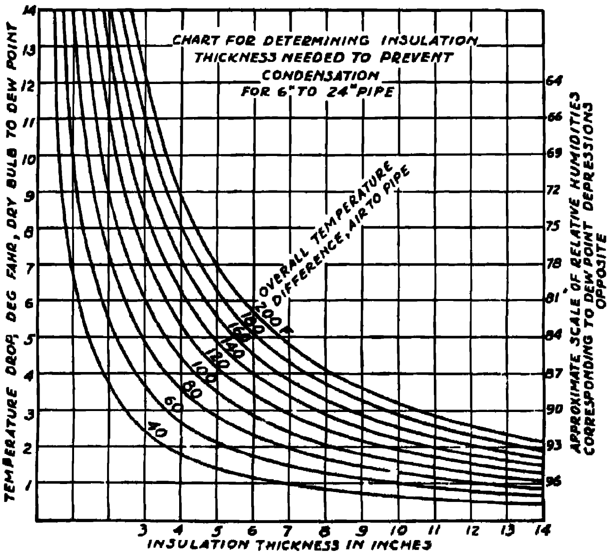

FIG. 11-32.

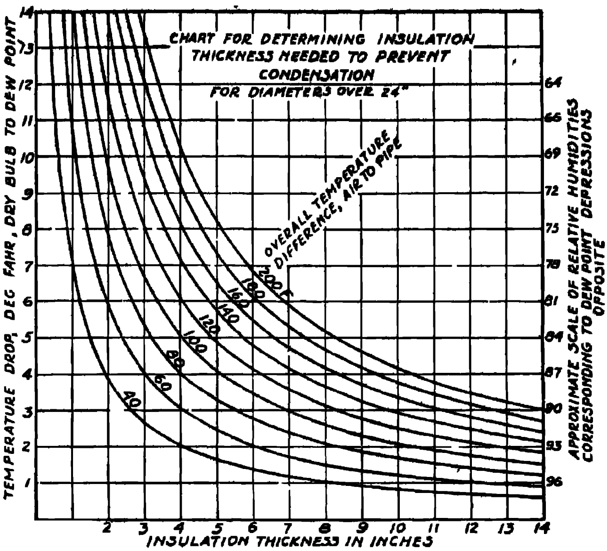

FIG. 11-33

Tables 11-16 to 11-17 show proper spacing of parallel pipes and hanger spacing It should be noted that wider spacing of parallel pipes is required if blanket-type insulation is specified The mechanic must be able to get his hands between the finished insulation on parallel pipes when he applies the sealing membrane. Table 11-16 is based on sectional, molded insulation

Table 11-16

Insulation thickness	Space required between parallel pipes in	Space required between pipe and adjacent surfaces, in
Ice water thickness		
Up to 6-in pipe—screwed fittings	7	5
Larger than 6-in pipe—screwed fittings	11	6
All pipe sizes—flanged fittings	12	8
Brine thickness		
Up to 6-in pipe—screwed fittings	9	7
Larger than 6-in pipe—screwed fittings	15	9
All pipe sizes—flanged fittings	16	10
Heavy brine thickness		
Up to 3-in pipe—screwed fittings	11	9
Larger than 3-in pipe—screwed fittings	19	13
All pipe sizes—flanged fittings	20	14

Hanger Details for Pipe Insulation. Steel saddles or shields on the outside of the insulation must be provided in all cases where the pipe hanger or support is not affixed to the pipe (see Fig 11-34)

In most cases, the saddle and outside strap hanger is preferable to fastening the strap or rod support to the pipe, as it does not involve the insulating and the sealing off of a through-metal connection between the cold pipe and warm air

Where extra-heavy loads, difficult to calculate, may throw excessive weight on one or several hangers, it may be advisable to attach supports directly to the pipe as shown in Fig. 11-35. An example of such a condition might be a horizontal line with several risers or branches near together and, possibly, the inclusion of a cluster of valves, etc., within a short length.

The third type of support, the trapeze style shown in Fig. 11-36, is not desirable because of the poor distribution of load on the pipe insulation.

Size of Shields or Saddles. Length = outside diameter of insulation; circumference = approximately 270 deg or ¾ of a circle; gauge of metal = 12 gauge where outside diameter of insulation is less than 8 in. and 10 gauge where outside diameter of insulation is 8 in. or greater.

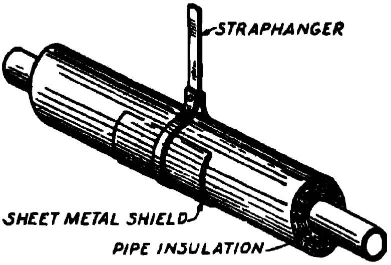

FIG. 11-34. Outside hanger and shield.

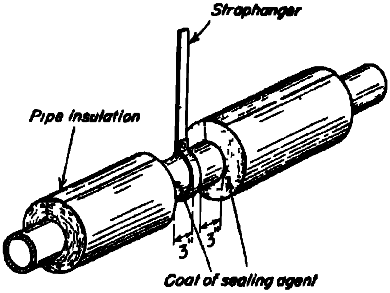

FIG. 11-35. Inside hanger.

Spacing of Hangers. Based on use of ¾ round metal shields and on consideration of structural properties of high-pressure pipe as well as indentation of insulation Where low-pressure pipe is involved, hanger spacing may need to be reduced Table 11-17 is based on straphangers as shown in Fig 11-34: Where unusual hanger loads are involved, the loads per shield should not be exceeded (see Table 11-18)

Table 11-17. Hanger Spacing Based on Pipe Size

Insulation thickness*	Pipe sizes and maximum hanger spacing, ft				
	1½ in.	2½ in	4 in	6 in †	8 in.†
Ice water or light duty	13	17	25	26	34
Brine or standard duty	12	16	23	24	32
Special brine or heavy duty	11	14	22	23	28

* Ice water or light duty, 1.5 to 2 in.; brine or standard duty, 1 8 to 3.2 in ; special brine or heavy duty, 2.8 to 4 0 in.

† Reduce spacing one-third for trapeze-type hangers.

Table 11-18. Loads per Shield

Insulation thickness	Pipe sizes and loads, lb				
	1½ in.	2½ in.	4 in.	6 in	8 in.
Ice water or light duty ..	500	600	900	1,000	1,500
Brine or standard duty.	600	800	1,000	1,100	1,600
Special brine or heavy duty	800	1,000	1,100	1,200	1,800

Sealing and Insulation. Sealing on hanger should be carefully done to keep moisture from running down into insulation. The following steps summarize operations shown in Figs. 11-35 and 11-37: (1) Insulation should be applied not closer than 2 to

3 in. from the hanger on either side. (2) The exposed ends of the pipe insulation should be coated with the sealing agent and the pipe and hanger with protective paint. (3) Two or three (3 for heavy brine) built-up layers of hair felt or equal should be applied with a double wrap of tape per layer sealed to insulation and hanger as shown, and sealing agent should be applied to each layer and all ends. (4) A supplementary built-up layer of hair felt about $1\frac{1}{2}$ in. thick should be carried up the hanger at least 12 in. (5) A double wrap of tape should be sealed to the hanger at the upper end and to the membrane over the insulation at the lower end.

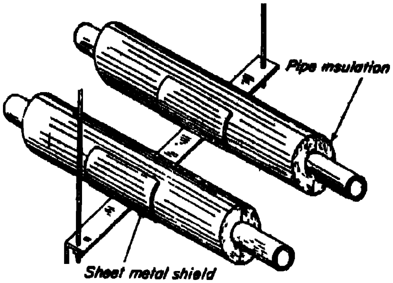

Fig. 11-36. Trapeze hanger and shields.

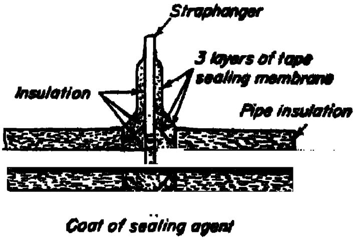

Fig. 11-37. Insulation and sealing of inside hanger.

Mathematical calculations for heat entry into refrigerated pipes and vessels are made in accordance with the same equations detailed for heat losses. There is not, however, a suitable equation for economic thickness of sectional-cold pipe insulation since the list prices of the commonly used materials do not follow any regular law.

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RUBBER-LINED PIPE AND PROCESS EQUIPMENT

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Rubber-lined pipe, storage, and process equipment is particularly valuable to industry as it incorporates the corrosion and abrasion resistance of the rubber with the structural properties of steel. Furthermore, as the requirements of the process with regard to installation and cure both in the factory and in the field are very flexible and as the limits with regard to size and detail of design are very broad, practically all types of industrial equipment are suitable for rubber lining or covering. In the combination, the rubber provides the necessary corrosion and abrasion resistance, and the steel provides the structural strength, whether the requirement be pressure, vacuum, or mechanical.

NATURAL-RUBBER AND GR-S LININGS

Chemical Resistance. With few exceptions, rubber linings are suitable for use with most industrial solutions. While it is not possible in a brief review of the subject to enumerate all possible materials, a statement of general classifications will suffice to give a reasonable general knowledge of possible applications. In any event, in the absence of specific experience it is best to refer detailed service conditions to the rubber manufacturer for his recommendation.

Rubber linings are suitable for use with the following:

1. Most inorganic acids, including hydrochloric, phosphoric, dilute sulfuric, hydrofluosilicic, etc.
2. Many organic acids, such as acetic acid, tannic acid, gallic acid, etc.
3. Inorganic salt solutions, such as ferric chloride, zinc chloride, tin chloride, sodium cyanide, ferrous sulfate, etc.
4. Inorganic bases, such as sodium hydroxide, calcium hydroxide, etc.
5. Plating solutions, with the exception of chromium, including nickel, brass, tin, zinc, silver, cadmium, etc.
6. Bleach solutions, sodium hypochlorite and chlorine solutions

Rubber linings are not suitable for use with the following:

1. Strong oxidizing agents, such as nitric acid, chromic acid, sulfuric acid above 50 per cent concentrations
2. Aromatic hydrocarbons, such as benzol, toluol, etc.
3. Aliphatic hydrocarbons, such as gasoline, petroleum, derivatives, etc.
4. Chlorinated hydrocarbons

Temperature Resistance. Under most conditions unprotected rubber linings are suitable for temperatures up to 180°F. With protective inner linings of wood or brick higher temperatures are possible. It is rather difficult to define the extreme upper temperature limits as in many cases the combined effect of corrosive solution and temperature must be considered. Under particularly severe conditions the maximum allowable temperature might be 130 to 140°F. Under other conditions 200 to 210°F might be permissible. Here again, in the absence of specific experience, the detailed service conditions should be referred to the rubber manufacturer for his recommendations.

Physical Properties of Rubber Linings. Rubber linings are supplied in a wide variety of compounds varying in hardness from soft rubber to ebonite and in type and degree of compounding as service conditions may require. Semihard and hard linings are usually installed with soft-rubber cushion next to the metal to compensate for differential expansion and contraction between rubber and metal. The usual thickness of rubber lining for pipe and fittings up to 5 in. nominal size, inclusive, is ⅛ in., and for piping of 6 in. nominal size and over and for tanks and process equipment $\frac{3}{16}$ in. For severe services thicker linings are used. In addition to corrosion, abrasion, and temperature resistance, the principal requirement of a satisfactory lining compound is low water absorption. This latter requirement is very important if penetration through the lining to the metal is to be avoided and if permanent protection of the metal is to be provided.

Tensile strength and elongation, which are properties commonly considered in making comparisons of rubber stocks, are not in themselves of particular importance when considering the relative suitability of lining materials. It is possible that a compound with relatively low initial tensile strength and elongation may be more suitable than a compound with relatively high initial tensile strength and elongation. Relative percentage loss of tensile strength and elongation upon exposure to corrosive solutions and temperature is important in that it is an indication of relative rate of deterioration in service, and good aging properties are essential. All the above factors are taken into consideration when the rubber manufacturer makes stock recommendations for any particular service.

Abrasion Resistance. Rubber linings can be compounded to provide remarkable abrasion resistance, with the result that they find wide application in the handling of abrasive slurries and in the hydraulic or pneumatic transportation of various abrasive materials. Such installations may involve abrasion only or a combination of abrasion and corrosion.

Dielectric Properties. Rubber linings are nonconductive and hence are extensively used in electrochemical processes such as electroplating where absence of current loss is essential to economy and quality. The usual test procedure is to ground the metal and expose the rubber surface to a potential of 15,000 volts. This potential will expose defects in material and workmanship but will be repelled by a lining of proper quality.

Other Properties. Rubber lining also has the virtue of noncontamination of product. In other words, a product in contact with rubber does not extract from the rubber materials that will discolor or otherwise contaminate. Rubber linings are frequently used primarily to prevent metallic contamination of product during process, storage, and transportation.

SPECIAL SYNTHETIC LININGS

There are certain synthetic elastomers such as neoprene, perbunan, hycar, etc., which will handle service conditions for which natural rubber and GR-S are not suitable. These materials are resistant to petroleum and its derivatives and have superior resistance to certain other hydrocarbons and chlorinated hydrocarbons. They lend themselves to installation in most types of equipment. Rubber manufacturers will be in position to make proper recommendations with regard to the use of these materials.

METAL PARTS

Metal parts of proper quality and finish are essential to first-quality rubber lining. Plate thicknesses and reinforcements should be adequate to prevent damage to rubber linings due to excessive distortion. Flange thicknesses should be such as to prevent distortion during assembly. Plates should be free from pitting and castings from porosity. Full welding is required at all points to be rubber-covered, and welds must be chipped and ground smooth and be free from porosity. More detailed specifications follow. These specifications are typical of the general requirements in the industry and will be readily met by any competent fabricator familiar with preparation of metal parts for rubber lining.

Metal Specifications for Welded-steel Tanks or Other Welded Parts. *Material.* Material shall be new full-weight American steel, free from laminations or other physical imperfections. All plates shall be flat, with no appreciable buckle or warpage. All sharp edges of sheared plates must be removed, especially on the inside of tanks. The thickness gauge and weight per square foot shall be no less than the figures listed in Table 11-19.

Dimensions. All dimensions shall be detailed on drawing. Unless otherwise specified, the dimensions for steel tanks are to be considered inside dimensions. A tolerance of plus or minus 1/8 in. is permissible on lengths, width, or height on dimensions 24 in. or less, unless otherwise specified. On dimensions over 24 in., a tolerance of plus or minus 1/4 in. is permissible. The same tolerance applies to over-all dimensions when parts are made up in sections.

Table 11-19

Plate thickness, in.	*Weight per sq ft, lb*
7/64	4 375
9/64	5 625
3/16	7 50
1/4	10 2
5/16	12 5
3/8	15 0
1/2	20 0

Construction. 1. Tanks or welded parts shall be fabricated in accordance with standardized commercial practice to obtain a practical product and uniform quality.

2. All tanks shall be constructed with a minimum number of pieces. Rectangular tanks of the open-top type shall be properly reinforced in accordance with accepted practice in order to provide adequate structural strength and to prevent bulging.

3. The companion angles of sectional tanks which are bolted together must be square, plumb, and smooth, and sections must fit exactly when bolted together. The open ends of sectional tanks shall be provided with suitable bracing so they will not become distorted in transit. Sectional tanks shall be permanently match-marked on spots that are not to be rubber-covered accurately to indicate their position when assembled. Parts must be lined up properly when correctly assembled and marked.

4. Fabricator shall supply necessary bolts, nuts, and washers to complete any assembly shown on the drawings as such.

5. On tanks with dished heads, the dished heads are to be flanged in such a manner as to eliminate wrinkles at the knuckle radius.

Welding. 1. All joints over which rubber is to be applied must be continuous solid welded.

2. All welds must be smooth with no porosity, holes, high spots, lumps, or pockets. Peening is required to eliminate porosity, and grinding is necessary to remove sharp edges and high spots.

3. All corners are to be ground to a minimum radius of 1/8 in.

4. Partitions, braces, supports, or other attachments on the inside of the tank must be fitted flush against the adjacent surface and full-welded from all sides. Spot welding is not permissible. (Welds are to be ground per 3 above.)

5. All body seams must be butt-welded, with continuous solid welding throughout, unless tap weld is necessary to meet special requirements.

6. Vendor shall assume all responsibility for strength of welds.

7. Misalignment of plates and butt-weld seams not to exceed 25 per cent of plate thickness and in no case to exceed 1/8 in.

Outlets. The size, construction, and location of outlets shall be in accordance with the instructions on the drawing. All flanges must be parallel with the tank, or in other words, be so applied that the pipe line will leave the tank at right angles to the surface of the tank, unless otherwise ordered. For all closed vessels, suitable for rubber lining, all manholes shall be not less than 18 in inside diameter. Orifices to be ground to a minimum radius of 1/4 in.

Domes and Fittings. When cast-iron or steel domes and fittings are specified, they shall be free from porosity and sand holes.

Tests. 1. Either test or working pressure, or both, of closed tanks will be specified on the specification print. If no pressure requirements are specified, all closed cylindrical storage tanks shall be thoroughly tested for leaks at 5 lb hydrostatic pressure

2. Vacuum tanks must be tested to the vacuum shown on the drawing.

3. If required, the vendor will furnish affidavit of tests.

Metal Specifications for Pipe and Fittings. *Standard Pipe—Screwed-on Flanges.* 1. Pipe shall be standard weight and grade of steel pipe. Pipe below $3\frac{1}{2}$-in. nominal diameter may be butt-welded, all other pipe shall be lap-welded and free from deep seams, ridges, or other imperfections on rubber-covered surfaces. The heavier standard weight for sizes 10 in. and above shall be furnished unless otherwise stated.

2. Briggs Standard pipe thread shall be used, and lengths shall be fitted with ASME 125-lb Standard CI flanges, unless otherwise stated.

3. Flanges shall be made up to a perfectly tight truly aligned fit by machine. No lubricant or oil shall be used or left on thread in making the assembly, but a rubber cement approved by the purchaser may be used. Pipe must be finished flush with face of flange after assembly (see 4 under Alignment, Tolerances, Finishing below).

Fabricated Steel Pipe—Welded-on Flanges. 1. Fabricated pipe shall be of a good weldable quality steel, free from laminations and other harmful ingredients or defects including those of 1 under Metal Specifications on page 777.

2. Flanges on welded sections shall be ASME 150-lb Standard steel slip-on flanges with plain faces unless otherwise specified.

3. Flanges shall be fillet-welded by shielded-arc process in accordance with dimensions set forth in American Tentative Standards for Pressure Piping, except that fillet on end of pipe shall be built up to allow for finishing to a clean metal surface in accordance with 4 under Alignment, Tolerances, Finishing.

4. Butt welds of single-butt type may be used on butted sections. All welds on surfaces to be rubber-covered must be ground down to a smooth metal surface, using a No. 24 grain grade Q wheel or finer and shall be free from pits or other imperfections.

5. All nozzles in headers and other fabricated branch connections shall be extruded where possible. Vendor shall notify the purchaser when extrusion is impossible. Notification in writing of the limitations of this process as to sizes, dimension, etc., shall be considered as notification that other fabrication methods will be used for all units not included within these limitations.

Alignment, Tolerances, Finishing. 1. A tolerance of $\frac{1}{64}$ in. in alignment of all flanges shall be allowed.

2. Boltholes on flanges on each end shall line up with each other within $\frac{1}{8}$ in. (the standard clearance between bolt and drilled flange holes).

3. Tolerance from face to face or center to face shall be $+0-\frac{1}{8}$ in.

4. The inside edge of pipe shall be turned to a radius of $\frac{1}{4}$ in. by tool or grinding wheel 24-grain grade Q or finer. In case where rubber covering is to go over and around pipe end, the end of pipe shall be turned to half circle or approximation of a half circle.

5. Pipe shall be washed free of all oil, grease, or other viscous coating.

Fittings. 1. Flanged fittings unless otherwise specified shall be ASME Standard, 125-lb CI.

2. Fittings shall be free of oil, grease, or other viscous coating and excessive rust.

3. Fittings shall be smooth, nonporous, and free from high or ragged projections as well as large, deep, or irregular pits.

4. Alignment of boltholes shall be as in 2 under Alignment, Tolerances, Finishing, except that, when offset boltholes are specified, tolerance shall be from offset as given on print.

5. Inside edge of fittings shall be finished as set forth in 4 under Alignment, Tolerances, Finishing.

General Standards. These specifications cover details pertinent only to rubber lining. Such details as are standard commercial practice have been omitted. The vendor shall be governed by latest modification of ASME and API concerning such details. Departure from such details shall be made only upon written permission of purchaser.

SCOPE OF THE RUBBER-LINING PROCESS

The rubber-lining process is sufficiently flexible to permit application to a wide variety of equipment under a wide variety of circumstances. If equipment is suitable for rail transportation, the usual procedure is to ship the metal parts to the rubber factory. Size is the controlling factor, and as a general statement, the limitation is 11 ft extreme width by 11 ft extreme height, although in certain cases it is possible to exceed these limitations. In case of doubt, definite clearance limitations should be obtained from the railroad.

If equipment is too large for shipment or if it can be more advantageously installed prior to rubber lining, materials and operators can be sent to the job site and the rubber lining installed in the field. In such cases, the purchaser has to defray the traveling and living expenses of the operators and has to provide at his own expense certain services and facilities which are given in detail below:

Common Labor. Purchaser shall provide such common labor as may be necessary for unloading, storage, and handling of materials, etc.

Storage of Materials. Purchaser shall provide suitable storage space for all materials required for the lining operations. This storage space should be provided as close as possible to equipment which is to be rubber-lined. Temperature of storage space should be maintained at 65°F minimum in order to ensure proper workability of materials.

Workroom and Cutting Table. Purchaser shall provide suitable workroom equipped with proper light, heat, and ventilation and a cutting table for handling of stock, which should be approximately 4 ft wide by 34 in. high by 20 ft long. (For special stocks it is sometimes necessary to have pipe coils installed under the worktable to maintain a temperature on the top of the table of approximately 125°F.) The workroom shall be located as close as possible to the equipment to be rubber-lined. The workroom temperature shall be maintained at 65°F minimum. Room for storage of materials and workroom shall be combined if possible in order to facilitate handling of materials.

Protection of Tanks. If equipment to be rubber-lined is located out of doors in certain latitudes and seasons, it will be necessary to protect the equipment from the weather and to provide heat necessary to maintain a minimum temperature of 65°F. In any event, the equipment must be properly protected against rain and in warm weather from direct exposure to the sun. A satisfactory installation cannot be made if conditions are such as to induce condensation on surfaces prior to cementing and lining. Neither can lining be properly cured unless equipment is protected against excessive heat loss during cure. Reasonable comfort conditions are necessary to ensure proper efficiency of operators and proper quality of the work. The purchaser shall provide such protection and heat as may be necessary depending upon local conditions.

Ladders and Staging. Purchaser shall provide ladders and suitable staging which are to be erected to conform to proper working and safety requirements. Purchaser is also to supply such common labor as may be necessary for dismantling and removal of staging, etc.

Sandblasting. Purchaser shall provide for sandblasting of all surfaces to be rubber-lined to remove all scale, rust, grease, etc., prior to rubber-lining. Sandblasting should be done immediately prior to the start of rubber-lining work.

Ventilating Equipment. Purchaser shall provide a suitable suction blower with nonsparking motor and necessary exhaust duct to discharge solvent fumes a safe distance from the tank.

Electrical Connections. Purchaser shall provide necessary electrical outlets adjacent to equipment to be lined and necessary electricity for lights, ventilating equipment, and testing apparatus.

Steam for Vulcanization. Purchaser shall provide necessary steam for vulcanization and necessary piping for same.

Compressed Air. Purchaser shall provide compressed air with necessary piping if required.

Steel-tank Specifications. Steel tank or other equipment furnished by purchaser for lining shall conform to standard specifications of design and finish required for proper installation of rubber lining.

Safety Regulations. Purchaser shall be responsible that all safety precautions consistent with best safety practice are maintained. The equipment to be rubber-lined must be kept clear of any fire hazards such as sparking motors, open flames, etc. No welding or burning operations are to be permitted in the area adjacent to the job during the installation of the rubber lining. Proper precaution must be taken to prevent any hazardous operation within the area adjacent to the job site. Manholes for access to closed tanks must have an inside diameter of 18 in. In some tanks it is also necessary to provide extra openings for installation of ventilating equipment.

Brick Linings. Rubber-lined equipment is also adaptable to the installation of inner brick linings of acidproof brick and cement to handle conditions involving excessive temperature or exposure to mechanical abuse. Brick linings are installed in the field subsequent to erection or installation of rubber-lined equipment. Thickness and specification will vary with the service requirements.

REPAIRS

Of considerable importance to the extensive use of rubber linings is the fact that repairs may be readily made. Minor repairs are usually made by regular maintenance personnel. Extensive repairs are best handled by experienced operators furnished by the rubber manufacturer. Repair materials of reasonable stability are available and may be stored for considerable periods if protected from extremes of temperature and will keep indefinitely if refrigeration is available.

The repair procedure is essentially as follows:

Surface Preparation. If metal is not exposed or damaged, the surface of existing rubber lining should be buffed so as to expose a fresh rubber surface over the entire area to be patched. If the metal is exposed or damaged or if the existing lining is blistered, the damaged rubber in the area to be repaired must be removed. Enough rubber should be removed to be certain that the lining remaining is securely bonded to the metal. The metal must be thoroughly cleaned and roughened, and the existing rubber lining should be beveled and roughened back for a distance of about three inches around the exposed metal. Sandpaper, power buffer, wire brush, file, etc., should be used as necessary. If a repair is necessary where the metal has been eaten through, the rubber must be cut back sufficiently to allow welding of a repair in the metal. If the welding is carefully done and the area not heated too much, it is usually necessary to cut the rubber back about six inches from the weld. After this has been accomplished, it will then be necessary to cut back the rubber all around the repaired area until a good bond is reached. Adjacent rubber surfaces should be buffed so as to expose a fresh rubber surface for a distance of 2 in. back from the exposed metal.

Cementing. Cements should be applied to the surfaces to be patched by brushing on evenly, allowing sufficient time between coats to allow complete drying. Drying time may be accelerated by exposing surfaces to a jet of compressed air. From three to four coats of cement will be necessary, depending upon manufacturer's specification and instruction.

Freshening. The cemented surface should be freshened lightly with high-test gasoline. The freshened surfaces should be allowed to dry until tacky, but not wet or slippery.

Application of Rubber Stock. 1. The edges of the stock should be skived to obtain correct construction of the finished job. The type of skiving will, of course, depend on whether the sheet is adjacent to a corner, has an overlapping adjacent sheet, or is a small repair to an existing lining.

2. The rubber should be rolled and stitched carefully to remove all air and to obtain contact with the cement throughout, working from the center of the sheet outward to the edges.

3. After the lining has been completely installed, or patches rolled in place, it should be looked over carefully for separated seams, "pulls," etc.

Cure. Exhaust steam should be introduced with perforated pipes, jets, etc., as necessary to get contact with all surfaces If only a small repair is being cured, the steam can be confined to the repaired area by means of a tarpaulin, box, or other enclosure. At least 24 hr in exhaust steam will be required to cure the lining, and more time may be necessary in a large tank or if the heat losses are large.

The above outlines the general repair procedure. Detail will vary with materials and instructions of various manufacturers.

COMMON TYPES OF RUBBER-LINED EQUIPMENT

Storage Tanks (Fig. 11-38). Rubber-lined storage tanks of all types are in general industrial use. The most usual types are the horizontal cylindrical closed type and the vertical cylindrical type, either open top or with cover. Horizontal cylindrical tanks may be provided with dished heads or flat heads and will be suitable for pressure or gravity discharge, depending upon detail of construction. Access is provided through a manhole (18 in. minimum), and flanged openings for various piping connections are provided as necessary. The usual capacities vary from 2,000 to 50,000 gal, although size is not a limiting factor.

This type of tank has many advantages. It is applicable to pressure discharge; it is totally enclosed, thereby affording complete protection from mechanical damage to the rubber lining. It is easy to transport and to install. It is easy to support, and all exterior surfaces are readily accessible for inspection and maintenance. Standard dished heads are readily available, and fabricators are equipped to roll the required longitudinal sections.

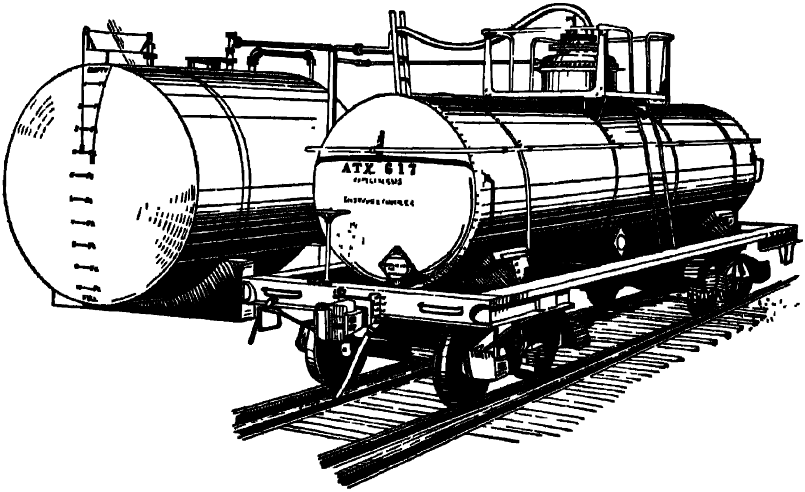

FIG. 11-38. Typical storage tank.

The vertical cylindrical type is usually supplied with a flat bottom, and the cover, if any, is usually of the flat removable type. This type of construction is not suitable for pressure discharge. It is, however, of simple construction, and for smaller sizes up to 10,000 gal is very common. It is readily handled, transported, and installed, although, unless supported on a flat slab, the problem of support may be somewhat complicated In the larger sizes, this type of tank lends itself to field erection, the principal problem being to provide a self-supporting cover for the large diameters Tanks 20 by 20, 30 by 30, and 40 by 40 ft erected and rubber-lined in the field are not uncommon.

Process Tanks. For numerous types of process tanks rubber lining is recognized as standard equipment. While it is not possible to describe each instance in detail, some of the principal uses are as follows:

Plating Tanks (Fig. 11-39). For plating-tank service rubber has several important characteristics. In the first place, most plating is done from an acid solution, and rubber provides the necessary resistance to chemical corrosion. In the second place, rubber being a nonconductor, prevents current leakage. In the third place, rubber prevents metallic contamination, such as might readily occur if metallic linings were used. It is of course important that the rubber contain no materials which can be leached out by the solution and which would poison the bath and brittle, discolor, or otherwise affect the quality of the plate. The rubber must also be of proper and uniform density and composition to prevent plating through, with resultant deposition of the metal on the surface of the rubber. Plating tanks may be of any size or design, and the design detail may include rubber-covered anode supports, overflow troughs, piping connections, etc.

Pickling Tanks (Fig. 11-40). Pickling is usually accomplished by immersion of metal in a hot-acid bath (235°F maximum) and is severe service not only on account of high temperature but also on account of the possibility of mechanical abuse. Rubber-lined steel tanks protected with inner linings of acidproof brick and cement are commonly used in this service. The steel tank provides a shell of adequate strength and rigidity. The rubber lining protects the steel against the corrosive action of the hot acid, and the brick lining protects the rubber from the excessive temperature and mechanical abuse. If high temperature and possibility of mechanical abuse are not present, inner brick linings are not required. Various sizes and types of rubber-lined pickling tanks are in service, varying from small batch tanks to large multiple tanks placed in series for continuous pickling operations.

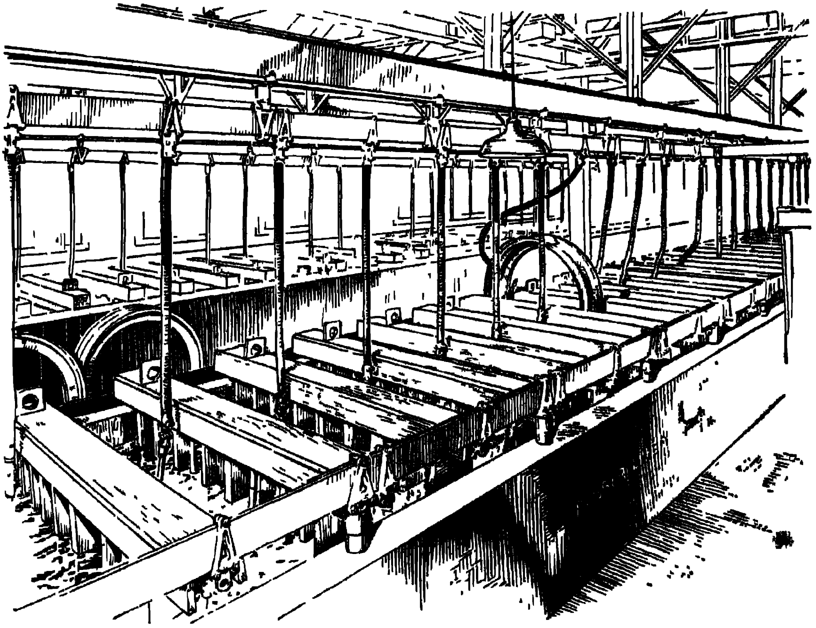

Fig. 11-39. Plating tank.

Bleach Tanks. Rubber linings have excellent resistance to chlorine and to bleach solutions, and rubber-lined tanks are used extensively in batch and continuous bleach operations. These tanks will vary in size and design in accordance with requirements. An outstanding application is the use of vertical cylindrical tanks, approximately 15 ft diameter by 60 ft high, in use in connection with the continuous bleaching of paper pulp.

Ion-interchange Tanks. Many ion-interchange tanks are rubber-lined owing to the fact that either the solution being treated is corrosive, or acid backwash is necessary to revivify the interchange agent. Conspicuous examples are in the treatment of sugar solutions, formaldehyde, and water purification.

Evaporators and Crystallizers. In recent years there has been a very considerable increase in the use of rubber linings for evaporators and crystallizers, particularly for the recovery of salts from waste solutions. Rubber linings are admirably suited for this service, having adhesion adequate to withstand full vacuum and the corrosive effect of the solution and further providing excellent abrasion resistance at the point of discharge.

Crystallizing Tanks. Rubber-lined crystallizing tanks are in common use for the separation of crystalline materials from corrosive mother liquors.

Mixing Tanks. Rubber-lined mixing tanks are used for the addition of acids to various solutions. In such cases, if commercial 66 Baumé sulfuric acid is used, it should be borne in mind that this material has a high specific gravity and tends to stratify and settle to the bottom of the tank. Strong sulfuric acid has a destructive effect on rubber. Hence in all cases stratification must be prevented by air or mechanical agitation. Also, the addition of strong sulfuric acid results in liberation of considerable heat, and steps should be taken to keep the maximum temperature within reasonable limits.

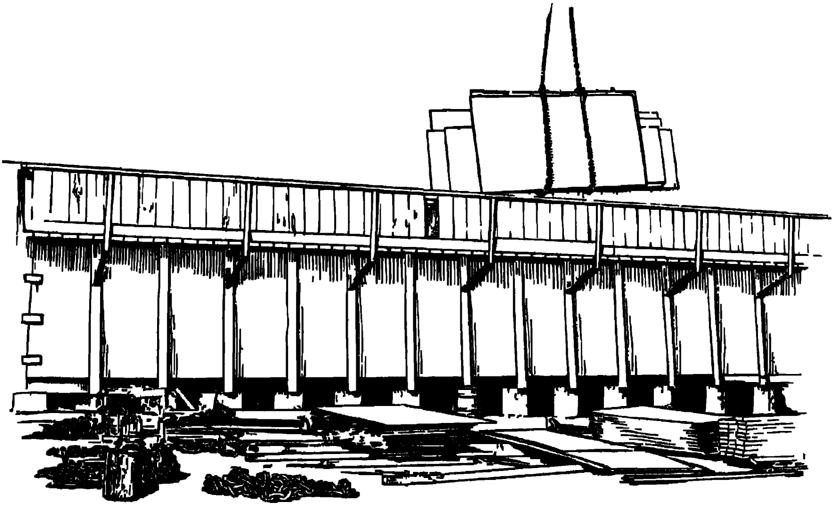

Fig. 11-40. Batch pickling tank.

Scrubbing Towers (Fig. 11-41). Rubber-lined scrubbing towers are frequently used for the absorption of soluble gases. Perforated grid plates can be furnished for support of coke, tile, etc. As the rubber lining is constantly wetted with a film of solution, the rubber is protected from direct contact with gaseous constituents which might ordinarily be destructive to rubber, with the result that rubber linings can frequently be used for service in scrubbing towers under conditions for which rubber would not normally be considered suitable.

Pipe and Fittings (Fig. 11-42). Rubber linings can be installed in piping of all sizes from 1 in up. Standard wrought pipe is ordinarily used, but rubber lining can be installed in fabricated or spiral welded pipe. In the majority of cases, flanged assemblies are used, although special assemblies such as victaulic joints can be supplied. Ordinarily, ASME Standard 125-lb cast-iron flanges are furnished, although any of the various types of steel flanges can be supplied. Welded-on steel flanges are preferable for vacuum service. Rubber lining can also be installed in special bent sections, fabricated manifolds or headers, and other special fabricated sections. Not all such fabricated sections can be handled as there are certain limitations to degree and extent of bends, length, and location of side outlets, etc., which will be discussed below under Engineering Considerations. Several types of rubber-lined valves are available, including gate valves, diaphragm valves, check valves, etc.

Special Equipment (Fig. 11-43). Rubber linings and coverings are adaptable to many types of process equipment, including fans, pumps, continuous filters, filter plates and frames, agitators, screw conveyors, centrifugals, etc. It is good practice to purchase special equipment complete from the equipment manufacturer. He is in better position to make recommendations as to the design, size, etc., required for a given performance. He is better able to modify design for incorporation of rubber if such modification is necessary. If special machining is required, he would have the necessary equipment to provide such machining. If dynamic balance were necessary, he would be able to provide it. The rubber manufacturer's liability is limited to workmanship and material and does not include responsibility for mechanical performance subsequent to rubber covering. The rubber manufacturer should, of course, confer with the equipment manufacturer as to the proper type of stock for the given service conditions. Rubber-covered equipment of each of the types enumerated above is operating with entire satisfaction under a wide variety of service conditions.

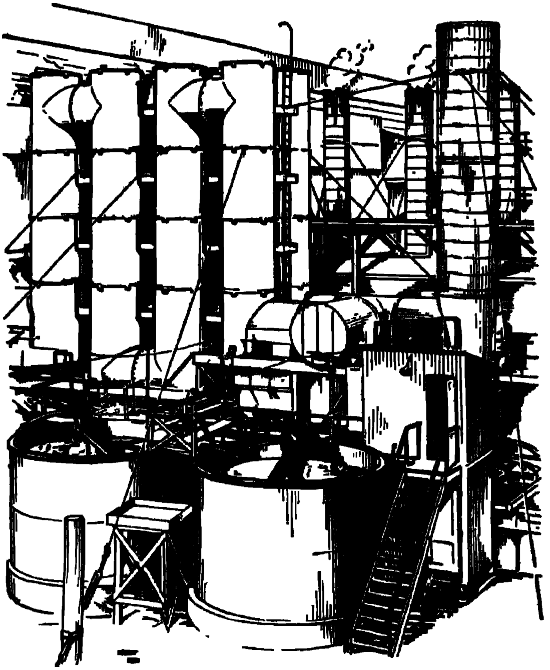

Fig. 11-41. Scrubbing towers and process tanks.

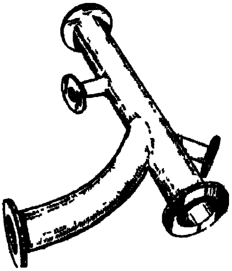

Fig. 11-42. Special fabricated assembly.

Transportation Equipment. Rubber-lined equipment is extensively used for the bulk transportation of corrosive liquids. Rubber-lined tank cars are standard for many services. In all cases, the rubber lining must conform to ICC Specification 103-B for riveted cars and to ICC Specification 103-BW for welded cars, and in each case the rubber manufacturer is required to give the owner a certificate stating that the rubber lining properly conforms to the ICC specification. Rubber-lined truck tanks and rubber-lined barge tanks are also in common use, and if these are engaged in interstate commerce, the rubber manufacturer must also certify that the rubber linings conform to the ICC requirements.

Auxiliary Equipment (Fig. 11-44). In connection with many of the items described above, auxiliary equipment such as covers, hoods, ventilating ducts, grid plates, baffles, stacks, etc., are required. It is common practice to rubber-line and rubber-cover these various items.

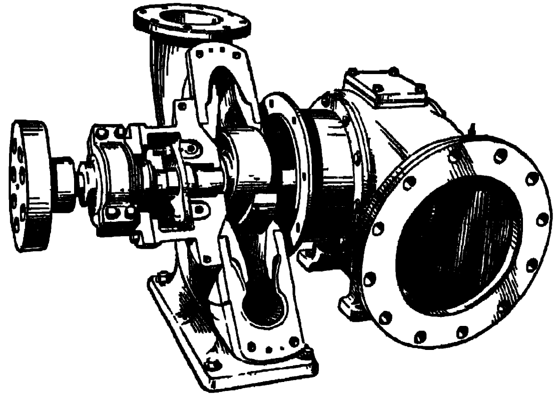

FIG. 11-43. Centrifugal pump.

Thickness of Rubber Lining for Most General Requirements. Standard thickness of rubber lining is $\frac{3}{16}$ in. For severe service conditions or where abrasion is present, $\frac{1}{4}$-in. thickness is recommended, and in certain cases additional thicknesses have been provided. For pipe and fittings up to and including 5-in. diameter, $\frac{1}{8}$-in thickness is standard. For larger sizes, 6-in. and above, $\frac{3}{16}$-in. thickness is recommended. For particularly severe conditions, $\frac{1}{4}$-in. linings should be used.

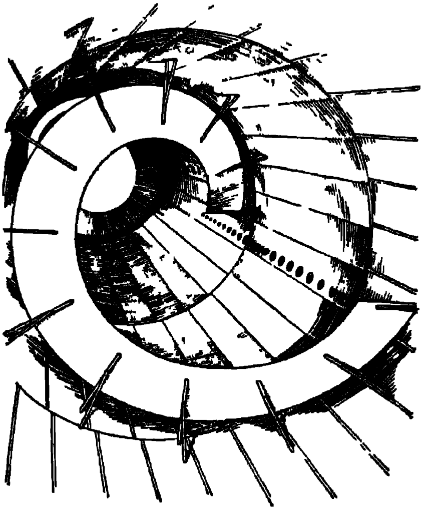

FIG. 11-44. Exhaust duct.

ENGINEERING CONSIDERATIONS

Only special considerations as applying to rubber-lined equipment will be considered. It is understood that, in so far as metal parts and construction are concerned, all details of standard engineering practice will apply and will be provided for.

Tanks. *Pressure Tanks.* In so far as rubber lining is concerned, installation is not ordinarily complicated by pressure consideration.

Vacuum Tanks. If vacuum vessels are to be rubber-lined, it is essential that the steel vessels be airtight. If the vessel is airtight, no difficulty will be experienced with rubber lining. If the metal is not airtight, the rubber lining will blister and pull away from the metal. All vacuum vessels should be vacuum-tested by the fabricator prior to installation of the rubber lining and should be certified. Vacuum test may be applied subsequent to rubber lining, but this is not ordinarily considered necessary if the vessel itself has been pretested. Furthermore, if hard-rubber linings are used, a test of considerable duration is necessary to demonstrate positively any weakness in the construction.

Brick-lined Tanks. If rubber-lined tanks are to be brick-lined, it is essential that adequate provision be made in the brick lining to compensate for expansion and contraction of the brickwork. It is also essential that the steel tank be so designed as to have adequate strength to withstand the stresses set up by expansion of the brick wall. If long tanks are involved, it is preferable that sections be welded together, subsequent to erection if necessary, rather than to rely on bolted flanged joints. Extreme care in design of steel and brickwork is particularly essential if tanks are subjected to low external temperature during operation.

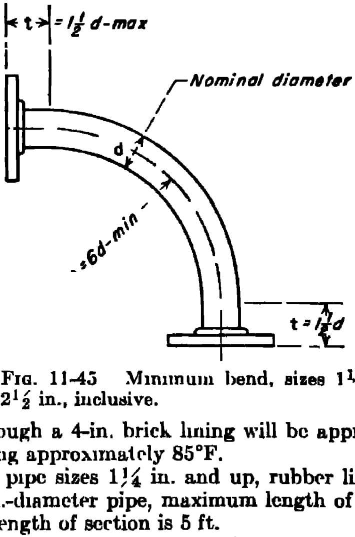

Fig. 11-45 Minimum bend, sizes 1¼ to 2½ in., inclusive.

Temperature Gradient. If temperature of liquid in tank is 212°F and room temperature is 70°F, the temperature drop through a 4-in. brick lining will be approximately 60°F and through an 8-in. brick lining approximately 85°F.

Pipe and Fittings. *Pipe Sections.* For pipe sizes 1¼ in. and up, rubber lining can be installed in 20-ft sections. For 1-in.-diameter pipe, maximum length of section is 10 ft, and for ¾-in. pipe maximum length of section is 5 ft.

Bent Sections. Within limits, rubber linings can be installed in bent sections. From a manufacturing standpoint, the essential feature is that it must be possible to see and to reach all surfaces to be rubber-covered in order to assure uniform bond, absence of blisters, etc. While it is not possible to establish any hard and fast rules, it is possible to indicate the approximate [illegible] Figures 11-45 to 11-47 indicate the safe minimums. If doubt exists as to the possibility of rubber-lining bent sections, the problem should be referred to the rubber manufacturer.

Fabricated Headers. Space considerations, reduction in the number of flanged joints, ease of installation, or other factors often make the use of fabricated headers, manifolds, or other special fittings with side outlets desirable. It is possible to rubber-line many such constructions. Here again, it is not possible to establish any hard and fast rules. The comments given above with regard to visibility and accessibility of areas to be rubber-covered apply. Also, it is possible to give some indication as to maximum limits, which are detailed in Figs. 11-48, 11-49, and 11-50. Again, constructions involving doubt as to the possibility for rubber lining should be referred to the rubber manufacturer.

Flange Construction. Details of handling rubber lining over flange faces will vary somewhat with different manufacturers. With soft-rubber linings, rubber may be brought out over full face of flange or may be brought out to the inside of boltholes

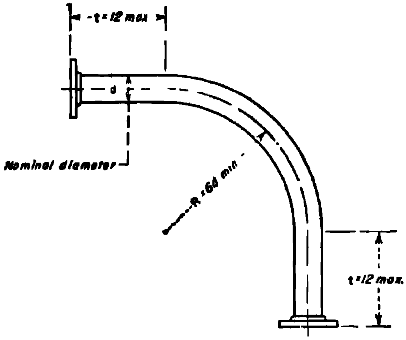

Fig. 11-46. Minimum bend, sizes 3 in. and over.

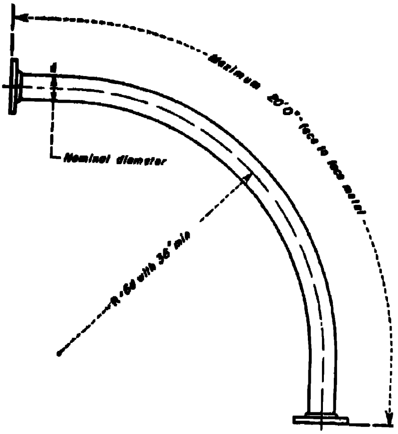

Fig. 11-47. Minimum bend, bar length, sizes 3 in. and over.

only. With the latter construction, noncompressible ring gaskets are used. This construction has the advantage that maximum compression which can be applied to the rubber lining is definitely limited. Figure 11-51 illustrates this construction. With hard-rubber lining, the rubber is brought out over the full face of the flange, and assembly is made with pure-gum soft-rubber gasket. It is important that soft gaskets be used in order to prevent damage to the rubber lining. It is also important that the gaskets be cut to proper size. Figure 11-52 details hard-rubber-lined flange assembly and proper gasket specifications and dimensions.

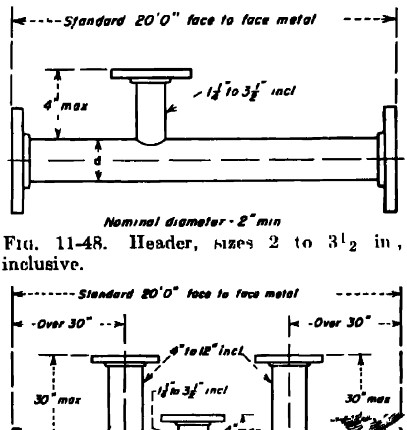

Fig. 11-48. Header, sizes 2 to $3\frac{1}{2}$ in., inclusive.

Gaskets. Necessary assembly gaskets for either soft- or hard-rubber linings are normally supplied by the manufacturer and are included in the purchase price. It is important that these gaskets be used for installation.

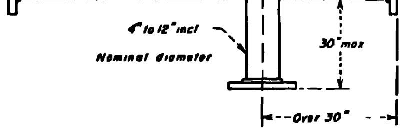

Fig. 11-49. Header, sizes 4 to 12 in., inclusive.

Pressure Limitations. Hard-rubber-lined flanged assemblies can be recommended for 200 lb maximum pressure. Soft-rubber-lined flanged assemblies are suitable for pressures up to 600 lb maximum. Special joints such as victaulic are also suitable for pressures up to 600 lb maximum and possibly for higher pressures under certain circumstances. It is recommended that manufacturers be consulted for recommendation if use of rubber-lined piping under high pressures is contemplated.

Vacuum. Rubber-lined piping for vacuum service is satisfactory provided metal parts are airtight. It is recommended that welded-on steel flanges be used if service under vacuum is required.

Pressure Drop. No comparative test figures are available for pressure dropped through rubber-lined piping as compared to piping of other materials. Experience shows, however, that, if calculations are based on coefficient for new steel pipe, calculations will be sufficiently accurate for installation requirements.

Specification of Section Lengths. As rubber lining is brought out over flange faces, allowance for it must be made in predetermining the lengths of the individual pipe sections. Allowance must, of course, be made for each joint whether of pipe sections or of fittings. In ordering rubber-lined piping, it is usually better to specify the desired over-all length of each section, leaving it to the rubber manufacturer to make proper allowance for lining and gasket. Purchaser can, of course, predetermine exact metal length should he so desire, but in such cases, proper joint allowance should be obtained from the selected rubber manufacturer as specifications of different manufacturers vary slightly, and it is important that proper allowance be used.

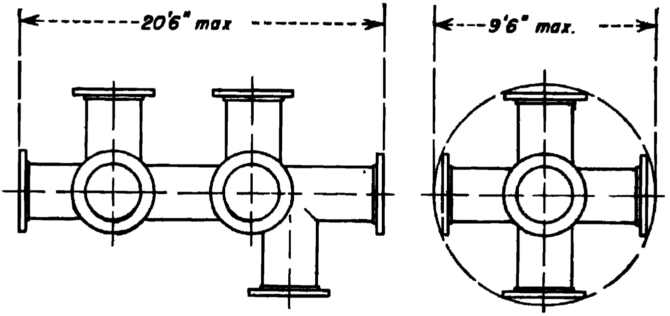

FIG. 11-50. Header, sizes 14 to 24 in., inclusive.

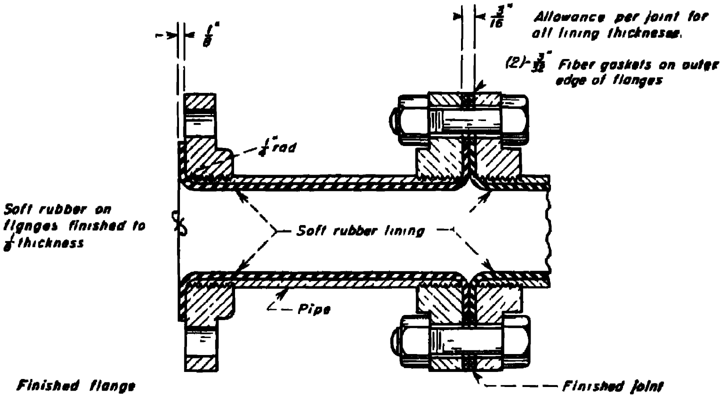
1/8"
3/16"
Allowance per joint for all lining thicknesses.
(2)-3/32" Fiber gaskets on outer edge of flanges
1/4" rad
Soft rubber on flanges finished to 1/8 thickness
Soft rubber lining
Pipe
Finished flange
Finished joint

FIG. 11-51. Flange construction, soft-rubber lining.

Allowance for Expansion and Contraction. It is important, particularly with large-diameter piping or where long runs are involved, that provision be made in installation to compensate for expansion and contraction. Stress is, of course, transmitted to the rubber over the flange faces, and it is important that this stress be held below the maximum allowable for the material. Requirements will vary depending upon whether soft- or hard-rubber linings are used.

Stress in Soft-rubber Linings. If it is desired to use spring in the line to compensate for temperature changes, the line may be computed as a square corner unit. (The Mitchel method is sufficiently accurate for the temperature changes encountered.) Due consideration must be given to the weight of the pipe and resistance to bending afforded by the supports. In the case of soft-rubber linings, steel stresses should not exceed those specified in Code for Pressure Piping, American Tentative Standard. If construction described above employing noncompressible gaskets is used, it should be remembered that centroid of eccentric thrust under bending movement is moved outward on to noncompressible gasket.

Stress in Hard-rubber Linings. With hard-rubber linings, the limiting value of pressure to ensure absence of shear on inner face of flange due to plastic flow and tightness after permanent set of soft-rubber gasket should be 300 psi on periphery of flange. This corresponds to stress in steel, figured as continuous tube, of 5,000 psi. This stress limitation will ensure that the original bolting pressure will not be exceeded and will eliminate loosening of joint and possibility of cracking of rubber at inner face.

Relief of Strain. Strain due to expansion and contraction must be relieved. In the smaller sizes for two- or three-dimensional layouts, no great strain may be present because of springing in the line, but straight runs between fixed dimensions must allow for thermal changes. This may be done by means of expansion joints, several types of which are available. Where relief is required in two- or three-dimensional lines, this may be obtained by the use of a section of flanged wire-reinforced hose. The bend in the hose should be uniplanar. The hose manufacturer should be consulted as to proper length and specification. Each length is a separate layout problem, depending upon amount of movement in the line, angle of intercept, and other service conditions.

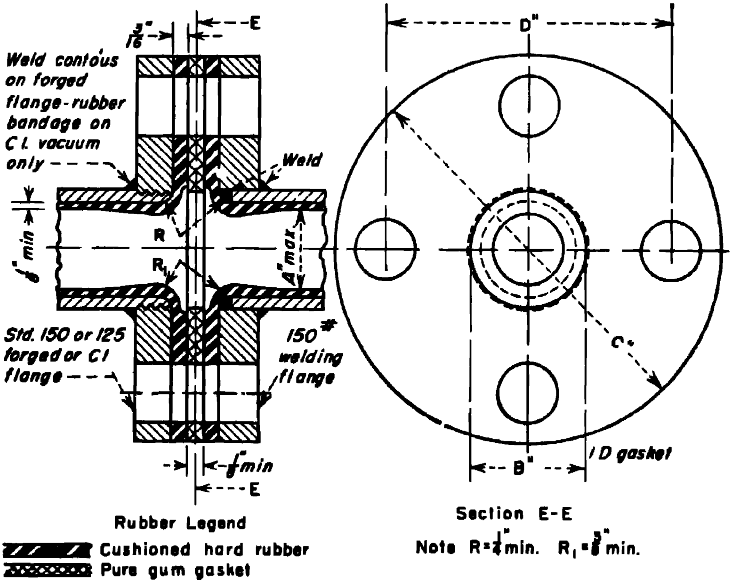

Fig. 11-52. Flange construction, hard-rubber lining.

Installation. It is important that reasonable care be exercised in the handling and installation of rubber-lined piping. Piping should be protected from oils and rubber solvents and from extreme rapid changes in temperature. Rubber-lined piping should first be brought to proper elevation and alignment so that flange is parallel to companion flange. Gaskets as recommended and supplied by the rubber manufacturer should be used. Bolts should be inserted and brought up evenly all around, using standard-length spud wrenches for the size of bolt installed without extension arm or hammer. After assembly, tests should be made with noncorrosive liquid at working pressure and the line inspected for tightness. If any particular joint requires excessive tightening, it should be opened and examined for the cause of the difficulty. Lines should be watched closely for leaks during the first week of service and bolts tightened where necessary. Inspections should be made at regular intervals thereafter.

Supports and Anchors. Supports should be designed to eliminate torque and excessive bending and to allow for deformation under temperature change. Tension due to bending is eliminated by proper anchorage and bends. Compression due to thrust should be minimized by expansion joints. Torque can be eliminated by proper blocking along line of thrust. Each piece of pipe should have an adjustable support, and any piece over 12 ft long should have two supports. Hangers should be long enough to allow for movement without undue restraint when spring in line is relied upon for temperature correction. Roller supports or roller hangers are preferable; where vibration is present and hard rubber is to be used, especially at low temperatures, spring hangers should be used with lock washers under the flange bolts. Spring hangers should also be used on large-size line where there is danger of buckling due to uneven temperature at top and bottom of line. This may occur especially in lines subjected to intermittent service at partial capacity with high temperature differential between the liquid and the outer atmosphere.

Special Equipment. If machining of special parts is required subsequent to rubber covering, this is usually best accomplished by the equipment manufacturer. Hard-rubber and semihard-rubber stocks are available which have proper machining qualities.

High-speed equipment, such as fans, centrifugals, etc., should be checked for dynamic balance subsequent to rubber covering. This is best accomplished by the equipment manufacturer on the same equipment used by them for initial dynamic balance. Small balancing pads are commonly vulcanized integral with the main rubber covering, and dynamic balance can be adjusted by cutting away a portion of this rubber pad. The equipment manufacturer should also be responsible that the design is such as properly to accommodate the rubber and that the construction is adequate to handle the extra weight introduced by the addition of the rubber covering.

CARE OF RUBBER-LINED EQUIPMENT

Users of rubber-lined equipment should observe the following simple rules for care and maintenance:

1. All rubber-lined equipment should be inspected at regular intervals.
2. Rubber linings should be protected from oils, greases, and rubber solvents.
3. Rubber linings should also be protected against mechanical abuse. Hard-rubber linings, particularly at low temperatures, should be protected against hammer or other external shock. If necessary to enter a tank by means of a ladder, the feet of the ladder should be protected with burlap or similar material. Inspectors should preferably wear rubber shoes. Measuring sticks should have blunt ends to prevent damage to rubber. Care should be taken not to drop sample bottles into rubber-lined tanks.
4. Inspection lamps should have proper guards. Under no circumstances should a bare light bulb be allowed to remain in contact with the rubber.
5. Hard-rubber linings should be protected against thermal shock. Steps should be taken to ensure gradual heating or cooling.
6. If it is necessary to keep rubber-lined tanks or piping out of service for considerable periods, it is better to fill them with dilute hydrochloric acid solution, care being taken to protect solution against freezing.
7. Rubber-lined equipment should not be transferred from one service to another without checking with rubber manufacturer as to the suitability of material to proposed change.
8. Certain solutions such as ferric chloride tend to coagulate in cold weather. Whenever necessary to warm solution to permit its removal, such warming should be gradual and should be continued only to minimum point required for liquefaction.
9. Exposed metal surfaces should be carefully protected against corrosion.
10. Necessary repairs should be made immediately to prevent extension of damage to rubber or metal.

SECTION 12

WELDING

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ELECTRIC WELDING IN PLANT DESIGN, OPERATION, AND MAINTENANCE

BY CHARLES G. HERBRUCK

Assistant Secretary, The James F. Lincoln Arc Welding Foundation, Cleveland, Ohio

Electric welding is a process of fusing metal through the application of heat that is produced by electricity. Since its first practical applications during the First World War, developments and refinements in techniques and equipment for welding have produced a process that is basically simple, yet highly versatile. In every industry that uses metal, welding has become an important tool of plant management. Welding is one of the basic tools in such work as building new structures or repairing or enlarging existing structures, installing or maintaining piping systems of all kinds, the fabrication of sheet metal into containers, tanks, ventilating ducts, and other conveniences, repairing broken machinery parts, restoring worn parts by building up the wearing surfaces, making replacement parts for broken castings, and the building of special machinery and equipment.

There are two general classifications of electric welding processes: electric arc welding and resistance welding. Electric arc welding is the process most useful in plant operation and maintenance. Resistance welding is chiefly employed in production operations. Its use is limited by the nature of the process and the equipment required. The data in this section, therefore, will deal largely with the techniques and applications of arc welding.

RESISTANCE WELDING

Resistance welding is the process of joining two or more metallic parts to form an assembly, the heat for the formation of the weld being created by the electrical resistance of the pieces being welded when electric current is supplied by the welding machine. Simply stated, it is a heat and squeeze process The parts to be welded are heated to point of fusion by the passage of electric current through them. When the welding heat has been reached, pressure is applied mechanically and the electric current is broken. This mechanical pressure and heat, generated by the electric current passing through the resistance of the materials being welded, melts and fuses the faying surfaces of the two parts. As the metal cools, the weld is formed.

The heat being generated in the weld is expressed by the formula

$$W = I^2Rt$$

where W is heat in watt-seconds, I is amperes flowing through the weld, R is the resistance in ohms, and t is the time in seconds. Thus, resistance factors affecting the amount of heat being produced by a given welding current for a unit of time are

1. The electrical resistance of the materials being welded
2. The contact resistance between weldments as determined by surface condition, scale, welding pressure, etc.

A great many types of metals have been satisfactorily welded with this process. Weldability is determined by the electrical resistance of the metal. Low-carbon steel is easily welded, while copper and aluminum present some difficulties.

The most important types of applications in the field of resistance welding are spot welding, butt welding, flash welding, and seam welding. The spot weld is made by overlapping the parts and gripping the overlapped sections between two electrode points through which the current is passed and pressure applied to make the weld in a single spot. The butt weld places the parts to be welded end to end, to be electrically heated and squeezed together. The flash weld is an adaptation of the butt weld.

The seam weld is similar to the spot weld except that a circular rolling electrode is used to produce the effect of a continuous seam. A special spot welding operation known as "shot welding," employing carefully controlled short time intervals and high currents, is often used for special alloys such as the "stainless" group.

Because of the requirements of special equipment and the critical control that must be maintained, resistance welding is best suited to mass-production operations in sheet metal less than ¼ in. in thickness. Its limited use in plant operation does not generally warrant the purchase of equipment for that purpose only.

ELECTRIC ARC WELDING

Arc welding is the process of fusing metal through the heat generated by an electric arc. The process has been developed and refined extensively, and today there are a number of methods of employing the arc in welding. The metallic arc process is the one which has the widest general use in plant operation and will, therefore, receive major treatment here. Before discussing the metallic arc, some of the other methods should be given brief consideration:

Atomic-hydrogen Arc Welding. In the atomic-hydrogen welding process an arc is established and maintained between two tungsten electrodes, while a stream of hydrogen gas is passed through the arc. As this hydrogen reverts to molecular form in the arc, an intense heat is given off which is used to fuse the metal being welded. The process is much the same as welding with a gas torch, and while it is more costly than other processes, it has some advantages in welding thin sheets where a high finish is required. A filler rod is used when extra weld metal is needed.

Inert-gas Arc Welding (Helium, Argon). This process is similar to atomic-hydrogen welding. It differs in that a single tungsten electrode is used to form the arc directly between the electrode and the base metal, and the inert gas is used only for protection, not heat. A filler rod is used, and the inert gas supplied through an annular opening in the insulated tube surrounding the electrode protects the metal during the welding.

Carbon Arc Welding. In this process the arc is formed between the work and carbon electrode. It is a puddling operation and cannot be used in vertical or overhead welding. The heat of the arc between the carbon and the work melts a small pool which is kept molten by playing the arc across it. A filler rod is used to add extra metal. The carbon arc is used successfully in welding aluminum and copper, and can be used economically as a cutting tool where a smooth cut is unnecessary.

Metallic Arc Welding. In the metallic arc process the current, either alternating or direct, which is generated in the welding set, is used to create an arc between a metallic wire and the work to be welded. The intense heat of the arc itself, which is said to be the hottest known commercial source of heat, brings the work to the melting point almost instantly. The end of the wire is also melted, and the tiny globules of metal formed on the tip of the wire are forced across the arc. These globules and the molten base metal combine and, when cool, solidify into a homogeneous mass. The globules of metal are actually forced across the arc stream and not merely dropped by gravity. It is this arc force which makes overhead welding possible with the metallic arc process.

Practically all welding with the metallic arc is done with what is called the shielded arc. In shielded arc welding the metal wire, or electrode as it is called, is covered with a chemical coating. In the heat of the arc, this coating burns and melts and forms a cloud of protective gas, completely enveloping the arc and molten pool in the base metal. The coating also creates a slag which protects the weld during cooling. The coating and the cloud and the slag which are formed by it have several functions which improve the quality of the welds made. The coating controls the chemical analysis of the weld by preventing the vaporization or oxidation of essential elements, such as manganese; in some cases additional elements not present in the core wire can be added by means of the coating. The primary purpose of the coating, however, is to form the protective atmosphere, and the slag, which acts as a fluxing agent, controls cooling rates and the shape of the bead and protects the weld from harmful oxides and nitrides present in the ambient atmosphere. The coating also provides better arc characteristics such as ease in striking and maintaining the arc, increased speeds, and better penetration (see Fig. 12-1).

Welds made with a shielded arc on mild steel have a tensile strength of 60,000 to 75,000 psi, which is 20 to 50 per cent higher tensile strength than possessed by welds made with an uncoated electrode. The ductility of welds made with a shielded arc averages 100 to 200 per cent greater than those made with a bare electrode. The resistance to corrosion of shielded arc welds is greater than mild steel itself.

Equipment. The chief quality desired in welding equipment for general plant use is flexibility. A welding set that generates d-c welding current will be found to have the widest range of applications. These motor-generator sets are available powered either by alternating or direct current or internal-combustion engines. The set should be easily portable. A unit with a NEMA rating of 200 amp will provide adequate capacity for all general plant work. Sustained welding in heavy plate, such as might be met in structural work or production work, would demand a 300-amp unit. There is a very limited amount of this type of welding in maintenance operations.

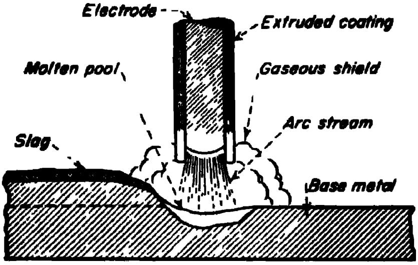

FIG. 12-1. Shielded arc in the process of arc welding.

Material to be welded generally must be formed and cut to the proper shape. A certain amount of other basic machinery therefore is necessary for welding operations. Cutting torches, saws, shears, brakes, and a drill press may be needed, as well as the other small tools generally found in the millwright section of every plant.

Protective clothing should be provided to protect the operator from heat and flying sparks. Leather gloves, apron, sleeves, and shoe covers are suggested where a considerable amount of welding is to be done. Electrode holders, face shields, cable, and a variety of electrodes must be all available wherever the process is used. The arc should not be looked at with the naked eye at any distance closer than 40 ft The infrared and ultraviolet rays from the arc can cause temporary discomfort. The eye burn from the arc rays will not cause permanent injury. In case of burn, the eyes should be washed every few hours with sweet oil. Some shield should be provided, either a portable screen or a permanent enclosure, in rooms where the welding operations are performed.

Welding operators for general plant work should be good general all-around welders. They should be trained in fundamentals of welding as well as a wide variety of techniques. It is well for operators to be familiar with blueprint reading, basic welding metallurgy, use of welding symbols, and other allied subjects. A mechanic who is used to working with tools generally requires only a few hours of training to be able to hold an arc and deposit a satisfactory weld for simple applications. Experience and imagination gradually build an operator to the point where he can perform difficult applications and develop the full possibilities of using welding.

Welding is a basically simple process and easy to understand. Its constantly expanded use over the past 25 years has accumulated a fund of information that appears to be complex and confusing to the person first approaching the subject. The process, however, requires only a minimum of equipment and training to be first applied, and the fuller understanding of its uses comes as its use is extended.

TECHNIQUES OF WELDING

In the case of welded joints in mild steel, which comprise roughly 95 per cent of all welding, the weld is actually stronger and has a much higher elastic limit than the plates joined. Therefore, a weld correctly made will never fail in actual service, as the failure will first occur in the structure itself. This can only be said, however, of welds that have been correctly designed and properly made.

The various technical and engineering societies have established codes that prescribe proper welding techniques for making good welds in the various classes of work such as structural, pressure vessels, piping, ships, etc. The following material will undertake a consideration of the general factors that are the basis for the various codes. The codes should be followed where the requirements of the work warrant it. An understanding of these basic factors is essential to interpreting the codes.

Good welds are the result of the proper selection of

1. Type of joint
2. Type and size of electrode
3. Welding procedure
4. Base metal of good weldability

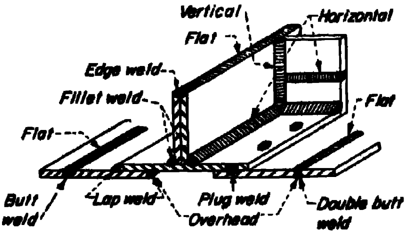

Fig. 12-2. Composite view of types of welded joints and their location.

Types of Welded Joints. The selection of the type of welded joint to be used in a particular application is governed by three factors:

1. The load and its characteristics; *i.e.*, whether the load is in tension or compression, and whether bending, fatigue, or impact stresses in any combination are present
2. The manner in which the load is applied, *i.e.*, whether load application is steady, variable, or sudden
3. The cost of the joint preparation and welding

The selection of the proper joint for economical welding is important in designing structures, machinery, or equipment. Maintenance welding on the other hand frequently allows no choice of joints. Understanding the principles of good joint selection is important, however, to safe and economical welding practices in all applications.

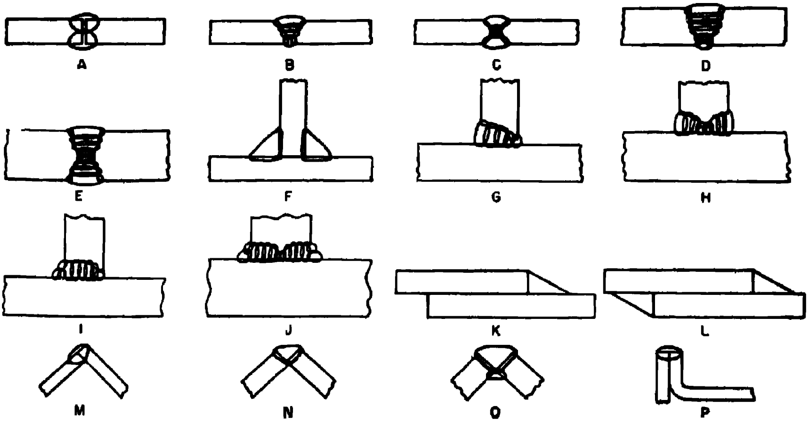

Fig. 12-3. Types of welded joints and variations. (*A*) Square butt joint,* (*B*) single-V butt joint,* (*C*) double-V butt joint,* (*D*) single-U butt joint, (*E*) double-U butt joint, (*F*) square T joint,* (*G*) single-bevel T joint,* (*H*) double-bevel T joint,* (*I*) single-J T joint, (*J*) double-J T joint, (*K*) single-fillet lap joint,* (*L*) double-fillet lap joint,* (*M*) flush corner joint, (*N*) half-open corner joint, (*O*) full-open corner joint, (*P*) edge joint.*

The classification of joints and their common names are shown by Fig. 12-2. A brief discussion of the joints most commonly used in maintenance welding and their application to given service conditions follows:

The square butt joint (Fig. 12-3.1) is suitable for all usual loads and requires full and complete fusion, particularly when load is of fatigue or intermittent nature. The base metal for this type of joint must be good weldable steel since a large portion of the base metal is melted during the welding. The thickness of plate on which the square butt joint is used is generally ⅜ in. or lighter although this type of joint has been used on other plate sizes. Preparation for welding this joint is simple, requiring only a matching of the edges of the plate, separated by a distance dependent upon the plate thickness. Because of the simple preparation, the square butt joint is low in cost.

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* Joints most commonly used in maintenance welding.

The single-V butt joint (Fig. 12-3*B*) is suitable for all usual load conditions. It is generally used with plate thickness considerably greater than the square butt joint—⅜ in. or heavier—although its use on thinner plate is not unusual. Preparation is more costly than the square butt joint, and more electrode is used in welding.

The double-V butt joint (Fig. 12-3*C*) is suitable for all usual load conditions. It is used for plates of greater thickness than the single V and for work which can be welded from both sides. Cost of preparation for welding is higher than for single-V butt joint, but double V requires approximately half as much electrode. Cost of machining should be weighed against the cost of welding and the joint selection made accordingly. Warpage can be reduced by alternating the beads, welding on one side and then the other to keep joint symmetrical during welding.

The square T joint (Fig. 12-3*F*) corresponds to the square butt joint in that no machining of plates is required. The square T is used for all ordinary plate thicknesses, principally for loads that place the welds in longitudinal shear. For severe impact or heavy transverse loads, the nonuniform stress distribution of the joint should be kept in mind and the stress intensity of the application duly considered. The square T requires considerable weld metal, and therefore the welding cost may be higher than other types of T joints.

The single-bevel T joint (Fig. 12-3*G*) is suitable for much more severe loads than the square T, owing to its better distribution of stress. It is employed in most instances for welding plates ½ in. or thinner in work that can be welded from one side only. While more costly to machine than the plain T, the single-bevel T joint is lower in electrode costs.

The double-bevel T joint (Fig. 12-3*H*) is suitable for heavy loads in longitudinal or transverse shear in joining heavy plate where welding can be done from both sides. Double bevel is somewhat higher than single-bevel T joint in machining cost but has lower electrode cost than some other types such as plain T.

The single-fillet lap joint (Fig. 12-3*K*) is frequently used and has the advantage of requiring practically no machining to fit the edges of the plate. When fatigue or impact loads are encountered, stress distribution should be carefully studied. Where loading is not too severe, the single-fillet lap joint is suitable for welding plate of all thicknesses.

The double-fillet lap joint (Fig. 12-3*L*) is suitable for load conditions much more severe than can be met by the single-fillet lap joint. In general, the two fillets should be full size, although one fillet may be smaller than the other in some instances. Because of its lower cost, the double-fillet lap joint is widely used. For extremely severe loads, the butt joint should be used.

The edge joint (Fig. 12-3*P*) is used in joining plates ¼ in. or thinner for light load . Careful consideration must be given to the load conditions, especially impact and fatigue, as this type of joint is not suitable for severe loads.

Selection of Electrodes. There are two aspects of the question to consider in deciding which electrode to use: the general type of wire and the proper size diameter of wire. The selection of the correct size of wire will be considered under Arc Speed The following material indicates information that must be known for selecting the proper type.

To tabulate a guide for correct selection of electrode for all the innumerable combinations of various conditions is impractical. Each job must be analyzed and the following factors weighed in relation to each other for their relative importance

1. The position in which the work must be welded
2. The type and thickness of the base metal being used
3. The preparation of the work, which affects fit-up
4. The type of available welding current

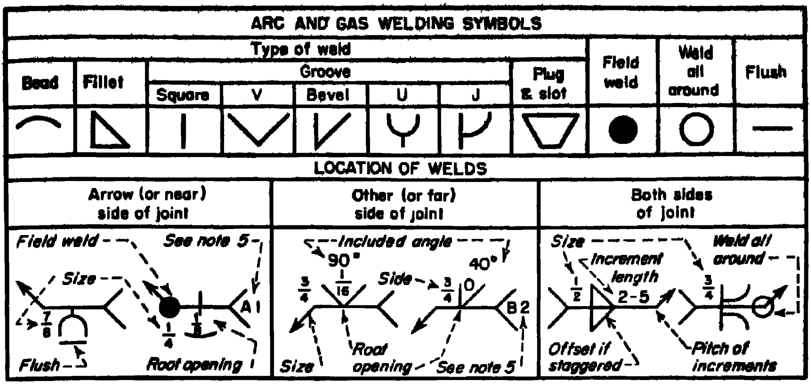


FIG. 12-4. Legend for use on drawings of symbols to indicate type of weld as specified by American Welding Society.

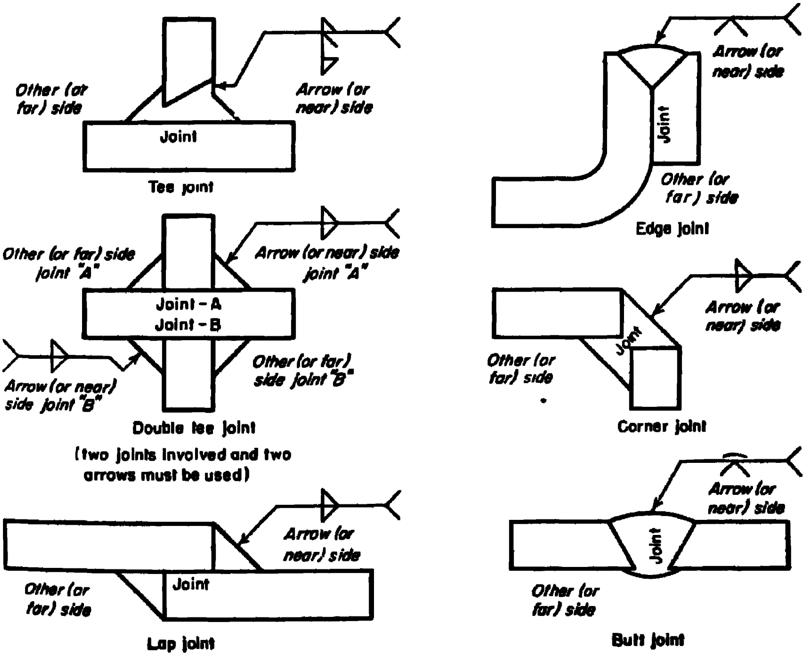


FIG. 12-4A. Location of line of joint in the various fundamental types of joints.

5. The class of work, *i.e.*, whether deep penetration or surface quality is the chief essential

The American Welding Society has established specifications for the manufacture of electrodes which have become the standard in the industry. Manufacturers publish complete information on the composition of their electrodes and the correct procedure for their use in compliance with AWS specifications. All this information should be readily available for reference, and every plant should be familiar with it.

The American Welding Society published in 1948 new specifications for steel and copper electrodes. These are available in booklet form for mild-steel arc-welding electrodes, low-alloy-steel electrodes, copper and copper-alloy electrodes, corrosion-resisting chromium and chromium-nickel steel welding electrodes. This information is available for reference.

Additional information can also be secured from the various electrode manufacturers on cast-iron, hard-surfacing, and nonferrous electrodes.

Welding Procedures. The varying combinations of welding conditions will influence the determination of the correct welding procedure just as they influence the selection of electrodes. The following material considers the general factors that enter into the development of procedures for all applications. They are

1. Type of welding current
2. Polarity
3. Arc characteristics
4. Arc speed
5. Location of welds in relation to strength requirements
6. Distortion

Direct Current and Alternating Current. Welding can be accomplished efficiently with either alternating or direct welding current. Certain factors may necessitate using one to the exclusion of the other, or in some cases certain advantages may be gained.

The type of work being welded should be carefully studied to determine whether alternating current or direct current is better suited to the job. Many stainless steels, low-alloy high-tensile steels, and nonferrous metals cannot be readily welded with alternating current. For certain types of welding an electrode for use with alternating current will cost 1 to 1½ cents per lb more than a comparable electrode for use with direct current. In sheet-metal work or when smaller diameter electrodes are used, the speed of welding is generally higher with direct current. When larger electrodes, over $\frac{3}{16}$ in., are used with higher currents, alternating current will sometimes increase the speed of welding. Power cost and maintenance costs are generally lower with an a-c machine, but these two factors constitute a small item in anything except production welding.

Polarity. Polarity is of importance only to welding with direct current. The polarity of the welding current is the nominal direction of the flow of current in the circuit and establishes whether the electrode is a positive or negative terminal. The polarity to be used with any particular electrode is established by the manufacturer to control heat balance, *i.e.*, to control whether more heat is produced on the electrode or the work side of the circuit. Manufacturers' recommendations should be followed in all cases. The use of the wrong polarity will result in a substandard performance of the electrode.

With alternating current, since polarity changes very rapidly, the electrode must be suitable for use with either polarity. This does not mean, however, that a negative or positive electrode will not be satisfactory with alternating current. A high-quality electrode may prove satisfactory on both alternating and direct current.

The Arc. Proper control of current and voltage and arc length will result in the proper "heat" for depositing the weld.

The manufacturers of electrodes specify the amperage range of all electrodes for any given size. These recommendations should be followed. In general terms, the amount of current used determines the heat of the arc and the depth to which the base metal will be melted. The thicker the plate being welded, the more heat will be required to secure sufficient depth for the weld, and hence the larger electrodes are necessary.

The arc voltage, which can be controlled in the machine settings, controls the "force" of the arc and should be varied according to the needs of the job. On thin metal, a high voltage is desired, but a low current so that the arc will not burn through the metal. On an overhead job, the essential requirements are high current for penetration yet sufficient arc force to help push the metal up. In welding a casting, the need is for a high current, but the voltage may be low. The best welding machines will, therefore, provide for independent control of voltage and current.

The length of the arc also influences the quality of the weld. An arc that is too long will lose some of its force. The tiny globules of electrode metal tend to form into balls on the end of the electrode and then spatter. The arc stream is not concentrated. The correct length will vary according to the size and type of electrode, the material being welded, and the amount of heat. In general, to control the amount of effective arc force, the electrode should be kept as close to the work as possible, even dragging the electrode along ahead of the molten pool, just keeping the coating lightly touching the plate.

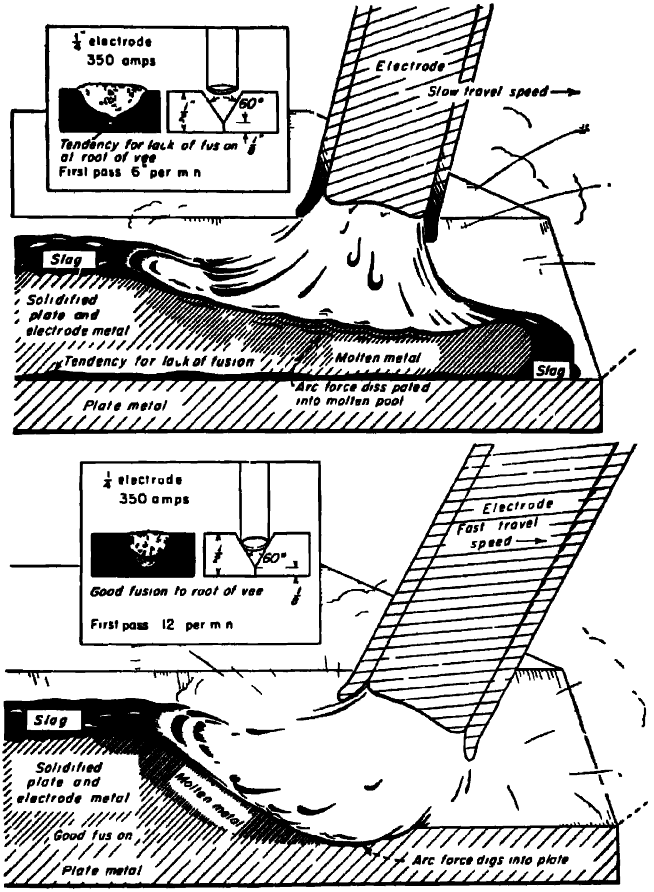

¼" electrode
350 amps
60°
¼"
⅛"
Tendency for lack of fusion at root of vee
First pass 6" per min
Electrode
Slow travel speed →
Slag
Solidified plate and electrode metal
Tendency for lack of fusion
Molten metal
Slag
Arc force dissipated into molten pool
Plate metal
¼" electrode
350 amps
60°
⅛"
Good fusion to root of vee
First pass 12" per min
Electrode
Fast travel speed →
Slag
Solidified plate and electrode metal
Molten metal
Good fusion
Arc force digs into plate
Plate metal

FIG. 12-5 The effect of slow travel speed (*top*) and fast travel speed (*bottom*) on penetration in a 60-deg V-groove weld

Arc Speed. The arc should be moved along the joint to be made with the greatest speed at which the surface appearance of the weld still remains satisfactory. Keeping the tip of electrode moving ahead of the molten pool at high speeds gives the force of the arc full opportunity to dig into the root of the joint. If the travel speed is too slow, the force is dissipated in the molten pool as it runs under the arc (see Fig. 12-5).

Proper speed thus results in deeper penetration into the root of the joint which in turn permits strong welds to be made with the deposition of less metal. A weld can be made either by piling up metal on the outside of the joint or penetrating deeply into the parent metal. Both welds will be equally strong, but the one made by penetrating deeply will be of much lower cost.

This essential arc speed can be obtained by giving proper attention to several factors.

1. The use of the proper type of electrode is necessary. Table 12-1 gives recommended applications for the main types of electrode for welding plain-carbon steel as classified by the AWS:

Table 12-1. Types of Electrodes

AWS classification	For deep penetration	For poor fit-up	For finish pass on deep-groove welding	For ease of operation in all positions	For steels with tendency for		
					Weld cracks	Internal porosity	Surface holes
E6010	x					x	
E6011	x					x	
E6012		x			x		x
E6013		x			x		x
E6013		x		x			
E6030	x					x	
E6020	x		x			x	

The largest electrode possible consistent with normal ease of operation should be used. This permits the use of higher currents.

2. Proper attention should be given to securing good fit-up in the parts being welded. Poor fit-up requires extra care and time to secure a satisfactory weld. The slower speeds also mean that more heat is being concentrated on the work which tends to increase problems of distortion.

3. The operator should always try to weld in the down-hand position with the joint level wherever possible. Speeds can be increased by as much as 400 per cent by changing from vertical to down-hand welding. Furthermore, considerably less skill is required to put *down* a good weld than is required to put *up* a good weld.

4. Excessive corrosion, paint, or other foreign matter should be removed from the edges to be joined.

5. Excessive build-up of weld metal should be avoided as wasteful. Only the specified amount of metal needed for strength should be deposited.

While all the above recommendations are for welding on mild steel, they should become the basis for the further development of all welding procedures. They are the fundamentals of good welding.

Illustrated in Fig. 12-6 are welds as they appear when made under various controlled conditions.

Location of Welds. The location of the welds in relation to the parts joined, in many cases, has an effect on the strength of the welded joint. As an example, repeated tests reveal that, when other factors are equal, welds having their linear dimension transverse to the lines of stress are stronger per average unit length than welds with linear dimension parallel to lines of stress. This is due to the stress distribution along the bead (Fig. 12-7*A*).

If the load on the weld is to be properly distributed, the welds should be located so as to take account of the shape of the sections joined. The ratio of the lengths of the welds at heel and toe of the angle (Fig. 12-7*B*) is such that there will be no tendency for the angle to turn and thus cause eccentric loads on the joint.

Resistance to a turning effect of one member at a joint is best obtained by welds well separated rather than by a single weld or welds close together. (Fig. 12-7*C*. A single weld at *A* is not so effective as welds at both *A* and *B*.)

If possible, welded joints should be designed so that bending or prying action is minimized. Symmetrical joints are most desirable as they are very much stronger than nonsymmetrical joints, the stress in symmetrical joints being more evenly distributed.

In some designs it may be desirable to take into account the distribution of stress through the welds in a joint. It is known that any abrupt change in surface (*e.g.*, a notch or saw cut in a square bar under tension) increases the local stress or causes stress concentration. In many cases such concentration of stress might be small and of minor consequence. However, in heavy or repeated loadings, this matter should have the attention of the designer (Figs. 12-7*D*, *E*, and *F*).

Stress in a weld having its linear dimension approximately parallel to the line of force is not evenly distributed. Under many load conditions, not at all unusual, the stress is greater at the ends of the weld than in the middle. It is therefore advisable in certain conditions to hook the bead around the joint. When this is done, far greater resistance to a tearing action on the weld is obtained (Fig. 12-7*G*).

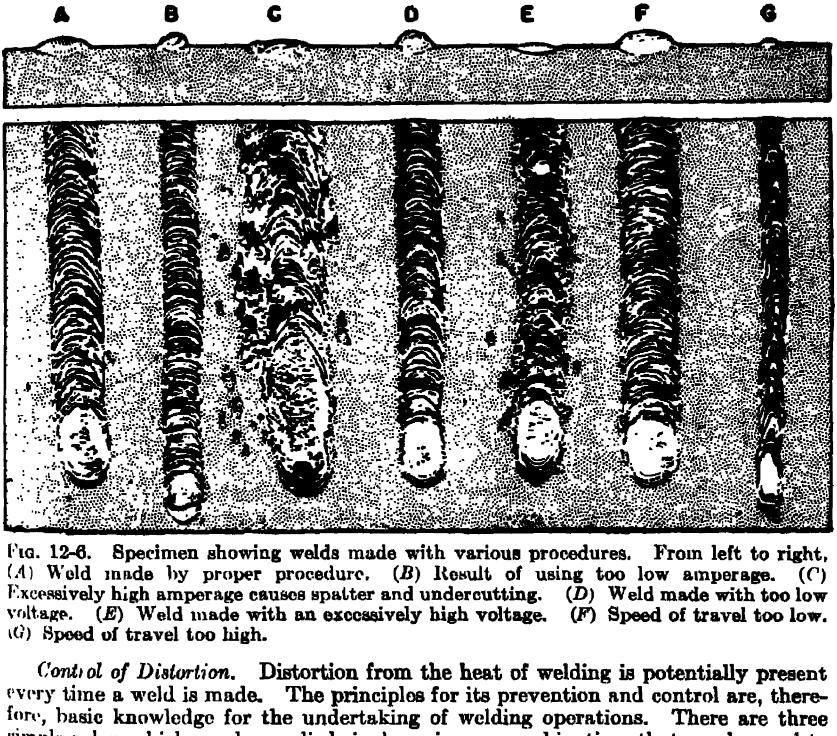


Fig. 12-6. Specimen showing welds made with various procedures. From left to right, (*A*) Weld made by proper procedure. (*B*) Result of using too low amperage. (*C*) Excessively high amperage causes spatter and undercutting. (*D*) Weld made with too low voltage. (*E*) Weld made with an excessively high voltage. (*F*) Speed of travel too low. (*G*) Speed of travel too high.

Control of Distortion. Distortion from the heat of welding is potentially present every time a weld is made. The principles for its prevention and control are, therefore, basic knowledge for the undertaking of welding operations. There are three simple rules, which can be applied singly or in any combination, that can be used to prevent and control distortion:

1. Reduce the effective shrinkage force. This can be accomplished in a number of ways:

a. Avoid overwelding. Addition of excess weld metal—not needed to meet the service requirements of the joint—is known as overwelding. This causes distortion and contributes nothing to the strength and performance of the joint. Weld metal should be kept at a minimum.

Another way of stating this principle is as follows: Use as little weld metal as possible, and make an intelligent use of the weld metal that is needed. For a conventional fillet weld for a T joint, it is known that the strength is determined by the effective throat (see Fig. 12-8). There is an excess of weld metal above the line *AA*, which does not increase the strength but obviously increases the effective shrinkage force. Less shrinkage force may be obtained with no loss of strength by using the deep-fillet technique. In contrast to the conventional weld, the deep penetration at

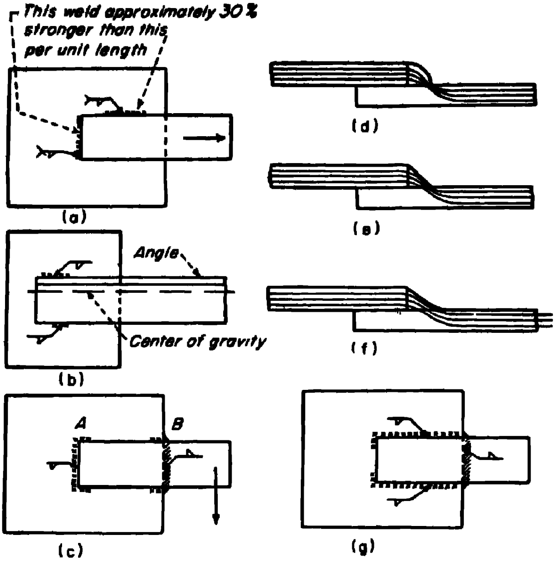

This weld approximately 30% stronger than this per unit length
(a)
(d)
(e)
Angle
Center of gravity
(b)
(f)
A
B
(c)
(g)

Fig. 12-7. Effect of location and type of weld on strength of joint: (*A*) Transverse welds are stronger than welds parallel to lines of stress. (*B*) Example of correct lengths of welds for equal load distribution (*C*) Example of proper placement of welds to resist turning effect of one member at joint. (*D*) Example of lap weld having poor distribution of stress through weld. (*E*) Example of lap weld having a more even distribution of stress through weld. (*F*) Example of lap weld in which there is fairly uniform transfer of stress through weld. (*G*) Example of welds hooked around the corners to obtain resistance to tearing action under eccentric loads.

the root of the fillet, using the deep-fillet method, gives 15 per cent more strength with the use of 30 per cent less deposited metal. Less weld metal means less distortion (see Fig 12-9).

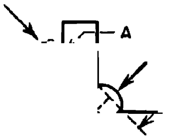


Fig. 12-8. Conventional fillet weld for T joint.

b. Use proper edge preparation and fit-up. It also is possible to reduce the effective shrinkage force through proper edge preparation. To obtain the proper fusion at the root of the weld with a minimum of weld metal, the bevel should not exceed 30 deg. Proper fit-up also is important, so the plates to be welded should be spaced $\frac{1}{32}$ to $\frac{1}{16}$ in. apart. A minimum amount of weld metal then will be needed to produce a strong joint.

c. Use few passes, which is another way to make an intelligent use of weld metal. Distortion in the lateral direction always is a major problem. Use of one or two passes with large electrodes reduces distortion in this direction. In general, lateral distortion is approximately 1 deg per pass.

In some cases, however, distortion in the longitudinal direction is a problem, and then, owing to the greater ability of a small bead to stretch longitudinally (compared to a large bead), the number of passes should be increased rather than decreased. This apparently paradoxical relationship is a function of the thickness of the plate and its natural resistance to distortion. There is inherent rigidity against the longitudinal bending of a plate provided the plate is thick enough. Light-gauge sheets have little rigidity in this direction and therefore will buckle easily. Unless the two plates to be

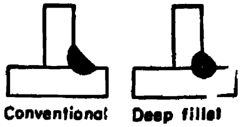


Fig. 12-9. Reduction of weld metal used through deep-fillet technique.

welded are restrained, there is no lateral rigidity whatsoever, since each of the two plates is free to move angularly with relation to one another, so lateral distortion is more common.

d. Place welds near the neutral axis. Another means of reducing the effective shrinkage force is to place the weld as close as possible to the neutral axis. In a conventional fillet weld, for example, the weld is so far from the neutral axis that it has sufficient leverage to pull the plates out of alignment. Use of the deep-fillet method places the weld close to the neutral axis which reduces the leverage (Fig. 12-10).

e. Use intermittent welds. To further reduce effective shrinkage force by minimizing the amount of weld metal, intermittent welds may be used in many cases instead of continuous welds. Often, it is possible to use up to two-thirds less weld metal and still obtain the strength required. EXAMPLE: Stiffeners for bulkheads and plates of all kinds. The use of intermittent welds also distributes the heat more widely throughout the structure.

f. Use "backstep" welding method. If the job requires a continuous weld, it still is possible to reduce the effective shrinkage force by the backstep technique. With this technique, the general direction of welding progression is, say, from left to right, but each bead is deposited from right to left (see Fig. 12-11).

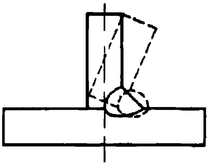


FIG. 12-10. Reduction of leverage by placing weld close to neutral axis with deep-fillet method.

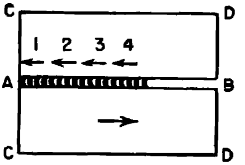


FIG. 12-11. "Backstep" technique of welding.

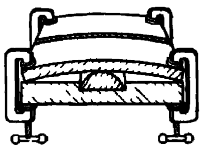


FIG. 12-12. Prebending before welding to exert counterforce against shrinkage force.

As each bead is applied, the heat from the weld along the edges causes expansion there which temporarily separates the plates at end *B*, but as the heat moves out across the plate to *C*, the expansion along the outer edges *CD* brings the plate back together. This occurs when the first bead is laid. The same will be true with each successive bead as it is laid, the plates expanding to a less and less degree with each bead because of the locking effect of each weld.

Where a continuous bead is laid in one direction, in many cases there is a tendency for the plates to spread and become locked in the spread position as the welding progresses. However, welding speed is the determining factor here. As a general rule, the greater the speed, the more the amount of spreading. In some cases a speed can be found at which the plates will not separate at all, and a slow speed may close the plates. This action is independent of current values for a given joint, being solely a function of the speed of travel of the arc.

2. Make shrinkage forces work to minimize distortion.

a. Locate parts out of position. A simple way to use the shrinkage force of weld metal to advantage is to obtain proper location of parts before welding. For example, a T weld may be made with the vertical plates out of alignment before the weld is deposited. When the weld shrinks, it will pull the vertical plate to its correct 90-deg position.

b. Space parts to allow for shrinkage. Another method is to space parts before welding. Experience indicates just how much space should be allowed for any given job, so that the parts will be in correct alignment after welding is completed.

c. Prebend. Shrinkage force can be put to work in many cases by prebending or prespringing the parts to be welded. For example, when the plates in Fig. 12-12 are sprung away from the weld side, the counterforce exerted by the clamps overcomes most of the shrinkage tendency of the weld metal, causing it to yield. But when the clamps are removed, there still is a slight tendency for the weld to contract, and this contraction of shrinkage force pulls the plates into exact alignment.

3. Balance shrinkage forces with other forces. Often the structural nature of parts to be welded in such as to provide sufficient rigid balancing forces to offset welding shrinkage forces. This is particularly true in heavy sections where there is inherent rigidity because of the arrangement of the parts. If, however, these natural balancing forces are not present, it is necessary to balance the shrinkage forces in the weld metal in order to prevent distortion.

a. Balance one shrinkage force with another. This can be accomplished by the use of proper welding sequence which places weld metal at different points about the structure so that, as one section of weld metal shrinks, it will counteract the shrinkage forces of previous welds already made. A simple example of this is welding alternately on both sides of the neutral axis in making a simple butt weld (see Fig. 12-13).

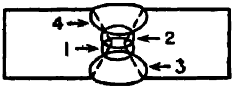


FIG. 12-13. Welding sequence to balance one shrinkage force with another.

Another application of this principle is the staggering of intermittent welds applied in a sequence. Here the shrinkage force of weld 1 is balanced by that of weld 2; the shrinkage force of weld 2 is balanced by that of weld 3, etc.

b. Peening. By peening the bead, it actually is stretched, counteracting its tendency to contract and shrink as it cools. Peening should be used with great care, for too much peening may damage the weld metal.

c. Use jigs and fixtures. The most important method of avoiding distortion, and one in which rule 3 is applied to the fullest extent, is the use of clamps, jigs, or fixtures to hold the work in a rigid position during welding. In this way, the shrinkage forces of the weld are balanced with sufficient counterforces to prevent distortion. What actually happens is that the balancing forces of the jig or fixture cause the weld metal itself to stretch, thus preventing the distortion.

WELDABILITY OF METALS

This discussion has been confined largely to the welding of mild steel. The fundamentals for welding mild steel are the basis for the development of procedures for the welding of special alloys and the other nonferrous metals. By applying certain cautions and special techniques, practically all metals have been successfully arc-welded.

Following is a brief description of the general principles involved in determining the weldability of metals and in establishing procedures. Space does not permit a complete detailed description of the many varied techniques. Electrode manufacturers provide complete procedure information for all their special electrodes.

Obviously some metals will weld more readily than others. The economy and quality of the welding on various metals may be affected by any one of the following factors:

1. Oxidation. (*a*) Oxidation producing a gaseous oxide of some one of the elements causing gas holes in the weld metal; (*b*) oxidation producing solid oxides which have a melting temperature higher than the metal, thus causing slag inclusions, (*c*) oxidation producing oxides which are soluble or which are heavier and sink in the molten metal and which render the weld metal brittle or of low strength.

2. Vaporization. Vaporization of some element in the metal which vaporizes at a temperature lower than the melting point of the metal.

3. Nonmetallic Inclusions. Some metals may contain finely divided nonmetallic inclusions which have a melting point higher than that of the metal and therefore did not coalesce when the metal was refined but do melt and coalesce under the high temperature of the arc and then form visible slag inclusions.

4. Change of Structure. Change of structure or arrangement of elements within the metal may take place during arc welding, causing change of physical properties or change of resistance to corrosion, etc.

5. Gas Solubility of Metal. (*a*) Different elements may affect the solubility of various gases at different temperatures, and a decrease in solubility of a gas with a decrease in temperature at the freezing point may cause porosity in weld metal. (*b*)

The fluxing out or elimination of an element during welding may cause the capacity of the metal for a given gas to decrease and thus cause the gas to be given up producing porosity in the weld metal. (*c*) Absorption of gases during welding which form stable compounds with elements in the metal and thus alter the composition and physical properties of the weld metal.

6. **High Coefficient of Thermal Expansion,** or high contraction of weld metal upon cooling.
7. **"Hot Shortness,"** or low strength of the metal at high temperatures.
8. **Thermal Conductivity,** or rate of transfer of heat from fusion zone.

The above outline indicates why some metals are more satisfactory than others.

A careful study of these factors indicates that most of the possible undesirable characteristics can be corrected by one or more of the following methods:

1. Selection of metal within the permissible class most suitable for arc welding
2. Use of a properly shielded arc
3. Use of proper fluxing material
4. Use of proper electrode or filler metal
5. Proper welding procedure
6. Subsequent heat-treatment which may be required in some cases

In considering the weldability of any metal it should be realized that the weld largely depends upon the characteristics of the weld metal which may come from two sources, *viz*, base metal and electrode of filler metal.

If little or no electrode or filler metal is used, the proper selection of the base metal becomes of prime importance. If the weld metal comes mostly from the electrode or filler metal, the selection of the proper electrode or filler metal becomes of prime importance. However, both electrode and base metal are subjected to similar requirements during arc welding, and both should be of best arc-welding quality, although in many cases the electrode or filler metal serves as a corrective for the base metal.

The carbon content of steel is the most important factor in determining heat-treatment procedures. When the carbon content is low, the weldment will require no special consideration. When it is high, preheating is generally necessary at a temperature usually from 450 to 650°F in order to control the rate of cooling.

Failure to preheat high-carbon steel before welding on it will usually result in the hardening of the steel adjacent to the weld. Since this will reduce the ductility of the steel, cracking in the metal next to the fusion zone is likely to result. This steel should be welded while it is hot and then allowed to cool slowly, increasing the cooling time proportionately as the carbon content of the steel is greater. For steel above 40 per cent carbon, overnight cooling in a furnace is sometimes necessary to avoid a hardened zone. Type E6015, E6016, E7015, and E7016 electrodes are recommended for welding high-carbon steels, although the normal mild-steel electrodes can be used.

Definite limitations and specifications for preheating should be established only after experience. Research is discovering that preheating is unnecessary in many applications where it is generally thought to be desirable. The service requirements of the weldment must be considered as well as the use of welding procedures to control the forces created by the heat (see Control of Distortion).

SURFACING BY ARC WELDING

The use of arc welding to deposit a surface of metal of a desired characteristic has many applications in plant operation: building up worn shafts, lining carbon-steel vessels with stainless steel, and the application of wear-resisting surfaces to machinery of all kinds.

The success of the application of surfacing metal by welding depends largely on the choice of the proper surfacing material which in turn is dependent on the following factors:

1. Material to be surfaced
    a. Its composition
    b. Its physical condition, such as hardness, previous heat-treatment, etc.

2. Dimensions
    a. Size and shape of parts
    b. Size and location of the area to be surfaced
    c. Thickness of weld deposit
3. Finish required
    a. As welded
    b. Machined
    c. Rough grind
    d. Fine grind
4. Service conditions
    a. Hardness required
    b. Corrosion, the corrosive medium
    c. Abrasion, its extent and nature
    d. Impact, its extent and nature

Materials to Be Surfaced. The hundreds of metal alloys can be grouped for consideration of their suitability to weld surfacing into two general groups:

In group A are metals or alloys whose physical characteristics are not greatly changed as a result of heating and cooling and which will withstand sudden localized temperature changes without cracking. This group includes plain-carbon steel with 0.30 carbon maximum, low-carbon alloy steels, austenitic steels such as the stainless chrome-nickel, and the high-manganese steels. Copper and most of its alloys would also be included.

In group B are those material or alloys whose physical characteristics are changed considerably (particularly as to hardness) as a result of the application of welding heat and subsequent cooling or which will crack with the sudden application of localized heat. This group includes medium- to high-carbon steels, tool steels, medium- to high-carbon low-alloy ferritic steels, cast irons, and, in general, all hard metals and alloys.

The metals in group A can be weld-surfaced without any particular precautions such as preheating since the thermal cycle incident to the welding will not cause harmful cracking or hardening.

Special precautions must be exercised in weld surfacing metals of group B to avoid cracking due to thermal shock. Either the hardness of the metal must be reduced through annealing or the thermal shock reduced by preheating. In some cases both may be necessary.

Further precaution is necessary in group B to minimize weld hardening. Metal adjacent to the weld will harden when cooled quickly following heating to temperatures high enough to cause solution of free carbides. This may result in subsequent cracking, failure of the part, or spalling of the weld deposit. To reduce the tendency to weld-harden, preheating, slow cooling, and postheating, or a combination of these, may be employed.

Dimensions. The size and shape of parts determine largely the heat capacity of parts to be surfaced and therefore determine to some extent the thermal cycle to which the part and the weld deposit are subjected. A large mass and large heat capacity will heat up slowly, reach a relatively low temperature, and draw the heat away from the weld area rapidly. A small mass will heat up rapidly, reach a high temperature, and draw the heat away from the weld area slowly. If local welding heat is applied to a large mass of metal, which is capable of being quench-hardened, the hardening will be drastic and cracking is likely to result due to thermal stresses. If the mass is small, the degree of hardening will be less because of its low heat capacity and cooling will be more uniform throughout, resulting in less severe thermal stresses. If the weld metal being deposited is of the type whose hardness is affected by the thermal cycle, the hardness of the deposit will be greater on a large mass owing to the quench effect.

Heat input will also be determined by the size and location of the area to be surfaced. The size of the area will, therefore, influence the thermal cycle as the size and shape of the part itself does.

Finish Required and Service Conditions. Electrodes for hard surfacing are not well adapted for vertical or overhead welding. Work should be placed in as nearly flat a position as possible. Thick deposits should be generally avoided. If thick deposits are desired, electrodes designed for such applications must be used. If machining is to be done on the deposit, it must be necessary to anneal. In some cases grinding is the only way to work the deposits.

In all cases, for the best results in hard surfacing, manufacturers' recommendations for the use of their electrodes should be studied and followed.

The various service conditions to which a weld-faced deposit may be subjected can be classified as follows:

1. *Abrasion.* (*a*) Abrasion is a grinding action due to rubbing against an abrasive material such as rock, sand, clay, or soil or due to (*b*) sliding, rolling, or rubbing action of one metal part against another. Either *a* or *b* may take place under high or low pressure.

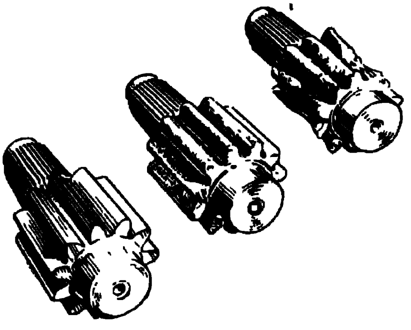


Fig. 12-14. Parts reclaimed through hard surfacing with arc welding. Badly worn pinion gears can be built up and ground to shape and made as good as new. Gear at left is ready for use.

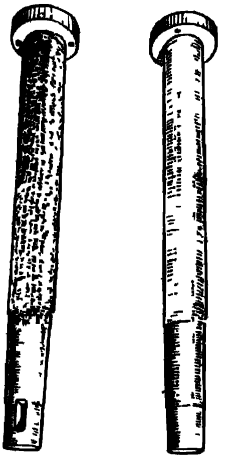


Fig. 12-15. Grinding-machine journal shaft rebuilt by electric welding. Shafts were first machined, $\frac{1}{32}$ in. being taken off the radius to remove worn spots, then built with application of abrasion-resistant weld metal. The entire job required 5 hr.

2. *Impact.* Impact takes place in various degrees from light to heavy. It tends to deform the surface or cause cracking or chipping.

3. *Corrosion.* Corrosion includes the action of various chemicals, ordinary water, and also oxidation or scaling at elevated temperatures.

It is obvious that, before the proper choice of weld-surfacing material can be made for a given application, the service conditions must be analyzed. In cases where more than one of the wearing factors are at work, their relative importance must be determined.

For determining the economy of weld surfacing, see Table 12-2. Surfacing metal is expensive, and by depositing only a limited quantity of it on a low-cost base metal, instead of making the part completely of the expensive metal, great savings can be made in both reclaiming worn equipment and building new machinery or tools (see Figs. 12-14 and 12-15).

Table 12-2 indicates that, for a surface material as high in cost as $5 per lb against an original cost of 20 cents per lb, 3 per cent may be surfaced by welding and the original cost not exceeded. More usually it will be found that 20 per cent or more may be surface metal without exceeding the original cost.

Table 12-2

Cost cents/lb, using single metal	Percentage of total weight which may be surface material without exceeding cost of using one metal throughout				
	Cost of surface materials				
	$1/lb	$2/lb	$3/lb	$4/lb	$5/lb
20	15.8	7.7	5.1	3.8	3.0
40	37.0	18.0	11.9	8.9	7.1
60	58.0	28.2	18.6	13.9	11.1
80	79.0	38.5	25.4	19.0	15.1
100	100.0	48.7	32.2	24.0	19.2
120		59.0	39.0	29.1	23.2
140		69.2	45.7	34.2	27.3

SHEET-METAL WELDING

The welding of sheet metal has frequent application in plant operation (Figs. 12-16 to 12-19). The same principles of good welding practice as previously outlined apply in welding sheet metal, but the nature of the work places special emphasis on several aspects. The problem of distortion requires special consideration in welding thin-gauge metals as well as the problems of burning through the metal.

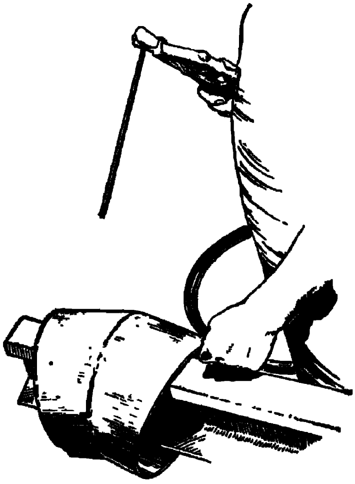


Fig. 12-16. Welded fabrication simplifies making of angles for piping in ventilating systems. Filing or grinding of the metal may be necessary to secure a fit-up that will result in airtight joints and the required angle. The angle illustrated is made of 11-gauge galvanized steel.

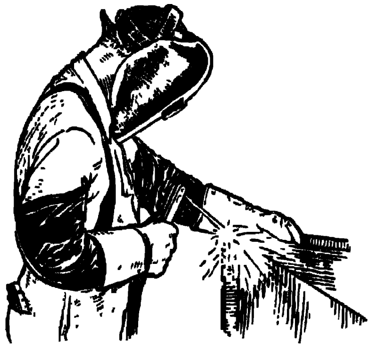


Fig. 12-17. Section of an air duct being welded out of 18-gauge galvanized iron Corner joints are tacked every 2 in., then welded with a ⅛-in. electrode at a 45-deg angle.

Special attention should therefore be given to all the factors involved in controlling distortion: the speed of welding, the choice of proper joints, good fit-up, position, selection of proper current, use of clamping devices and fixtures, number of passes, and sequence of beads.

Within the limits of good welding appearance, the highest arc speeds and the highest currents should be used. In sheet-metal work, however, there is always the

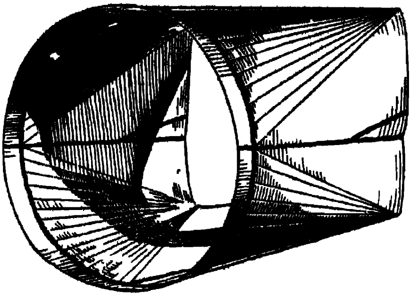


FIG. 12-18. Exhaust stack made of 14-gauge 18-8 stainless steel. All joints are butt-welded with a 1/8-in.-diameter electrode.

Table 12-3. Sizes of Electrodes for Sheet Metal

Type of welded joint	20 ga			18 ga			16 ga			14 ga			12 ga			10 ga			8 ga		
	F*	V*	O*	F	V	O	F	V	O	F	V	O	F	V	O	F	V	O	F	V	O
Plain butt	3/32	3/32	3/32	3/32	3/32	3/32	1/8	1/8	1/8	1/8	1/8	1/8	5/32	5/32	5/32	5/32	5/32	5/32	3/16	5/32	5/32
Lap	3/32	3/32	3/32	3/32	3/32	3/32	1/8	1/8	1/8	5/32	5/32	5/32	5/32	5/32	5/32	3/16	3/16	5/32	3/16	3/16	5/32
Fillet				3/32	3/32	3/32	1/8	1/8	1/8	1/8	1/8	1/8	5/32	5/32	5/32	3/16	5/32	5/32	3/16	5/32	5/32
Corner	3/32	3/32	3/32	3/32	3/32	3/32	1/8	1/8	1/8	1/8	1/8	1/8	3/16	5/32	5/32	3/16	5/32	5/32	3/16	5/32	5/32
Edge	3/32	3/32	3/32	3/32	3/32	3/32	1/8	1/8	1/8	1/8	1/8	1/8	3/16	5/32	5/32	3/16	5/32	5/32	3/16	5/32	5/32

* F—flat position; V—vertical; O—overhead.

Table 12-4. Welding Currents for Sheet Metal

Type of welded joint	20 ga			18 ga			16 ga			14 ga			12 ga			10 ga			8 ga		
	F	V	O	F	V	O	F	V	O	F	V	O	F	V	O	F	V	O	F	V	O
Plain butt	30*	30*	30*	40*	40*	40*	70*	70*	70*	85*	80	85*	115	110	110	135	120	115	190	130	120
Lap	40*	40*	40*	60*	60*	60*	100	100	100	130	130	130	135	120	120	155	130	120	165	140	120
Fillet				40*	40*	40*	70*	70*	70*	100	90	85	150	140	120	160	130	130	160	160	130
Corner	40*	40*	40*	60*	60*	60*	90*	90*	90*	90	80	75	125	110	110	140	130	125	175	130	125
Edge	40*	40*	40*	60*	60*	60*	80*	80*	80*	110	80	80	145	110	110	150	120	120	160	120	120

* Electrode negative, work positive.

Table 12-5. Minimum Arc Volts for Welding Sheet Metal

Type of welded joint	20 ga			18 ga			16 ga			14 ga			12 ga			10 ga			8 ga		
	F	V	O	F	V	O	F	V	O	F	V	O	F	V	O	F	V	O	F	V	O
Plain butt	18	18	18	18	18	18	29	29	29	29	29	29	25	26	25	25	27	25	27	27	26
Lap	21	21	21	22	22	22	25	25	25	25	25	25	25	26	25	28	26	25	28	26	25
Fillet				21	21	21	27	27	27	25	24	24	25	24	24	25	24	24	25	24	24
Corner	21	21	21	24	24	24	24	24	24	24	28	25	24	25	26	24	28	26	27	28	26
Edge	21	21	21	23	23	23	25	25	25	27	26	26	28	26	26	28	27	26	28	27	27

limitation imposed by the threat of burn-through. As the gap in the work increases in size, the current must be decreased to prevent burn-through which of course will reduce welding speeds. A clamping fixture will improve the fit-up of joints and thus make possible the higher speeds. If equipped with a copper backing strip, the clamping fixture will make for easier welding by decreasing the tendency to burn through, and will also remove some of the heat which causes warpage. Sheet-metal joints, where possible, should be welded downhill at about a 45-deg angle with the same or slightly higher currents as are used in the flat position.

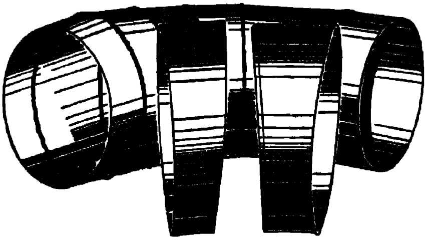


FIG. 12-19. Technique of welded elbows. Sections are butt-welded into rings, and edges are drawn with hammer and dolly.

Tables 12-3, 12-4, and 12-5 offer a guide to the selection of the proper size electrode, current, and voltage for the various types of joints as used with sheet metal ranging from 20 to 8 gauge.

STRUCTURAL WELDING

Extreme simplicity is a fundamental advantage of arc-welded design in structures. Arc welding joins two members directly to each other without the use of a third or connecting member (Fig. 12-20). While this simplicity is responsible for savings in designing, detailing, and erecting, nevertheless stresses at all connections must be calculated and the required weld specified.

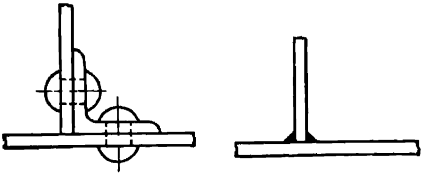


FIG. 12-20. Simplicity of welded design.

Table 12-6. Length of Fillet Weld to Replace Rivets
(Larger value in each case for emergency allowable)

Rivet diam, in.	Rivet shear value @ 15,000 psi*	Length of fillet welds (to nearest ⅛ in.), "fusion code" (structural),* shielded arc welding				
		¼-in. fillet, in.	5/16-in. fillet, in.	⅜-in. fillet, in.	½-in. fillet, in	⅝-in. fillet, in.
½	2,950	1½	1¼	1⅛	⅞	¾
⅝	4,600	2¼	1¾	1½	1¼	1
¾	6 630	3	2½	2⅛	1⅝	1⅜
⅞	9,020	4⅛	3⅜	2⅞	2⅛	1¾
1	11,780	5¼	4¼	3⅝	2¾	2¼

* AISC value for rivet in shear is 15,000 psi, for weld in shear it is 13,000 psi (American Institute of Steel Construction).

NOTE: ¼ in. is added to calculated length of fillet.

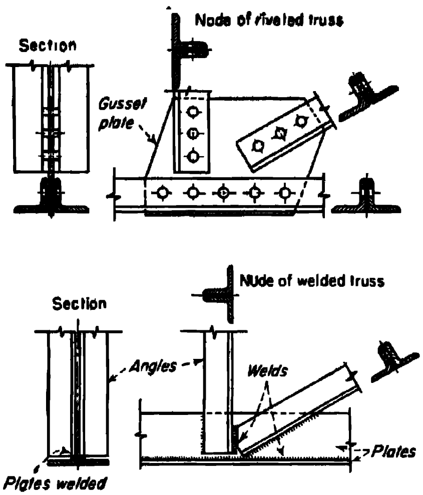


FIG. 12-21. Node of riveted and welded trusses showing how welded design cuts dead weight and reduces secondary stresses for greater factor of safety.

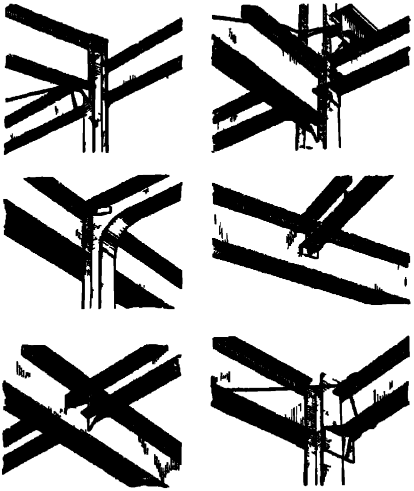


FIG. 12-22. Examples of facilities provided in steel erection by use of welding are the various clip angle seats, split I-beam segments, tie-rod clips, etc., which are shown. (*Top left*) Junction of roof beam and eaves strut at front, (*top right*) typical connection of girder and beam to column, (*center left*) junction of roof beam and eaves strut at rear; (*center right*) junction of roof beam and girder; (*bottom left*) junction of lower roof beam at girder; (*bottom right*) intersection of lower roof beam and girder.

The chief economy is less material. Trusses properly designed for arc welding require practically no gusset plates (Fig. 12-21). An entire structure fabricated by arc welding acts as a single member, and therefore lighter members usually can be used. Savings in material on structures have run generally between 15 and 25 per cent.

Welding also reduces the handling of material in fabrication. Welded connections require only two operations and only a welding machine for equipment. Much of the handling of the main members is therefore completely eliminated.

Welding permits the designer to place material in the structure where it is needed to meet the load demands without being restricted by considerations for the fabricating process. This simplification makes practical the construction of rigid frame structures with their freeing of internal space from supporting columns.

A detailed discussion of the design considerations in welded columns, beams, girders, trusses, bracing, bar joists, and rigid frame construction is not possible here but should be investigated before undertaking welded structural work of any kind. Figures 12-22, 12-23, and 12-24 indicate some of the possibilities (see Table 12-6 for the necessary length of fillet weld to replace rivets).

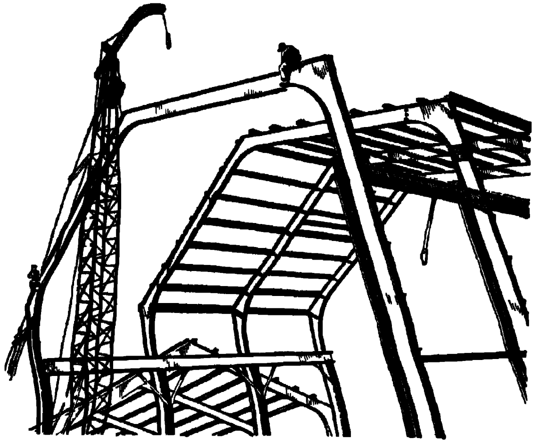


FIG. 12-23. Erecting roof members for a rigid frame building. (The Austin Company.)

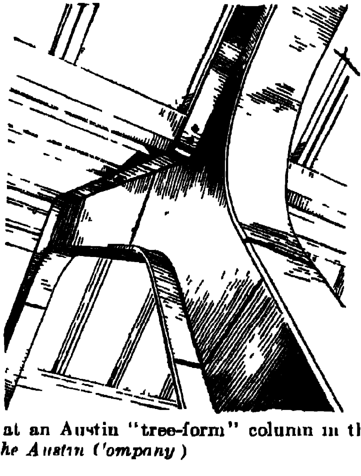


FIG. 12-24. Looking up at an Austin "tree-form" column in the saw-tooth section of a rigid frame building (*The Austin Company*)

PIPING

Piping systems welded in shop and field have been installed in buildings, plants, offices, powerhouses, and ships. The advantages of arc-welded piping include permanently tight connections of greater strength and rigidity, less resistance to flow owing to elimination of projections inside the pipe, more pleasing appearance, easier and cheaper application of insulation, simplification of design, and the elimination of many fittings.

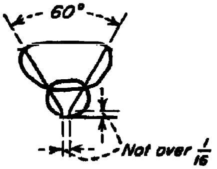
Table 12-7

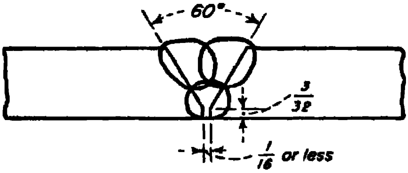
Size of pipe		Beads or passes	Electrode size in	Current, amp	Arc time min/joint	Pounds of electrode/joint
Diam, in	Thickness in					
2*	0.154	1 2	5/32	120 140	5.5	0.35
3*	0.216	1 2	5/32	120 140	7.0	0.44
4†	0.237	1 2 3	5/32 3/16 3/16	120 160	9.0	0.64
5†	0.258	1 2 3	5/32 3/16 3/16	120 160	11.3	0.81
6†	0.280	1 2 3	5/32 3/16 3/16	120 160	13.5	0.96
8†	0.322	1 2 3	3/16	150 170	17.8	1.33
10‡	0.365	1 2 3 4	3/16	150 170	22.0	1.65
12‡	0.375	1 2 3 4	3/16	150 170	26.5	2.0
14‡	0.375	1 2 3 4	3/16	150 170	31.3	2.3

* See Fig. a

† See Fig. b

‡ See Fig. c.





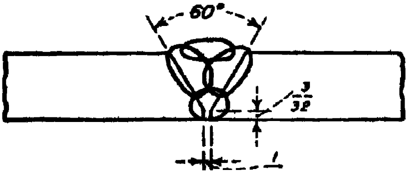


Table 12-7 and Figs. *A*, *B*, and *C* contain procedure data, speeds, and amount of electrode required for welding butt joints in standard-weight pipe in horizontal position.

Pipe above 12 in. in diameter may be obtained in a great many different wall thicknesses. Inasmuch as pipe is being used in these larger diameters for h-p and high-temperature service, the data in Table 12-8 are suggested for use in determining the number of passes to be used for the various pipe thicknesses:

Table 12-8

Pipe thickness	*Number of passe*
¼	3-4
⅜	4-5
½	6-8
¾	11-14
	13 16

The electrode size will be $\frac{5}{32}$ and $\frac{3}{16}$ in., depending upon the particular design, the load conditions, and the specifications. For example, on 14-in. pipe, $\frac{3}{16}$-in. electrode is normally used for passes; whereas in case of very h-p work, a $\frac{5}{32}$-in. electrode is used for the first pass, and $\frac{3}{16}$ in. for the following passes.

Piping for h-p and high-temperature systems has been increased in efficiency by the use of welded design. Welded joints permit high operating stresses and thinner sections. However, these systems must be designed according to code, giving due consideration to stress distribution, and for all unusual conditions of loading.

In the field work of installing a welded piping system, a method must be worked out for the inexpensive and quick laying out of the various types of joints. There are several methods which may be used. One that is simple and can be used in almost all cases is the use of fixture with movable fingers which will give the desired contour merely by holding it against the pipe in the necessary position. Another method is to use templates, giving the operator the exact curve, which can be fitted around the pipe's circumference to indicate the precise cutting curve The third method is the use of special curves on a master drawing. The only equipment needed is the master drawing, a string, a piece of chalk, and a scale.

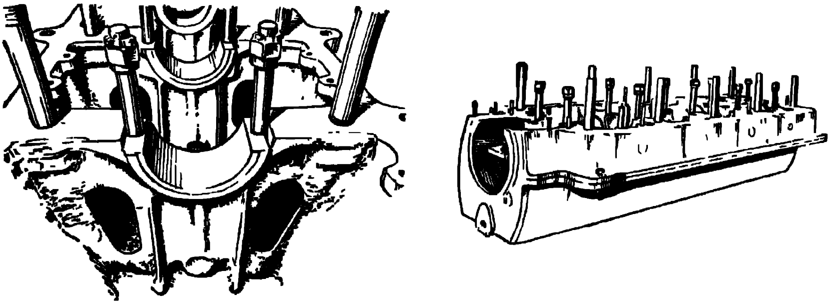


FIG. 12-25. Welded repair of a diesel-engine base. Studding was used together with an electrode of the mild-steel type.

TANKS AND PRESSURE VESSELS

Arc welding can be used to fulfill all the requirements imposed by the operating conditions of all types of tanks. Pressure, temperature, and chemical reaction are withstood without leakage under all load conditions.

A properly designed and welded joint is 20 to 30 per cent stronger than the plate itself. Plate thickness, therefore, need not be so great, and all punching and caulking operations are unnecessary. Maintenance costs are practically negligible as the joints are permanently tight and have greater resistance to corrosion.

Pressure vessels should be built in accordance with the ASME Boiler Code which requires definite procedure of welding and stress relieving. Butt joints are used entirely in this class of work. Plate thicknesses generally range from 1 to 3 in., and practically all the welding is done by the shielded arc process.

MISCELLANEOUS REPAIRS

In addition to the various types of work that have been classified in the foregoing sections, the repair or replacement of broken parts of all types of machinery is greatly

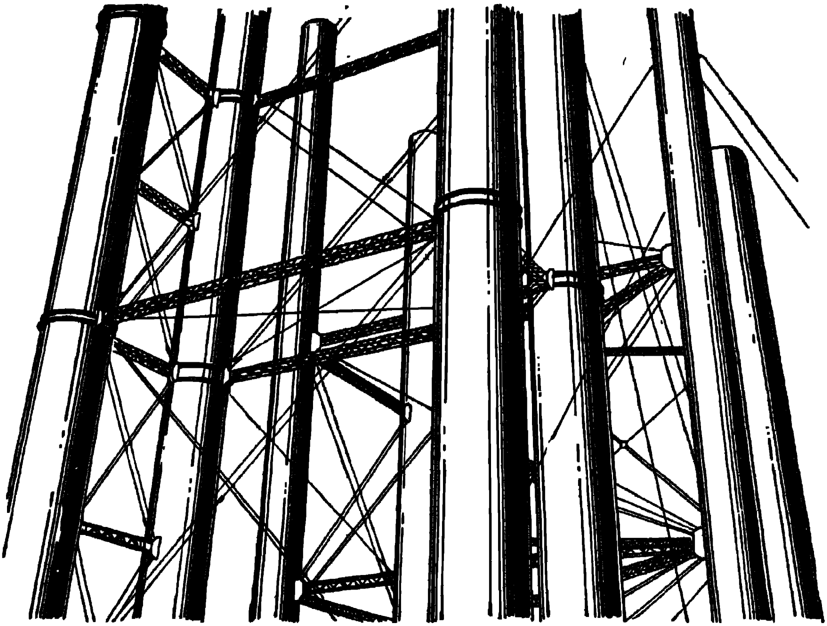


FIG. 12-26. Restoring rusted smokestacks by welding on ¼-in. plate rolled to circumscribe the stack. Plates were beveled to 60 deg and welded in two passes.

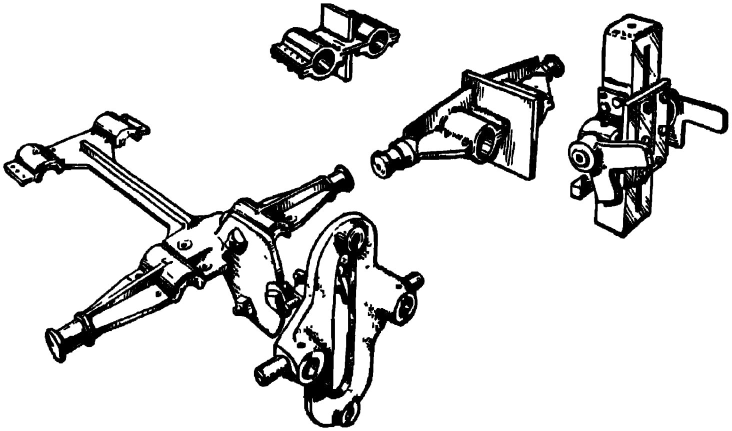


FIG. 12-27. Welded steel parts (right) to replace broken castings (left) from a 20-year-old dragsaw.

facilitated through the use of arc welding. The procedures used will vary with the nature of the job, and no attempt can be made to outline the many possibilities. When the repair is made without dismantling the machine, naturally compromises in procedure have to be made to meet the conditions. Sound welding practice should be the guide in all cases.

Figures 12-25, 12-26, and 12-27 are typical examples of repairs or replacements that were effected through the use of arc welding.

SILVER-ALLOY BRAZING

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INTRODUCTION

Brazing is a process of joining metals by means of heat, wherein a nonferrous filler metal having a melting point lower than those of the metals or alloys being joined is used. To differentiate brazing from soft soldering, an arbitrary minimum temperature of 800°F for the melting point of a brazing alloy has been quite generally accepted. The present trend is to classify brazing as a welding process, although the common concept of a weld has been that of a joint made by actual fusion of the base metals being joined. Distinguishing features of a braze are that a filler metal is always used and that this filler metal always melts below the melting point of the base metals. The metals being joined are never heated to the point of fusion. Another feature of a braze is that the surfaces of the joint are fitted closely enough to permit the flow of the brazing alloy through the joint by capillarity.

Brazing is an extremely old process of joining metals. It is probable that it was originally called "brassing" because of the fact that the first brazing alloys were brass. These brass alloys contained copper and zinc in about equal parts and melted at approximately 1600 to 1650°F. Because of their high zinc content, they were also called spelter solders. Many years ago it was discovered that the addition of silver to the brass alloys lowered the melting points of the brass and produced alloys having increased ductility which could be fabricated into sheet and wire—forms which were more convenient to use than the usual granulated form of the spelter solders. These new alloys were called "silver solders" or "hard solders." In recent years the silver alloys have been quite generally called "silver brazing alloys" in order to avoid any confusion with the tin-lead alloys and similar soft solders, as the properties of joints made with the silver brazing alloys bear closer resemblance to joints made with the brass alloys than to joints made with soft solders.

The silver brazing alloys are composed chiefly of silver, copper, and zinc, but other metals, such as cadmium, tin, manganese, and nickel may be added to impart specific properties. The melting points of the alloys will vary from approximately 1150 to 1600°F depending upon the composition. Other silver alloys are available with melting points both above and below this temperature, but they have not been used to any great extent commercially. It is these relatively low melting points of the silver alloys as compared to brass alloys, combined with high joint strengths obtainable with a corresponding minimum loss in strength or other desirable properties of the base metals being joined, which has been an important factor in their wide acceptance for industrial applications.

Because of their malleability, silver brazing alloys can be obtained in various shapes and sizes. Sheet can be obtained in all gauges down to 0.003 in. and can be used for inserts in joints when making lap or scarf joints. Washers can be made from sheet and used as inserts for joining flanges or spuds to tanks. Strips of various widths are used for hand feeding or can be formed into rings for use as inserts.

Wire is also available and is convenient and economical for use in hand feeding or forming into rings for preplacing.

Silver brazing alloys in the form of filings or powder in sizes from 20 to 150 mesh can also be obtained. This form is useful when it is more convenient to sprinkle the alloy on the joint before heating. Because of the large surface area of the alloy in this form, particular care must be taken to protect the alloy with flux to prevent oxidation.

USES

The current shortages of metals have given a strong impetus to the trend in industry to lighten, without loss of strength, equipment of all sorts. This has led to the building up of equipment from simple basic components which are readily fabricated by mass-production methods, thus eliminating the use of heavy, complicated castings or costly machining from a single piece with the production of excessive scrap. A natural consequence of this is the displacement of all types of mechanical joints wherever possible by the use of various welding processes.

As silver brazing alloys can be used for joining practically all ferrous and nonferrous metals and alloys except such low-melting materials as aluminum and zinc-base alloys, the use of silver brazing has expanded rapidly. The fact that silver brazing alloys are free flowing, corrosion resistant, and will produce strong joints makes them particularly suitable materials for accomplishing this objective.

A complete list of the uses of silver brazing alloys would cover practically every industry. They are used extensively in making airplane equipment for such things as cable tubes, gas and oil lines, instrument connections, radio shields, propeller connections, carburetor parts, and landing-gear parts. In the automotive industry they are used on flexible couplings, horns, gas and oil lines, spark plugs, steering wheels, vacuum gear-shift lines, and many other items. The dairy industry, where corrosion resistance and sanitary joints are important factors, uses silver brazing alloys. A recent development is the joining of pipe lines in radiant-heating systems where permanent leakproof joints are essential.

During the Second World War many vital parts of our fighting ships were joined with silver brazing alloys. Such things as deck fittings, electrical lines and switch boxes, oil lines, and pipe lines were brazed and successfully stood the test of battle conditions. In addition, brazed joints were used in most of the instruments aboard the ships.

Manufacturers of chemical equipment, tools, beverage dispensers, cooking utensils, sporting equipment, firearms, lawn mowers, bicycles, hardware, railroad equipment, and silverware all have applications where they use silver brazing. This list of users could be greatly expanded, but this will serve to show the wide field of commercial use of silver brazing.

The applications cited above deal with the fabrication of various types of equipment, but an important use of brazing which should not be overlooked is the repair and salvaging of broken parts. Examples are the repair of broken high-speed tools, such as broaches and milling cutters, without destroying the temper, retipping carbide tools, and the repair of broken shafts. Expensive equipment can frequently be reclaimed at a fraction of the replacement cost.

FUNDAMENTALS OF BRAZING

While silver brazing operations are relatively simple and unskilled workmen may obtain a high proficiency in a short training period, there are certain fundamentals which must be followed if the full advantages of the process are to be obtained.

The essential requirements for obtaining good joints may be divided into six general classifications, as follows:

1. Design
2. Preparation of joint surfaces
3. Fluxing
4. Assembling and jigging
5. Heating
6. Final cleaning

These requirements will be discussed individually.

Design. The three principal types of joints used in silver-alloy brazing are butt, scarf, and lap or shear, and each of them may be used with flat, round, or tubular members. Examples of these types, showing both good and poor design, are illustrated by Fig. 12-28.

Although not commonly used, except in those cases where lap- or shear-type joints cannot be accommodated by the shape of the parts to be joined, butt joints can be made which have remarkably high strengths. With the softer metals, such as copper or low-carbon steels, the strength of butt joints will often exceed the strength of the base metals; and even with many of the harder alloys the strength will customarily be higher than the elastic limit of the base metal. An excellent example of their strength is the use of butt brazes for the repair of broken broaches. The life of the repaired broach often is as long as that of a new tool. Another example would be hollow handles, where the two halves are stamped out of sheet and the edges of the stampings ground to give a close fit.

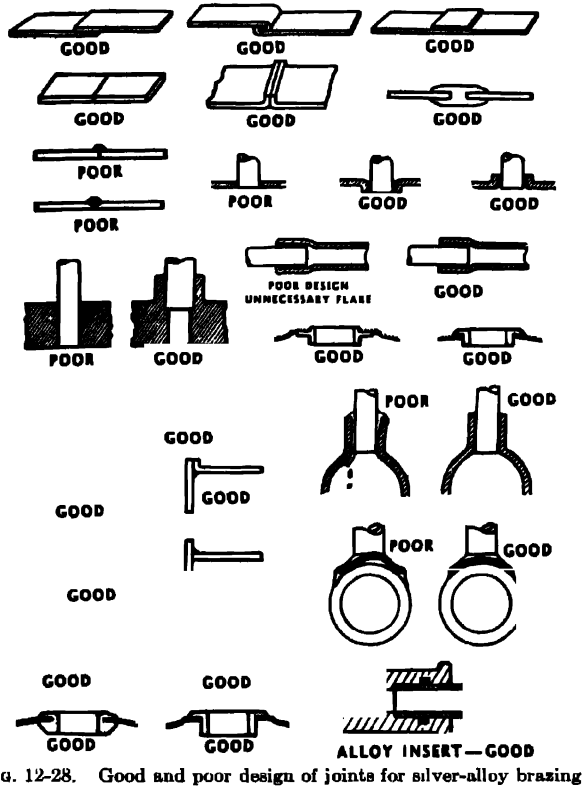

Fig. 12-28. Good and poor design of joints for silver-alloy brazing.

A scarf joint is a modified form of butt joint in which a larger bonded area is obtainable for the same thickness of metal. The strength of a scarf joint, however, does not increase proportionately to the increase in area when tested in tension. This is accounted for by the fact that such a joint fails by shear, and as the shear strength of a joint is lower than the tensile strength, the increase in area is to a great extent offset by the lower strength per square inch. A scarf joint does have an advantage in certain cases where it results in a different distribution of stresses than would be obtained in a butt joint. Typical uses for scarf joints are in joining band saws or heavy rod which is subsequently drawn to a smaller size.

Lap or shear joints are the most commonly used. They have the advantage that the bonding area can be varied to provide any desired factor of safety against breakage or leakage. The use of lap or shear joints frequently simplifies the problems of assembling and jigging, which is a distinct advantage where automatic brazing methods are used. Also, they lend themselves readily to the use of preplaced brazing alloy.

Where pressure can be applied to a lap joint, as in brazing sheet, tests have shown that, with such metals as copper, brass, or Everdur, strengths greater than that of the metal joined can be obtained with laps equal to the thickness of the sheet. With higher strength materials, such as monel or steel, this amount of lap is not sufficient. Also, with tubular members where pressure cannot be applied, as a rule, the strength of joint will not be so high as on sheet laps. In this case, a lap of about three times the thickness of the thinner member is recommended.

In order to obtain the maximum strength of joint, clearances of 0.001 to 0.003 in. are usually required. With less than 0.001 in. clearance, slight irregularities on the surface from file or machine marks may cause blocks where there are actual metal-to-metal contacts which will prevent flow of the brazing alloy. Too large clearances may break the capillarity through the joint and stop the flow of the brazing alloy.

The recommended clearances are made on the assumption that similar metals are being brazed and that the same clearance will be maintained when the joint is heated to brazing temperature. It is sometimes necessary with dissimilar metals to make allowances for differences in thermal expansion so that the desired clearances are obtained at the brazing temperature. For instance, in the case of a brass sleeve on a steel tube, the brass will expand more than the steel so that, to maintain a suitable clearance at brazing temperature, the clearance should be held on the minimum side or even to a drive fit when the parts are at room temperature. Conversely, if the brass fits inside the steel, the fit at room temperature should be looser, or else, when the parts are heated, the expansion may result in so tight a fit the brazing alloy cannot flow through.

The method of heating may also affect the clearance even though metals of similar thermal expansion are being joined. For example, in induction heating, the greatest heat is developed at the surface. If the heating coil is placed on the outside, the outer member will expand more than the inner member, and interference fits of as great as 0.005 in. are desirable. On the other hand, if the coil was placed inside the assembly, the reverse is true, and a clearance larger than normal would be necessary.

Preparation of Joint Surfaces. An important factor in getting good joints is to have the surfaces free from grease, oxide, scale, and dirt of any kind. Either chemical or mechanical methods may be employed.

Oil or grease is preferably removed by chemical means. There are many reliable cleaning compounds such as trichlorethylene, sodium phosphate, and others available for this purpose. Where only a few parts are to be cleaned, brushing the compound on will serve. If production warrants it, automatic machines for this purpose may be desirable. Emerying or other mechanical methods are not suitable means of removing oil or grease as they do not completely remove the film.

Oxides may be removed by a suitable pickle, the composition of the pickle depending upon the base metal. For copper or copper-base alloys a hot 10 per cent sulfuric acid pickle is usually satisfactory. Either dilute sulfuric or hydrochloric acid will work well on most steels. In some cases, such as nickel-base alloys or stainless steels, special pickles may be required. Directions for making and using such pickles can be obtained from the suppliers of the alloys.

Oxides or scale may also be readily removed by mechanical means. Filing, grinding, emerying, or sandblasting are commonly used. With sandblasting, care must be taken to wipe off any particles adhering to the surface as they will result in inclusions which will impair the strength of the joint. When filing is used, it should be a fine cut as deep scratches may cause trouble.

Deep scratches are to be avoided, for when the edges coincide on the two surfaces being joined, the fit may be too close to permit the free flow of the brazing alloy. Also, they may form channels through which the alloy will flow in preference to wetting the entire surface leaving excessive unbonded areas in the joint. On the other hand, a highly polished surface is also undesirable as the adherence of the brazing alloy is not so good as with a slightly roughened surface.

Parts should not be allowed to stand for extended periods of time after cleaning before brazing. Some metals build up oxide films on standing sufficient to interfere with the proper flow of the brazing alloy. In any case, dirt and other films may form on the joint surfaces.

Fluxing. With few exceptions, flux is required for silver brazing. Fluxes are used to prevent the oxidation of the brazing alloy and of the surfaces of the metals being joined during the heating. The flux will also dissolve oxides that may form during the heating operation.

The flux should be fluid, chemically active, and free from excessive fuming at the brazing temperature.

Borax and mixtures of borax and boric acid have been used as fluxes for silver brazing for many years. Borax does not have a definite melting point, but it is sufficiently fluid at 1400°F to make a good brazing flux. As the temperature drops below 1400°F, it gradually becomes more and more viscous. As the viscosity increases, the danger of entrapping flux in the joint also increases. The addition of boric acid increases the viscosity of the flux so that a 50 per cent borax and 50 per cent boric mixture is not fluid until a temperature of 1600°F is reached. Such a flux should be used only with the higher melting brazing alloys.

Borax is an excellent solvent for oxides. Boric acid is a poor solvent for oxides but because of its viscosity and imperviousness to air or other gases is a good flux for preventing oxidation. Fused borax left on the joint after brazing is relatively insoluble and therefore hard to remove. Fused boric acid is readily soluble in hot water, and mixtures of borax and boric acid are much more easily removed from the joint than borax alone.

With the development and extended use of brazing alloys melting below 1300°F, fluxes melting lower than borax were required in order to take full advantage of the lower brazing temperatures. Fluxes with a borate base but containing alkali fluorides have been developed which are fluid at temperatures above 1100°F, and they have largely displaced the older borax fluxes. The fluxes containing fluorides are particularly effective in dissolving refractory oxides, such as chromium oxides, and this has greatly broadened the use of silver brazing with stainless steel and alloy steels.

Special fluxes containing other halides are also available for difficult applications where the above fluxes are not satisfactory. A special flux is required, for instance, in brazing aluminum to steel.

Flux is usually used in a paste form thin enough so that it may be brushed over the surface. Brushing is preferable to other methods of application as the brushing tends to break down any films left on the surface from the cleaning operation to leave a thin, uniform coating over the entire surface. If the flux does not adhere readily to the surface during brushing, it is an indication that the previous cleaning was not complete.

Dipping in hot flux solutions is sometimes used but is not so satisfactory, as even clean pieces will usually show bare spots after dipping. These bare spots oxidize during heating and, as the brazing alloy will not flow on oxidized metal, are frequently the cause of unsound joints.

Dry flux is occasionally used. To get better adherence of the flux, the joint should be warmed before applying the flux. Complete coverage of the surface by this method is difficult, and it is not recommended.

Flux pastes are customarily made with water. The evaporation of the water during heating results in some bubbling and puffing of the flux. Where this is objectionable, the fluxes may be made of fused or water-free salts made into a paste with alcohol or some other organic vehicle that does not carbonize readily.

Assembling and Jigging. After fluxing, the parts to be brazed must be assembled. Where it is possible, it is advantageous to preplace the brazing alloy. Between flat surfaces, thin sheet or washers may be used. On tubular members, rings made of rectangular strip or round wire are used. These rings may be inserted in grooves cut in the tubes or laid on projecting shoulders. It is preferable to place the alloy inside the joint away from the direct source of heat and in intimate contact with the parts.

Otherwise, as the brazing alloy is usually of lighter gauge than the parts, it heats more rapidly than the assembly and may melt and fall off before the assembly is heated to the brazing temperature.

The brazing alloy is distributed through the joint chiefly by capillary action. However, gravity does help, particularly in the initial flow, and this should be considered in the placement of the alloy in the joint.

Many joints are so designed as to be self-supporting, but on some joints external jigging is required. This is an individual problem with each assembly and sometimes requires considerable ingenuity. The jig should be as simple and as light as possible. The contact area between the jig and the portion of the assembly adjacent to the joint should be kept to a minimum. A heavy jig in close contact with the joint will conduct sufficient heat away to make the heating of the joint to the brazing temperature difficult.

Heating. Several different methods of heating are used in silver brazing. The choice of the method depends on such factors as the size and shape of the pieces and the quantities to be brazed. Regardless of the source of heat, it is important that both members of the joint are heated uniformly and reach the brazing temperature at about the same time. In general, the shorter the heating time, the better, for with a long heating cycle the parts become oxidized excessively, and the flux becomes saturated with oxide so that it is no longer effective.

When a heavy section is being joined to a light section, most of the heat should be applied to the heavy part to avoid overheating and oxidizing the light section. The brazing alloy tends to flow towards the hotter section, and if one section is heated to the brazing temperature too soon, the brazing alloy may flow over the heated area rather than into the joint. Another consideration, in the case of joining dissimilar metals, is differences in heat conductivity. In brazing copper to steel, for instance, the steel with a low heat conductivity will heat rapidly, while the copper will tend to draw the heat away from the joint. This difference in heat conductivity may more than offset effects caused by differences in mass.

Torch Heating. The common types of torches used for silver brazing are air-gas, air-acetylene, oxygen-gas, and oxyacetylene. As a rule the air-gas torches give the least heat, and the oxyacetylene torches give the most heat. Air-gas torches are available not only for city gas but also for natural gas, butane, and propane.

In brazing, the size of the torch should be sufficient so that the required heat can be obtained while using a neutral or slightly reducing flame. The torch should be kept in motion to distribute the heat uniformly over the joint area. Preheating slightly away from the joint until the flux partially fuses and adheres firmly to the metal is advisable as a heavy blast of the flame in the initial stages may blow the flux off and leave bare, oxidized areas.

If the brazing alloy is fed by hand from the outside, the parts to be brazed should be heated slightly above the melting point of the brazing alloy before the alloy is applied. When this temperature is attained, the brazing alloy, previously coated with flux, is brought in contact with the joint and, by momentarily applying the torch to the brazing wire or strip, sufficient alloy is melted to fill the joint. The size of the wire or strip selected should be large enough so that not more than an inch or two of alloy needs to be melted. If the joint is uniformly heated, the alloy will flow over the entire surface from one point. Where long joints are being brazed, it may not be possible to have the entire joint at the brazing temperature at one time, and it will then be necessary to heat the joint in sections applying the brazing alloy at several points.

Bonding of the alloy with the base metals is practically instantaneous at the brazing temperature, and prolonged heating after flowing of the alloy should be avoided as the molten alloy will absorb gases from the flame. Absorbed gases are released when the alloy solidifies and may be trapped in the joint causing unsoundness.

Another common cause of poor brazes is the application of the brazing alloy before the work is at the proper temperature. When the work is too cold, the brazing alloy will "ball up" on the surface. The balling up tends to raise the brazing alloy above the protective film of flux exposing it to direct oxidation. The tendency is then to overheat in an attempt to flow the oxidized alloy, but instead of correcting the trouble, this exaggerates it. The proper method consists of cooling the joint sufficiently to permit applying more flux and then reheat carefully to the brazing temperature.

Torch heating can be adapted readily to automatic, high-production jobs by passing the work through a series of stationary burners on rotating tables or endless chains. Production of several hundred small parts per hour can be attained in this way with a relatively small capital investment.

Furnace Heating. Brazing is frequently done in standard types of heat-treating or annealing furnaces. Both batch-type and continuous-type furnaces, with or without protective atmospheres, are used. In general, the furnace is preheated to a temperature several hundred degrees above the melting point of the brazing alloy so that the parts are heated rapidly, and the time in the furnace is regulated so that the assembly is only heated sufficiently to properly flow the brazing alloy. Preplaced brazing alloy must be used and the parts jigged to hold them securely in place.

Reducing atmospheres are helpful in keeping the parts clean and in reducing the amount of subsequent cleaning. Flux is usually essential even in a reducing atmosphere, but the life of the flux is greatly increased. This is of material benefit when large pieces requiring prolonged heating to attain the brazing temperature are being brazed.

Induction Heating. Because of the rapid and highly localized heating that can be obtained with induction heating this method is being extensively used. Successful use of this method depends largely on the proper design and placement of the heating coils. The makers of the equipment are prepared to supply such coils and should be consulted for advice. This method merits consideration where there are large numbers of an item to be brazed.

With induction heating the heat is developed mainly at the surface and reaches the joint surfaces by conduction. As the heating cycle is very short, there is necessarily a steep thermal gradient between the surface and the joint area except in case of very thin sections. This results in a greater variation in the expansion of the parts than occurs with slower heating methods which must be considered in determining clearances. Contrary to the usual silver brazing practice, press fits are frequently used on assemblies where the coil is placed externally. The intimate contact between the parts also assists in the heating of the inner parts by conduction. Conversely, if the heating is done from the inside, larger clearances are required, or the expansion of the inner part will produce a fit at the brazing temperature too close to allow the brazing alloy to flow into the joint.

Salt-bath Heating. Molten salt baths are also used for heating. They have been particularly useful when a large number of closely spaced joints are to be brazed, as in radiators or heat exchangers. The best results have been obtained when the brazing alloy can be inserted in the joint and not directly exposed to the salt bath. While the salts used are chemically active in reducing oxides, it is customary to coat the joint surfaces with silver brazing flux to get more positive fluxing in the joint. When this is done, care must be taken to dry the flux thoroughly before dipping the assembly in the bath as any moisture present will react with the molten salts with explosive violence.

When the brazing alloy cannot be inserted in the joint but is placed in the form of rings on the outside, the assembly may be dipped into the salt up to the joint and the joint heated by conduction rather than directly from the salt. Another method consists of preheating by partial immersion and then total immersion an instant before the assembly is at the brazing temperature. If these methods are not followed, the light, externally placed ring will melt before the assembly reaches the brazing temperature and will fall off the joint.

One application of salt-bath brazing has been the combination of the case hardening and brazing in one operation in cyanide baths.

Alloy Baths. Molten alloy baths have been used for joining small parts when the control of the temperature and the localization of the heat is difficult by other methods The brazing alloy is melted in a thermostatically controlled crucible and covered with a suitable flux. The parts to be brazed are dipped through the flux into the bath and held there until the alloy flows onto the joint. Large quantities of fine enameled wire for electrical devices are brazed in this manner.

The composition of alloy baths will vary either by volatization of low-melting constituents, such as zinc, during prolonged heating or by solution of metal from the parts being joined. As this will change the melting point of the alloy, the composition of the bath must be controlled either by suitable additions or by occasional replacement.

Electrical-resistance Heating. Tongs have been developed in which the work to be brazed can be clamped between two carbon blocks. Current is passed through this circuit and the carbon blocks are heated to a red heat by resistance. This heat is transferred to the work primarily by conduction. The equipment may be either portable or stationary. It has the advantage of heating and cooling the joints under pressure, thus obtaining strong joints with the minimum amount of alloy. Jigging is usually unnecessary as the pressure clamps will hold the work in position. Heating is localized at the joint area and may be readily controlled.

Spot welders may also be adapted to this type of heating. With spot welders either copper or carbon electrodes may be used. Copper electrodes are usually used on parts having low electrical conductivity, and the resistance of the work itself is used to develop the heat. Carbon electrodes are used on high-conductivity materials where the resistance of the work is too low to generate the required heat. By judicious combination of copper and carbon electrodes, dissimilar materials of widely varying conductivity can be brazed. Tungsten tips may be substituted for the carbon electrodes when long life of the electrodes is desirable.

Flux must be used with this as with other silver brazing processes. As the work is in the electrical circuit and as dry flux is a poor conductor of electricity, the flux should be used as a thin paste and should not be allowed to dry out before the current is passed through the circuit. Otherwise, a heavy surge of current is necessary to break through the resistance offered by the flux, and this may even melt the parts being brazed.

Final Cleaning. Final cleaning consists of the removal of residual flux from the joint area and pickling to remove oxides formed on the parts. The flux is likely to corrode the parts if it is not removed. The corrosive effect is seldom sufficient to damage the assembly seriously, but it is unsightly.

Hot water is generally sufficient to remove the low-melting fluoride containing fluxes. Quenching the assembly after brazing below a red heat but while it is still warm will materially assist in cracking and flaking the fused flux off the work.

Flux that has become saturated with oxides during brazing is only difficultly soluble in water. A warm dilute sulfuric acid or hydrochloric acid pickle will assist in the removal when this occurs. However, such a condition is indicative of faulty brazing practice. It is the result of either too long a heating period, too high a brazing temperature, or the use of too little flux. More liberal use of flux on the outside of the joint will, as a rule, keep the concentration of oxide in the flux low enough to correct this condition.

Proper control of the amount of brazing alloy used will avoid the necessity of the removal of excess brazing alloy. This is one of the advantages of silver brazing in contrast to other processes where large quantities of excess metal are generally used, the removal of which makes the final cleaning cost a significant portion of the cost of making the joint.

CHARACTERISTICS OF SILVER BRAZING ALLOYS

Table 12-9 gives the composition and properties of a representative group of silver brazing alloys. Many other compositions are possible and have been used commercially, but with one or another of these alloys practically any silver brazing job can be done. The selection of a particular brazing alloy depends upon different factors, such as composition and melting points of the metals to be joined, requirements as to color, temperatures, and stresses and corrosion conditions that the joints will be subjected to in use. The cost of the alloy will vary with the silver content, but as the amount of alloy used in a joint is generally small, other considerations, such as the shorter heating time and less damage to the parts by the lower temperature, will often more than offset the greater cost of the alloys with higher silver content.

Pure metals will melt at a single temperature, but most alloys melt over a temperature range. The limits of this range are defined in Table 12-9 by the temperatures given in the columns headed Melting Point and Flow Point. The melting point is the temperature at which, on heating, a part of the alloy starts to become liquid. The flow point is the temperature at which the alloy flows freely. In between these two temperatures the alloy contains both solid and liquid and is in a slushy condition. The part of the alloy that is liquid is of different composition from the part that is solid.

The melting point is akin to what is technically known as the solidus, and the flow point is similar to the liquidus of the constitutional diagram of the particular alloy system. Actually, however, certain brazing alloys flow freely and are regularly used below the true liquidus.

In making a brazed joint the members joined should be heated above the flow point, and this is the temperature of main interest to brazing.

Table 12-9. Typical Silver Brazing Alloys

Number	Chemical composition						Melting point, deg F	Flow point, deg F
	Ag	Cu	Zn	Cd	P	Ni		
1*	10	52	38				1450	1565
2*	20	45	30	5			1140	1500
3	30	38	32				1370	1410
4*†	45	30	25				1250	1370
5*	50	34	16				1275	1425
6	60	25	15				1260	1325
7*	70	20	10				1335	1390
8	72	28	..				1435	1435
9*	80	16	4				1360	1490
10†‡	15	80			5		1185	1300
11†‡	50	15.5	16.5	18			1160	1175
12†‡	50	15.5	15.5	16		3.0	1195	1270

* ASTM standard alloy, Specification B73-29.
† Meet Federal government Specification QQ-S-5616.
‡ Proprietary alloys.

The length of the melting range varies widely with the alloy, in some cases being as much as several hundred degrees and in others only a few degrees. In torch brazing, when the alloy is hand-fed and almost instantly melted, the length of the melting range is of little consequence and has no material effect on the flow of the alloy. On the other hand, when other methods of heating are used and when the alloy is preplaced and is heated slowly through its melting range, along with the work, the length of the melting range is of importance. The part of the alloy to melt first is likely to flow into the joint leaving a lump of a higher melting alloy behind. As the temperature is raised, this process continues with the composition of the residue constantly changing. The composition of the residue may change to such a degree, and its flow point be raised to such an extent, even above the original flow point of the alloy, that the residue will not melt even at several hundred degrees superheat. The final result is that the lump is left on the joint after the brazing is completed, only part of the alloy is actually used, and a smooth fillet is not obtained.

With short-melting-range alloys the change from solid to liquid is almost instantaneous, even on slow heating, and the alloy flows more freely without the sluggishness of the long-range-melting alloys.

Both types of alloys have advantages. The short-melting-range alloys are preferable for slow heating, as in furnace brazing, because they are not so sensitive to changes in the rate of heating, and they will flow readily into very narrow spaces. They tend not to fillet well, however, and tend to run through the joint without filling gaps which the long-melting-range, more sluggish, alloys, will completely fill. Therefore, when close fits are not possible, the long-melting-range alloys usually give better results.

The melting point of the brazing alloy is sometimes of importance because of its effect on the strength of the joint just below the brazing temperature. A joint is weak during the period when any part of the alloy is molten. Therefore, on cooling from the brazing temperature or on any subsequent heating, care must be used not to strain the joint while above the melting point of the alloy.

Information is frequently sought regarding the tensile strength and other physical properties of silver brazing alloys in the forms in which they are supplied to the trade. Such information is of little or no value in determining the suitability of an alloy for brazing or for predicting the strength of a brazed joint. The strength of the completed joint is dependent on so many factors that initial physical properties of the alloy are of minor importance. A good joint depends in a large degree on design and brazing technique. The properties of the metals being joined are of major importance, as with the same brazing-alloy joint strengths varying from about 30,000 psi on copper to 130,000 psi on high-tensile-strength steels can be obtained. In general, the strength will vary almost directly with the strength of the metals being brazed.

Shear strengths of brazed-steel joints will be roughly 20,000 to 50,000 psi, again depending upon the design and the properties of the material being joined. The maximum load which can be applied to a joint in shear is limited by the load required to start stretching any member of the joint. As soon as stretching occurs, high stresses occur at the edge of the joint, and the joint will fail by tearing rather than by true shear. Increasing the shear length beyond that required to withstand this load will not increase the total load the joint will carry before failure.

The strength of joints will also vary with the temperature. Short-time tests on steel brazed with silver alloys indicate that at 600°F the joint strength is about 50 per cent of that at room temperature. At 800°F the strength is roughly 25 per cent of room strength, and at 1000°F only about 10 per cent of the original strength remains.

EFFECT OF BRAZING TEMPERATURE ON METALS JOINED

An important consideration in brazing is the effect of the brazing temperature on the properties of the metals being joined. The temperatures required are above the annealing temperatures of practically all nonferrous alloys, and these alloys will lose any hardness which has been produced by cold working. However, the annealing effects can be minimized by localized and rapid heating, and in any case the relatively low temperature required as compared to other welding processes decreases the danger of material impairment of the properties of the base metals.

Heat-treated carbon steels, such as the usual carbon tool steels, will also be softened at the temperatures required for silver-alloy brazing. It is possible, however, to select a brazing alloy from Table 12-9 with a melting point sufficiently high to permit heat-treating subsequent to the brazing operation. At times the brazing and the quenching of the steel from temperatures of 1400°F or above are combined in one operation. The success of this procedure depends largely on the stresses introduced during the quenching operation. As the joint itself is weak at these temperatures, the assembly should be designed so that only compressive stresses are applied to the joint. Similarly case-hardening low-carbon steels and brazing are occasionally successfully combined in one operation.

The structural alloy steels, such as SAE 4130, may be heat-treated prior to silver brazing and if brazed rapidly with one of the lower melting silver brazing alloys will retain practically all the hardness and strength introduced by the heat-treatment.

Heat-treated and tempered high-speed tool steel may also be brazed without appreciable loss in hardness by means of silver brazing alloys melting below 1200°F. Silver brazing is frequently used for repairing broken tools or for brazing high-speed-steel tips to low-carbon-steel shanks. Although the usual tempering temperature for high-speed steel is about 1050°F, the softening of the steel is a time-temperature reaction, and with a short heating cycle it is entirely practical to braze the steels at temperatures in the vicinity of 1200°F without adversely affecting the hardness.

Silver-alloy brazing is used extensively for brazing tungsten-carbide tool tips. In joining tungsten carbide to steel stresses of sufficient magnitude to crack the carbide are often developed because of the difference in thermal expansion of the carbide and the steel. At the low temperature required for silver brazing these stresses are lower than when other methods of joining employing higher temperatures are used, and the losses resulting from the cracking of the carbide are largely eliminated.

Among the nonferrous alloys the brazing of the precipitation hardening beryllium-copper alloys is interesting. The usual procedure for hardening these alloys is to anneal at about 1500°F to form a complete solid solution, quench, and then reheat at a temperature of 500 to 600°F for 1 to 2 hr to precipitation-harden. It has been found that, after the quench from the high-temperature, beryllium-copper can be quickly brazed with a silver brazing alloy at about 1200°F and that the brazed assembly can then be precipitation-hardened by annealing at 500 to 600°F to the same extent as with the usual procedure.

CORROSION

Some reference should be made to the corrosion resistance of silver brazing alloys. They are, in general, highly resistant to the majority of corrosive conditions for which nonferrous alloys are used. Wherever dissimilar metals are used, galvanic corrosion is an important problem. Silver alloys, particularly those of high silver content, are cathodic to many metals and alloys in use under corrosive conditions. As galvanic corrosion is generally in proportion to the relative areas exposed to attack and as the surface exposure of the silver alloy in a properly made joint is small, the attack on the brazing alloy is normally negligible. In addition, the length of the brazed joint is usually long as compared to the surface exposure, and this is a favorable factor in increasing the life of a joint even under adverse conditions.

There are a few conditions which should be noted in which certain silver brazing alloys are seriously attacked. For instance, phosphorus-bearing brazing alloys should not be used where they are exposed to hot gases containing sulfur-bearing compounds as the brazing alloy will gradually disintegrate. Also stainless steels subjected to brines or chlorine compounds, as in marine fittings or dairy equipment, will fail if brazed with most grades of silver brazing alloys. Fortunately, in this case, the addition of a few percent nickel to the brazing alloy will overcome this difficulty and an alloy such as No. 12 in Table 12-9 will give excellent corrosion resistance. Finally, nitric acid in almost all concentrations is a solvent for silver brazing alloys, and the alloys should not be used in solutions containing nitric acid.

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GAS WELDING AND CUTTING

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Although frequently called gas welding and cutting, the process should correctly be called oxyacetylene welding and cutting since oxygen and acetylene are the gases usually used. Other combinations of gases, such as acetylene and air, natural gas and oxygen, or natural gas and air, do not produce flames that are nearly so hot. Consequently they are not often used for welding or cutting but only for heat-treating, brazing, or soldering.

OXYACETYLENE WELDING

Basically, oxyacetylene welding is the bringing together of two pieces of metal and the melting of the edges in contact by the oxyacetylene flame produced at the tip of the welding blowpipe. The molten metal flows together from the two edges until they are completely fused and form one continuous piece after the metal has cooled.

The oxyacetylene flame is also used for heating in cutting, shaping, and flame machining of ferrous metals and as a convenient source of localized heat for a wide variety of operations such as hard facing, flame hardening, flame softening, and flame descaling.

Principles of Operation. With suitable equipment, the operator can produce at will an oxyacetylene flame of the proper size and character for the work at hand. By changing slightly the proportions of oxygen and acetylene used, the chemical characteristics of the oxyacetylene flame and consequently its action on molten metal can be varied over a wide range.

Types of Flames. When an exactly 1-to-1 mixture of oxygen and acetylene is lighted at the blowpipe tip, the resulting flame is called a neutral flame because it neither oxidizes nor carburizes the weld metal. The neutral flame has two sharply defined zones. The inner portion of the flame consists of a brilliant cone from 1/16 to 3/4 in. long, depending on the size of the opening in the tip. Surrounding this is a larger "envelope flame" only faintly luminous and of a delicate bluish color.

When there is slightly more acetylene in the mixture, the flame consists of three zones. There is still a sharply defined inner cone and the bluish outer envelope, but between these is an intermediate cone of whitish color. This flame is variously called an excess acetylene, a reducing, or a carburizing flame.

When there is more oxygen than acetylene in the mixture, the flame has the general appearance of the neutral flame, but the inner cone is shorter, has a sharper point, and acquires a purplish tint. The three types of flame are shown by Fig. 12-29.

Flame Adjustment. It is usual practice when lighting a blowpipe to open the acetylene valve more than the oxygen so that the flame first shows an excess of acetylene. As the blowpipe acetylene valve is closed gradually, the excess-acetylene cone gets smaller and finally disappears completely. The neutral flame is formed just at the point where the excess-acetylene cone disappears. In adjusting for the type of flame desired, it is usual to adjust it first to neutral and then to make the necessary adjustment for the excess-acetylene or excess-oxygen flame as required.

The amount of excess acetylene in the flame is determined by comparing the length of the acetylene feather with the length of the inner cone, measuring both from the blowpipe tip as shown by Fig. 12-30. For instance, a 2X excess-acetylene flame has an acetylene feather that is twice as long as the inner cone; a 3*X* excess-acetylene flame has an acetylene feather that is three times as long as the inner cone.

Similarly, the oxidizing-flame adjustment is sometimes given as the amount by which the length of the neutral inner cone should be reduced, *e.g.*, one-tenth. Starting again with the neutral flame, the operator increases the oxygen (or decreases the acetylene) until the length of the inner cone has been shortened to the desired amount. In some cases the exact adjustment has to be determined by the action of the flame on the molten metal.

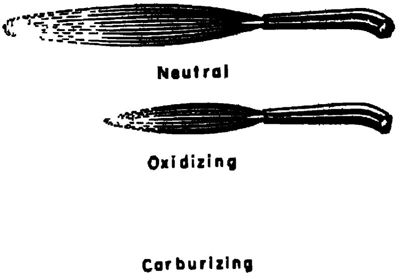

Fig. 12-29. The three types of oxy-acetylene-welding flames.

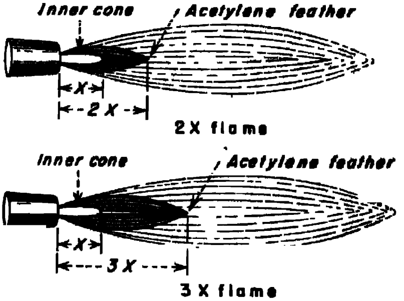

Fig. 12-30. Determining the amount of excess acetylene in a flame.

For welding mild steels, a neutral flame is desirable. For higher carbon and alloy steels, the carburizing flame may be preferable so as to preserve the alloying elements from oxidation. An oxidizing flame is generally used in the fusion welding of bronze and brass to assist in the formation of the protective oxide film. Use of a slightly oxidizing flame in brazing must also take into consideration the possible undesirable effect on the tinning action.

Welding Rods. Because any metal added during welding must have the same properties as the base metal, welding rods of various composition have been devised for

Table 12-10. Welding Data—Ferrous Metals

Metal	Welding method	Flame adjustment	Flux	Welding rod
Steel, cast	Fusion	Neutral	No	Steel
Steel pipe	Fusion	Neutral	No	Steel
Steel plate	Fusion	Neutral	No	Steel
Steel sheet	Fusion	Neutral	No	Steel
	Bronze weld	Slightly oxidizing	Yes	Bronze
High-carbon steel	Fusion	Carburizing	No	Steel
Manganese steel	Fusion	Slightly oxidizing	No	Base-metal composition
Cromansil steel	Fusion	Neutral	No	Steel
Wrought iron	Fusion	Neutral	No	Steel
Galvanized iron	Fusion	Neutral	No	Steel
	Bronze weld	Slightly oxidizing	Yes	Bronze
Cast iron, gray	Fusion	Neutral	Yes	Cast iron
	Bronze weld	Slightly oxidizing	Yes	Bronze
Cast iron, malleable.	Bronze weld	Slightly oxidizing	Yes	Bronze
Cast-iron pipe, gray.	Fusion	Neutral	Yes	Cast iron
	Bronze weld	Slightly oxidizing	Yes	Bronze
Cast-iron pipe	Fusion	Neutral	Yes	Cast-iron or base-metal composition
Chromium-nickel.	Bronze weld	Slightly oxidizing	Yes	Bronze
Chromium-nickel	Fusion	Neutral	Yes	Base-metal composition
Steel castings				25-12 chromium-nickel steel
Chromium-nickel steel (18-8) and (25-12)	Fusion	Neutral	Yes	Columbium stainless steel or base-metal composition
Chromium steel	Fusion	Neutral	Yes	Columbium stainless steel
Chromium iron	Fusion	Neutral	Yes	Columbium stainless steel

Table 12-11. Welding Data—Nonferrous Metals

Metal	Welding method	Flame adjustment	Flux	Welding rod
Aluminum	Fusion	Slightly carburizing	Yes	Aluminum
Brass	Fusion	Oxidizing	Yes	Bronze
	Bronze weld	Slightly oxidizing	Yes	
Bronze	Fusion	Neutral	Yes	Bronze
	Bronze weld	Slightly oxidizing	Yes	
Copper (deoxidized)	Fusion	Neutral	No	Deoxidized copper
	Bronze weld	Slightly oxidizing	Yes	Bronze
Copper (electrolytic)	Fusion	Neutral	No	Copper
	Bronze weld	Slightly oxidizing	Yes	Bronze
Everdur bronze	Fusion	Slightly oxidizing	Yes	Base-metal composition
Nickel	Fusion	Slightly carburizing	No	Base-metal composition
Monel metal	Fusion	Slightly carburizing	Yes	Base-metal composition
Inconel	Fusion	Slightly carburizing	Yes	Base-metal composition
Lead	Fusion	Neutral	No	Base-metal composition

all sorts of ferrous and nonferrous welding. Obviously, it is important that the correct welding rod be selected for any particular welding problem. Allowances are made in the chemistry of good welding rods for the changes that take place in the welding process, so that the deposited metal will be of the correct composition.

Fluxes. No flux is required for the fusion welding of mild and low-alloy steels, but with most commercial nonferrous alloys containing such readily oxidizable ingredients as copper, zinc, aluminum, chromium, or silicon, a flux must be used (see Table 12-10 and Table 12-11 for details). Fluxes should not be substituted for proper cleaning in the preparation of the base metal. The flux should be easily removed upon completion of the work and should have no corrosive action on the finished joint. Fluxes are used to accomplish the following effects:

1. The flux should form a fusible slag with the oxides formed during welding and should float to the top of the molten puddle.
2. The flux should protect the molten puddle from reaction with the gases of the surrounding atmosphere.
3. The flux should clean and protect the surfaces of the base metal and, in some cases, the welding rod during the preheating and welding or brazing.

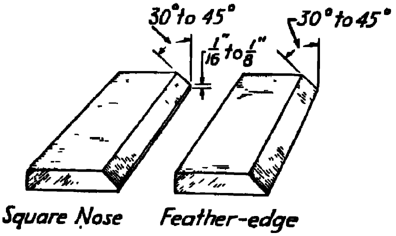

FIG. 12-31. Plate material 1/4 in. or more in thickness is beveled so that a V is formed.

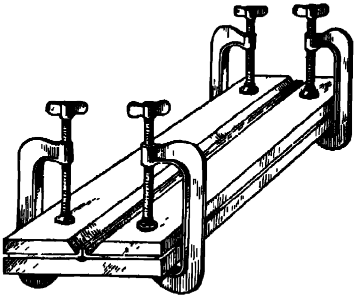

FIG. 12-32. Jig for making butt or flange welds in sheet metal.

Base-metal Preparation. The areas to be welded should be cleaned of scale and dirt of any sort. The edges of abutting plates, which are 1/4 in. or more in thickness, are beveled so that a V is formed as shown by Fig. 12-31. This is done to ensure complete penetration of the fused portion through the base metal. The angle of the vee opening varies from 50 to 90 deg, and a square nose is usually left at the bottom of the joint, although a feather edge is permissible for some work. In plate more than

½ in. thick, a double-vee opening on both sides of the plate is used. Thin sheet is often joined by flanging the edges, butting the flanges against one another, and fusing the flanges, generally without the addition of welding rod. A jig for holding the flanged sheets during welding is shown by Fig. 12-32.

Other joint designs, such as a U contour, are possible, but more expensive to prepare than the simple vee, which can be produced rapidly by flame cutting.

Bronze Welding. Bronze welding is a method of joining iron or steel without fusing the base-metal edges. Consequently, the disadvantages that may occur from bringing the base-metal edges to fusing temperature do not exist, or they are at least minimized. The bronze rod used in the process melts at about 1600°F. This is considerably below the melting point of many commercial metals. It is this fact that makes the process possible.

In bronze-welding, the veed edges are first thoroughly cleaned, then raised to a dull-red heat. In the presence of a suitable flux, the bronze metal from a rod is flowed on so as to form a tinned surface. Successive passes are made as necessary, until the joint is completed.

For joints made of malleable iron and galvanized iron, bronze welding is superior to fusion welding because the base metal is little affected. The malleability of cast iron therefore is preserved, and the coating of galvanized iron is affected only locally.

Multilayer Welding. Multilayer welding is a variation in welding technique that is recommended for joining heavy plate. It is accomplished by depositing rod metal in successive passes along the joint until it is filled. Since the "bite" taken with each pass is small, the weld puddle is reduced in size. The small puddle is more easily controlled, and thus the operator can avoid oxides, slag inclusions, cold shuts, and lack of fusion with the base metal. The major advantage of this process is, however, the increase in ductility of the deposited metal owing to the grain refinement resulting from the welding heat of each successive layer. The final layer, of course, will not possess this refinement unless the blowpipe is passed over the surface again to normalize the metal deposited in the last pass.

Oxyacetylene Welding Equipment. There are two general types of oxyacetylene blowpipes: medium and low pressure. Low pressure blowpipes make use of what is known as the injector principle. A cross section of this type of blowpipe is shown by Fig. 12-33. The oxygen passes through a small opening in the injector and produces a suction which draws acetylene into the oxygen stream. In the medium-pressure blowpipe, cross-section diagram shown by Fig. 12-34, the oxygen and acetylene are regulated by valves in the torch to approximately the same pressures. The mixing chamber serves only to blend the gases thoroughly before they enter the tip.

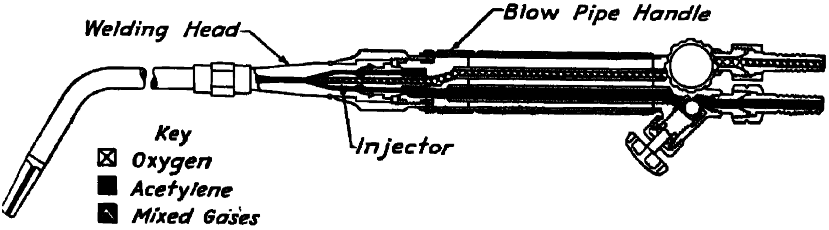

FIG. 12-33. Section diagram of injector-type welding blowpipe. It can operate on low- or medium-pressure acetylene.

Selection of Torches. There are a number of models of oxyacetylene-welding blowpipes. The choice of a blowpipe depends on the nature of the work to be done. For very small flames necessary on thin sheets or other delicate work, there are tips available having orifices as small as 0.018 in. The acetylene flow through these tips is approximately 1 cu ft per hr, and the blowpipes with which the tips are used are very small and light in weight.

Where large sections are being welded and a great deal of heat is needed, tips with orifices as large as 0.250 in. can be secured. Also, there are multiple-flame tips that have acetylene consumptions as high as 250 cu ft per hr. These multiflame tips are more often used for heating than actual welding operations.

Machine Equipment. For the production of certain welded articles, machine welding has been found to produce welds that are better and more uniform in quality and, in most cases, more economical than hand welding.

The oxyacetylene welding machines are designed for three types of welding: fusion welding of steel with welding rod added, fusion welding of steel without the addition of welding rod, and bronze welding. The last type requires the use of a flux, and is more expensive than fusion welding, but possesses the virtue of greater speed as well as other advantages which have been discussed. The edges must be sheared correctly and held in contact during welding.

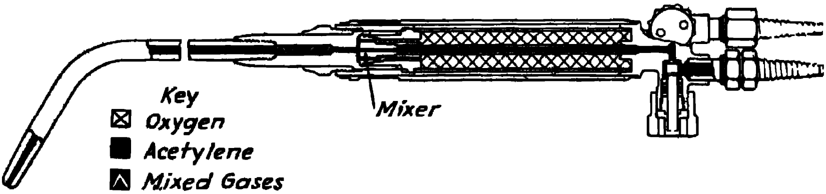

FIG. 12-34. Section diagram of welding blowpipe designed for medium-pressure operation.

Joint detail is very important in the fusion welding of sheet steel without the addition of welding rod. The edges must be sheared correctly and held in contact during welding. To do this, side pressure must be applied to the clamp jaws in order to force the edges together. To avoid deformation due to contraction stresses, jigs and clamps should be heavy enough to absorb heat from the workpiece quite rapidly. When production rates are high, or when items welded are small, water-cooled jigs should be used to absorb the heat quickly. This type of operation produces a flush weld with a minimum of reinforcement. Pieces with curved sides that cannot be made by rolling or bending flat sheets can also be machine welded. The parts may be stamped or drawn in two halves. These halves are then placed together in a jig and positioned so that the seams are in the same horizontal plane. The joints are then welded simultaneously with two welding heads moved by a common carriage. This method of welding will be somewhat slower because the horizontal-vertical position of the weld requires a smaller weld puddle.

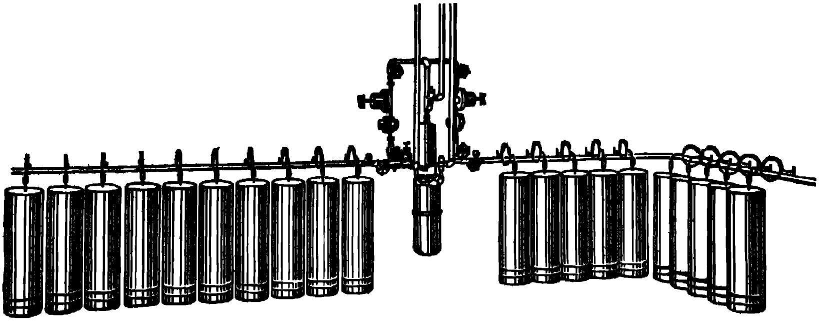

FIG. 12-35. Cylinders of acetylene connected to a manifold.

Gas Supply. Acetylene is distributed in steel cylinders containing a porous filler and acetone, in which the acetylene is dissolved. The most common cylinder sizes have rated capacities of 100, 275, and 300 cu ft.

The gaseous acetylene should not be allowed to discharge too rapidly from the cylinder because a fine mist of acetone may be carried along with the gas. To avoid such an occurrence, the rate of discharge per cylinder in cubic feet per hour should not exceed one-seventh the *rated* gas content capacity of the cylinder in cubic feet. Where the apparatus consumes acetylene faster than the recommended discharge rate for a single cylinder, the acetylene should be supplied by two or more cylinders discharging simultaneously through a suitable manifolding device as shown by Fig. 12-35.

Acetylene can also be made as it is used. Acetylene generators produce the gas by reaction between calcium carbide and water. There are two general types: the medium-pressure type produces acetylene at from 1 to 15 psi; the l-p type generates acetylene at less than 1 psi. Both types are available in a wide range of sizes for both stationary and portable service. A typical generator for stationary service is shown by Fig. 12-36.

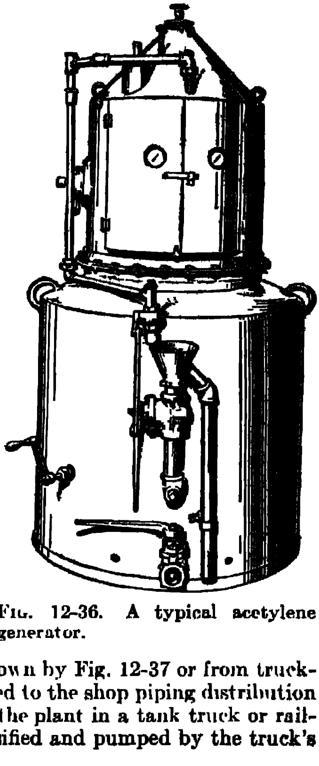

FIG. 12-36. A typical acetylene generator.

The small portable generators have carbide capacities of about fifteen pounds, with generating capacities of approximately 30 cu ft per hr. In contrast, some of the large stationary industrial generators have carbide capacities of 1 ton per hr with generating capacities of thousands of cubic feet of acetylene per hour.

Oxygen for commercial purposes is supplied in seamless steel cylinders. The gas is compressed to 2,200 psi, at 70°F. Two sizes of cylinders are available: the larger and more common holding oxygen equivalent to 244 cu ft at 1 atmosphere and 70°F, the smaller cylinder containing half this volume.

In order to supply the large quantities of oxygen required in some industries, the compressed gas may be obtained from an appropriate number of cylinders connected to a manifold as shown by Fig. 12-37 or from truck-trailer storage units, either of which can be connected to the shop piping distribution system. Oxygen in liquid form may be delivered to the plant in a tank truck or railroad tank car. The liquid oxygen may then be gasified and pumped by the truck's apparatus into the plant's storage facilities, or discharged as liquid into a converter which automatically furnishes gaseous oxygen to the plant's distribution piping system.

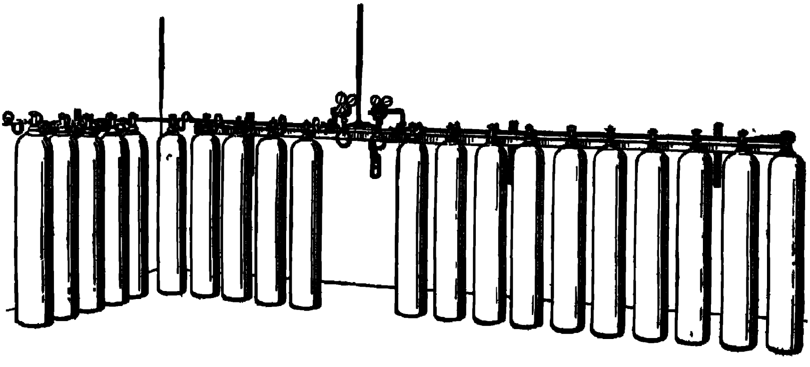

FIG. 12-37. Oxygen cylinders connected to a manifold.

Manifolds and Pipe Lines. Centrally located supply units with piping systems to conduct the gases to the point of use are a decided advantage, since cylinder and regulator handling is considerably reduced, cylinders are removed from working areas in the shops, a master control of available working pressure is obtained, and a continuous supply of gas is afforded.

The manifolds and pipe lines for oxygen are different from those for acetylene because the gases have such different natures. The oxygen system is generally operated at pressures from 50 to 100 psi. Pure oxygen at these pressures violently supports combustion. The pressure of the acetylene system is not greater than 15 psi, but the gas is extremely flammable.

At the blowpipe station, the high oxygen and acetylene pressures are reduced to working pressures by the use of regulators. These instruments automatically reduce the pressure to the desired value and maintain it there without fluctuation.

At all points where gases are drawn off to supply a blowpipe, or other consuming device, a regulator or hydraulic flash arrester is required for the acetylene if the gas is delivered by a manifold or from medium-pressure generators. In exceptional cases, where all the operators are doing the same work and using the same oxygen pressure, the oxygen station-regulator can be dispensed with and regulation secured at the manifold. For acetylene from l-p generators, a suitable waterseal or hydraulic flash arrester is used at each outlet.

The purpose of the water seals is to prevent the passage of oxygen back into the acetylene lines. The regulators at acetylene outlets also serve the same purpose. Water-seal devices should be installed where each important branch line joins the main.

Station outlets for welding and cutting should be provided with shutoff valves so that service regulators may be removed at will, either for repair or because regulators need not be provided as permanent attachments to all stations at once. Acetylene-station outlets are frequently provided with mechanical check valves in addition to regulators.

Oxyacetylene Welding Applications. Oxyacetylene welding is used on the whole range of commercial ferrous and nonferrous alloys. As in any welding process, however, physical dimensions and chemical composition must limit the weldability of certain materials and pieces.

In fusion welding, the temperature range through which the metal is taken is practically the same as that of the original casting procedure. The base metal in the weld area loses those special properties that were given to it by heat-treatment or working. Therefore the ability to weld such materials as high-carbon and high-alloy steels is limited by the equipment available for heat-treating after welding. However, such metals are commonly welded with success, when the size or complicated nature of the piece does not prohibit heat-treating operations.

For wrought-iron and "foolproof" or plain carbon steels, the welding procedure is straightforward and offers little difficulty to the operator. Sound welds are produced in other materials by appropriate changes in technique, heat-treating, preheating, and fluxing.

The oxyacetylene process can be used for welding thicknesses of metal up to that used in heavy piping and for problems normally encountered in maintenance and repair.

Iron and Steel. As has been indicated, the low-carbon, low-alloy steels, cast steels, and wrought iron are the materials most easily welded by the oxyacetylene process. Only a simple technique and no fluxes are required.

In oxyacetylene welding, straight-carbon steels over 0.70 per cent of carbon are considered high-carbon steels, and necessitate special care to maintain their particular properties. Alloy steels of the type called "air hardening" also require precautions to maintain their properties even though the carbon content may be 0.35 per cent or less. The joint area is usually preheated in order to slow the cooling of the weld by conduction into the surrounding base metal. This prevents the hardness and brittleness associated with rapid cooling. The operator should use a carburizing flame, and should be careful not to overheat the base metal and burn out the carbon.

For stainless and similar steels, several modifications in procedure are made. Because of the higher content of chromium, these steels have a lowered heat conductivity. Therefore, a smaller flame is recommended than for equal thickness of plain carbon steel. A neutral flame is used because the chromium easily oxidizes and a flux is used to dissolve oxides and protect the weld metal. A high-chromium composition welding rod supplies metal to fill the weld.

Cast iron, malleable iron, and galvanized iron all present particular problems in fusion welding of any sort. In cast iron, the gray cast-iron structure can be maintained through the weld area by use of a preheat, a flux, and an appropriate cast-iron welding rod.

Malleable iron cannot be fusion-welded. Remelting offsets its properties which depend entirely on a long and expensive heat-treating and working operation. Bronze welding affords an excellent method of joining without excessive heating. Therefore, it is recommended for malleable iron and for galvanized iron, in order to preserve the galvanizing. This method is also used in joining dissimilar metals, and in sheet-steel fabrication.

Nonferrous Metals. As is the case with the various ferrous alloys, the particular properties of each nonferrous alloy must be considered in the technique used for welding it. When the logical precautions are taken, little difficulty due to the nature of the metal should be encountered.

For instance, aluminum gives no warning by change in color prior to melting but appears to collapse suddenly at the melting point. Hence practice in welding it is required to learn to control the rate of fusion. Aluminum and its alloys also suffer from hot shortness. Therefore they should be adequately supported at all points heated during the welding. Finally, any exposed aluminum surface is always covered with a layer of oxide and forms a fusible slag which floats on the top of the molten metal.

In copper welding, allowances are necessary for chilling of the weld owing to the very high thermal conductivity of copper. Also, considerable distortion can be expected in copper because the coefficient of thermal expansion is higher than in other commercial metals. These characteristics obviously pose difficulties which must be surmounted in satisfactory welding.

These examples will suggest the type of problems encountered in oxyacetylene welding, as well as the solutions.

Surfacing Operations. In hard-surfacing, a layer of metal with the special properties desired is deposited from a welding rod and sweated on to the base metal surface. In this way, a very hard, abrasion-resistant surface of alloy may be spread on a mild steel part, and so prolong the part's usefulness. The hard-surfacing operation offers a simple, low-cost means of producing parts that have two to twenty-five times the life of steel under similar conditions of abrasion.

Broken edges, gear teeth, and mismachined surfaces can be built up by depositing layers of metal that are like the base in composition. The built-up surfaces can then be machined and heat-treated so that they will have the same properties as the original piece. This application makes it possible to reclaim all sorts of damaged pieces, and results in savings in part costs and replacement time.

Fields. The process is widely used in such fabricating industries as: sheet metal, tubing, aircraft, industrial piping, and automotive. It is also used gainfully in laying pipe lines and in shipyards, as well.

"*Maintenance and Repair.*" The one field in which the process is most widely used and accepted is in maintenance and repair, where its flexibility and mobility make for great savings in time and labor. Oxyacetylene welding has been shown to be particularly suited to the needs of steel mills, railroad shops, machine shops, and automobile-repair shops, as well as in shops devoted entirely to welding, whose main business may be in repair of many sorts of small industrial and household equipment.

OXYGEN CUTTING

Equipment and Operation. Although the process is variously called oxyacetylene cutting or flame cutting, "oxygen cutting" is the preferred technical term.

When iron or steel is heated white hot and a stream of pure oxygen is directed against it, the metal burns rapidly, producing the magnetic oxide of iron (Fe_3O_4) with

Table 12-12. Ferrous Metals Cuttable by Usual Methods

Name of metal	Carbon	Other alloying elements		
Low carbon iron rolled or wrought				
Charcoal iron	Up to 0.04			
Wrought iron	0.04 max			
Low-carbon steel rolled wrought or cast				
Rivet steel	0.10–0.15			
Flange steel	0.15–0.30			
Firebox steel	0.15–0.30			
Structural steel	0.15–0.35			
Steel forgings*	0.20–0.45			
Cast steel†	0.20–0.45			
Medium-carbon steel†	0.30–0.50			
High-carbon steel				
Plain carbon steel†	0.80–1.60	Manganese		
Spring steel†	0.45–0.90	0.30–1.20		
Tool steel†	0.85–1.10			
Copper-bearing steel		Copper	Nickel	
Plain		0.20–0.35		
Seamless	0.10–0.20	1.00 max	1.00	
Manganese steel		Manganese	Silicon	
Structural*	0.25–0.35	1.60–1.90		
Hadfield's*	1.00–1.50	10.0–14.0		
Silico-manganese†	0.50–0.60	0.60–0.90	1.80–2.20	
Silico-manganese†	0.55–0.65	0.60–0.90	1.80–2.20	
Silicon steels				
Structural†	0.25–0.40	0.50–0.80	0.20–0.30	
Transformer*	0.10 max		Up to 4.00	
Nickel steels		Manganese	Nickel	
Low-carbon	0.10–0.20	0.30–0.60	0.40–0.60	
Low-carbon‡	0.10–0.20	0.30–0.60	1.25–1.75	
Medium-carbon‡	0.10–0.25	0.30–0.60	3.25–3.75	
High carbon†	0.25–0.35	0.50–0.80	3.25–3.75	
Nickel-chromium steels				Chromium
Low-carbon	0.10–0.20	0.30–0.60	1.00–1.50	0.45–0.75
Medium carbon	0.15–0.25	0.30–0.60	1.00–1.50	0.45–0.75
Chromium alloy	0.10–0.20	0.30–0.60	1.50–2.00	0.90–1.25
Nickel alloy	0.17 max	0.30–0.60	3.25–3.75	1.25–1.75
Molybdenum steels			Molybdenum	
Chrome-molybdenum†	0.25–0.35	0.50–0.80	0.15–0.25	0.50–1.10
Nickel molybdenum	0.10–0.20	0.40–0.70	0.20–0.30	1.65–2.00 Ni
Nickel-molybdenum†	0.15–0.25	0.40–0.70	0.20–0.30	1.65–2.00
Chromium steels		Manganese	Chromium	Silicon
Cromansil†	0.25 max	1.05–1.40	0.30–0.60	0.60–0.90
Low-chromium*	0.15–0.25	0.30–0.60	0.60–0.90	
1% chromium†	0.35–0.55	0.60–0.90	0.80–1.10	
4–6% chromium†	0.25 max		4.00–6.00	
Chromium vanadium steels				Vanadium
Low carbon‡	0.10–0.20	0.30–0.60	0.80–1.10	0.15 min
High carbon‡	0.20–1.05	0.60–0.90	0.80–1.10	0.15 min
				Tungsten
Low tungsten steels	0.50–0.70	0.30 max	0.50–1.00	1.50–2.00
				Vanadium
Vanadium steel	0.90 max	0.70–0.95		0.15 min
		Aluminum		
Aluminum steel		10.00 max		

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* The metal should be preheated before cutting to prevent hardening of the metal in the face of the cut. Usual preheating temperatures range from 500 to 900°F. Preheating is used for certain grades to facilitate cutting. If not preheated, a heat treatment, furnace or otherwise, should follow to improve or recondition the structure of the cut surface, provided the material is kept warm between the time of cutting and heat-treatment.

† Material should be preheated as described under *.

‡ Heat-treating after cutting may be necessary if the carbon content is on the high side.

the liberation of a considerable amount of heat. The oxide formed is in the molten state and flows or is blown away, exposing more metal to the action of the oxygen. Because of the chilling effect of the surrounding metal, it is necessary to continue to supply heat at the point of cutting even though the cutting action liberates heat. The preheating flames of the blowpipe serve to raise the temperature of the metal to the reaction point and thereby supply the heat necessary to keep the reaction going.

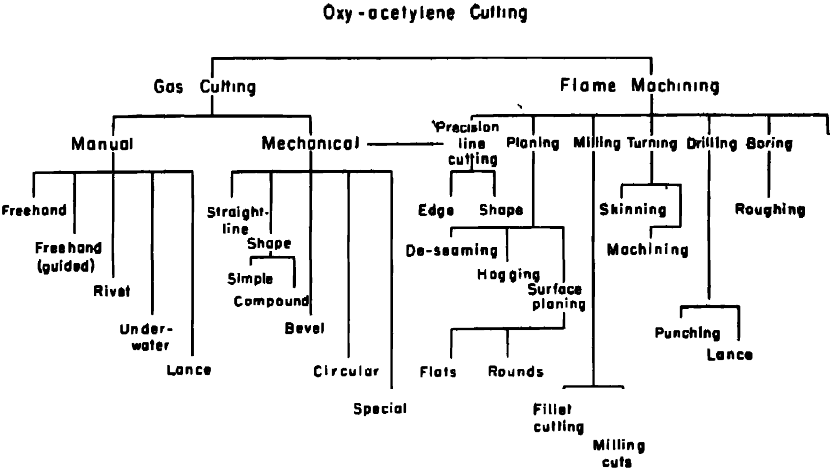

Fig. 12-38. The scope of oxygen-cutting processes is illustrated by this diagram.

Scope of Process. The variety of operations performed by oxygen cutting is shown graphically by Fig. 12-38. Only the more important of these operations will be discussed in this subsection. The chart will serve as a guide for obtaining further information if it should be needed. Carbon- and low-alloy steels are readily oxygen-cut by the ordinary straight-line technique. Table 12-12 lists the steels that can be cut by regular or special methods. Certain high-alloy steels, such as stainless steels,

Table 12-13. Ferrous and Nonferrous Metals Cuttable by Special Techniques

Name of metal	Carbon	Other alloying elements		
		Chromium	Tungsten	Vanadium
High-speed steel	0.60–0.90	5.00	10.–20.	2.00
Cast iron*		Silicon		
White cast iron	2.50 and up	1.50 max		
Gray cast iron	2.75–3.75	1.50–3.00		
Malleable cast iron	2.50	1.00		
Semisteel	2.00–3.00	1.50–3.00		
Duriron	0.75	15.00		
			Nickel	
High nickel	0.50 max		34.0 max	
Stainless steels†				
Low-carbon		Chromium		
Chromium	0.12 max	12.–18.		
High-carbon				
Chromium	0.30–0.60	12.–18.		
Chromium-nickel	0.05–0.20	16.–20.	7.–10.	
		Manganese	Chromium	
High-tungsten steels	0.50–0.70	0.30 max	3.–4.	

---

* The cut surfaces of the cast irons in heavy sections are hardened by the cutting operation and should be heat-treated if they are to be machined after cutting.

† Heat-treatment usually is necessary to counteract harmful effects of the heating produced during the cutting operation unless the metal contains a stabilizing element to prevent such heat effects.

show considerable resistance to oxygen cutting. These materials, as well as the nonferrous alloys can usually be cut by special techniques. A detailed list of the materials requiring special cutting techniques is given in Table 12-13.

Thicknesses and Tolerances. Speed ranges for various thicknesses of plain carbon steel, using different types of nozzles, are given in Table 12-14.

Table 12-14. Cutting-speed Ranges for Various Thicknesses of Metal

Thickness of metal, in.	Machine cutting speed, in./min		
	Standard cutting nozzles	Precision cutting nozzles	High-speed cutting nozzles
Up to ⅛	19-27		
⅛	18-26		
¼	17-25	16-21	
⅜	16-24	14-19	
½	15-23	13-18	26-28
¾	13-21	11-15	22-25
1	12-18	7-10	21-24
1½	11-15	6-8	14-16
2	10.0-14.0	5-7	12-15
3	7.8-10.9	5-7	11-14
4	6.4-8.9	4-6	10-13
5	5.4-7.4	4-5	8-11
6	4.7-6.5	4-5	7-10
8	3.7-4.9	3-4	6-9
10	2.9-3.8		3-7
12	2.4-3.0		3-6
16	3.0-3.5		
18	3.0-3.5		

The final tolerance on any oxygen-cut part is affected by three items: machine accuracy, oxygen-jet accuracy, and the distortion factor. The newer machines will follow a line or duplicate a pattern to within 0.015 in. The oxygen jet will remain within about 0.006 in. of a line so that the combined accuracy is not better than 0.021 in. The distortion factor is the allowance that must be made for the thickness of the material. The bottom of the cut may be out of line with the top of the cut by as much as 0.005 in. per inch of thickness.

Quality of Cut. The quality of the cut varies with the nozzle used, the speed of cutting, and the gas pressures. Shapes or parts that are to fit closely together with little or no finishing require high-quality cuts. In scrapping or demolition work, the object is merely to separate one part from another and so the cut surface is not important. Any discussion of the quality of an oxygen cut necessarily involves the consideration of "drag." "Drag" (sometimes called "lag") may be defined as the amount by which the bottom of the cut lags behind the top expressed as a percentage of the plate thickness. Straight-line cuts may be made with considerable lag without much detrimental effect, since the drag lines lie within the plane of the cut and do not prevent both top and bottom edges from being left clean and sharp. However, in shape cutting less drag is permissible, since the bottom of the contour would tend to come out differently from the top. Such cuts must be made with small drag and are known as "high-quality" cuts. High-quality cuts require somewhat more oxygen than cuts that have long drags and barely penetrate the material. Precision-machine cuts are necessarily of the high quality.

Equipment Required for Oxygen Cutting. The implements that permit the controlled use of the combined oxygen-jet and oxyacetylene preheat flames, is called a "blowpipe" or "torch."

Cutting Blowpipe. Blowpipes are of two types: first, those in which the fuel gas and oxygen for the preheating flames are mixed at the tip; and second, those in which the mixing takes place in the head or some location back of the head. This latter type is designated as the premixed type. If both the oxygen and acetylene are under appreciable pressure, the blowpipe is designated as a "medium-pressure" blowpipe. If the fuel gas is at a lower pressure, generally a fraction of a pound per square inch, an injector is necessary to mix it with the oxygen and the blowpipe is designated as a "low-pressure" or "injector" type. A cross section of a typical low-pressure blowpipe is shown by Fig. 12-39.

Machine-cutting Blowpipe. Machine-cutting blowpipes are usually straight and usually equipped with a rack that engages a pinion in the blowpipe holder, to adjust the height of the blowpipe above the work. Special heavy-duty machine-cutting blowpipes are available for cutting heavy sections or hot steel. Such blowpipes are usually water-cooled, operate on medium-pressure acetylene, and require a surprisingly low cutting-oxygen pressure.

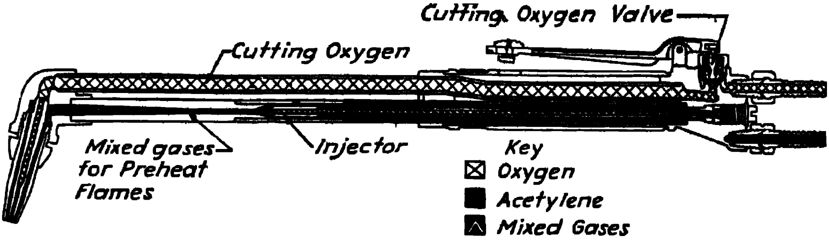

FIG. 12-39. Cross section of "low-pressure" or "injector"-type cutting blowpipe.

Cutting Nozzles. Simply by changing the size of nozzle, a single cutting blowpipe can be used to cut a wide range of metal thicknesses. The nozzles for the modern cutting blowpipes are made with the oxygen-cutting jet surrounded by several preheating flames as shown by Fig. 12-40. With this arrangement, the blowpipe can be moved in any direction without losing the effect of the preheating flame.

In addition to the standard cutting nozzles, there are several special-purpose nozzles. High-speed cutting nozzles, precision cutting nozzles, gouging nozzles, and rivet-cutting nozzles are examples of such special-purpose equipment.

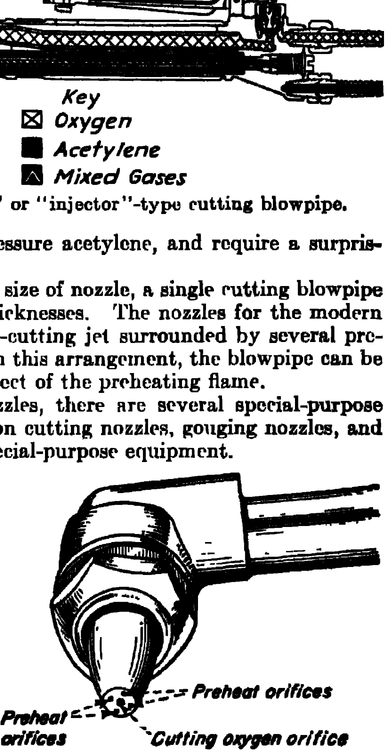

FIG. 12-40. The usual type of cutting nozzle for manual operation.

Cutting Machines. Cutting machines really consist of an oxygen-cutting blowpipe and a means of propelling it. They may be simple and light and usually of limited application or heavy, complicated, and quite versatile. In this subsection are illustrated a few of the typical oxygen-cutting machines selected to show the wide range available.

Small cutting machines weighing under 100 lb are usually classified as portable. Usually these machines are designed to make straight-line cuts or bevels. For this purpose, they operate on a track supplied with the machine. These machines are usually limited to operations on material up to 10 in. thick. By means of a radius rod, these machines can be used for cutting circles or operating directly on the steel plate. In an emergency simple, irregular shapes can be cut by hand-guiding the machine. Stationary cutting machines are particularly adapted to operations where the same pattern or design is to be cut repeatedly. Devices are available so that several blowpipes can be mounted on the same machine, and a number of parts of the same shape can be cut simultaneously. In any stationary shape-cutting machine, it is necessary to provide for motion in two directions at right angles to each other. All stationary shape-cutting machines are equipped with a tracing device, driven by a variable-speed motor and guided along the outline of the shape to be cut. There are three methods of guiding the blowpipe when using the stationary cutting machine. They are the line tracer, the strip templet, and the magnetic templet In line tracing, the tracing head is manually guided by the operator along the outline of the part The manually guided tracing head is shown by Fig 12-41 The accuracy of the cut depends on the

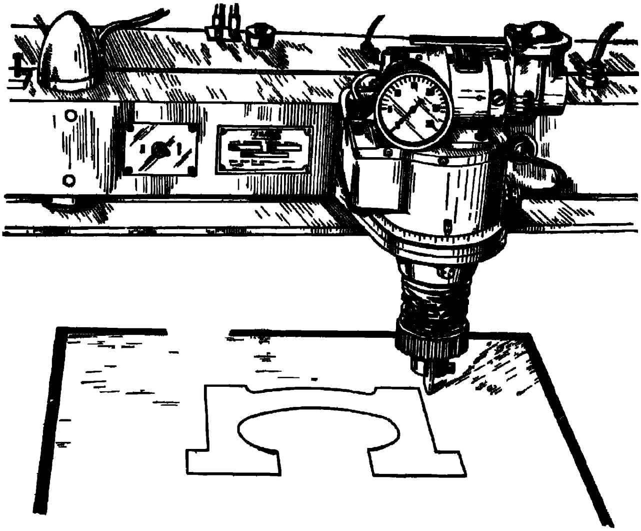

Fig 12-41 A manually guided tracing head

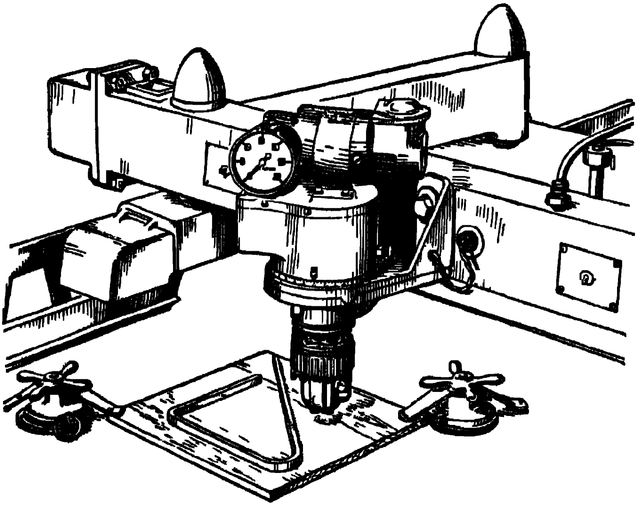

Fig 12-42 A strip templet guided tracing head

skill of the operator For strip-templet guiding, the templet is usually made of a $\frac{1}{4}$ in strip of aluminum The strips are bent to the outline of the part and firmly attached to a sheet metal or composition board base as shown by Fig 12-42 Such templets must be laid out carefully, but they can be used for making hundreds of successive cuts The tracers usually consist of knurled rollers operating on the sides of the templet A magnetic templet makes use of a magnetized knurled-roller that operates on the edge of a templet cut from steel plate about ½ in thick Templet and magnetic tracing head are shown by Fig 12-43

The Oxygen Lance The oxygen lance differs from the cutting blowpipe in that there is no heating flame to maintain the material being cut at the kindling temperature The oxygen lance is essentially a black iron pipe connected to a hose from a regulated supply of oxygen Once the cutting has started, the pipe itself becomes the fuel that burns and furnishes the heat necessary to keep the cut going The oxygen lance is used for rough-cutting heavy sections or for piercing holes in thick metal for the start of an oxygen cut

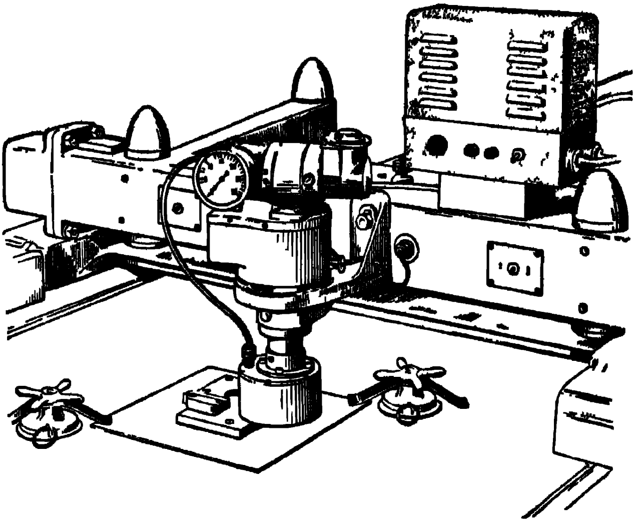

FIG 12-43 A magnetic templet guides the welding head

Preparations for Cutting. The work should be level on the worktable and so positioned that the piece can be cut properly at all points without cutting the work support The relative positions of templet and work can be checked in advance by manually moving the machine over the templet

Dirt, rust, paint, or scale should be removed from the surface before the cutting begins For precision cutting, a flame-descaling operation sometimes precedes the cutting itself

Manual Oxygen-cutting Operations. In addition to the usual precautions in regard to the fire hazards of hot slag and sparks, the operator should protect himself by wearing gloves, a hat, and welding goggles

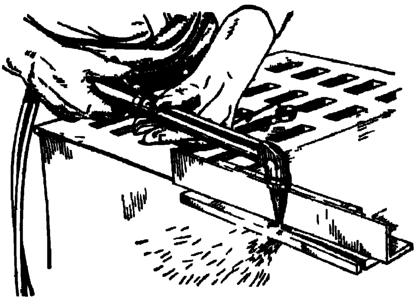

FIG 12-44 A piece of angle iron serves as a guide in making a manual cut

Most manual cuts are made freehand, however, when it is necessary to improve the accuracy, some sort of guide may be used A steel bar or a piece of angle iron will serve to maintain the correct distance above the plate and also to ensure giving a reasonably straight line Such a guide in use is shown by Fig 12-44 The technique of beveling parts for welding is similar to that for straight-line oxygen cutting In selecting the size of nozzle to be used, the metal should be measured along the bevel. The use of a piece of angle iron or other guide helps to make a straight bevel.

Circles and Irregular Shapes. Freehand cutting of circles and irregular shapes is done only when it is necessary to "rough out" a single piece. Circle-cutting attachments usually consist of a short rod with a clamp on one end that fits over the blowpipe head and an adjustable center point.

Piercing Holes. In piercing a hole in steel plate, the preheat flames are held about 1/32 in. above the spot where the hole is to be started. When the surface just begins to melt, the nozzle is raised slightly, and the cutting oxygen lever is slowly depressed. The blowpipe is moved slightly to one side and moved in a small spiral motion as the cutting action starts. When the cut has pierced all the way through the plate, the blowpipe should be lowered so that the tips of the preheat flames are about 1/16 in. above the plates, and the cut continued with a spiral motion until the desired diameter hole has been obtained.

Machine-cutting Low-carbon Steel. Mechanically holding and guiding the blowpipe over the work has many advantages over manual cutting, such as better workmanship and greater accuracy and efficiency. Cutting with straight-line cutting machines is even simpler than manual straight-line cutting. Also, many of the principles of this type of operation apply to other forms of machine cutting. The type of nozzle should be selected according to the quality of cut desired and the size of nozzle should be selected according to the thickness of metal to be cut.

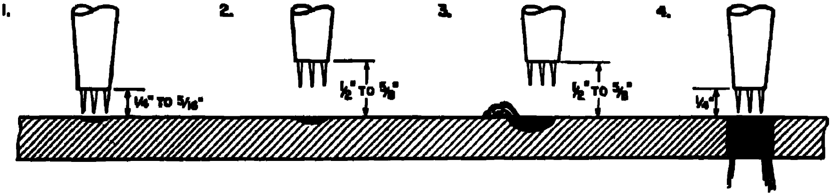

FIG. 12-45. Procedure for starting a machine cut.

Starting the Cut. The lighted blowpipe should be positioned so that the flames fall on the surface of the plate near the edge, with the tips of the inner cone about 1/32 in. away from the surface, shown by Fig. 12-45. When the metal is ready to melt, the nozzle should be quickly moved off the plate so that the center line of the nozzle is about 1/8 in. from the edge. At the same time the cutting oxygen should be turned on and the machine started. This method is generally termed a "flying start" and can be used successfully on thicknesses up to 6 in. or greater with practice.

A simple method of checking the smoothness of the cutting stream is to light the preheat flames, adjust them to neutral, and open the cutting valve. The stream of cutting oxygen can be clearly seen through the envelope of the flame and should be straight and true with a uniform diameter. The operator should check this before starting every cut.

Appearance of the Cut. In a good cut, the surface will be smooth and square, the drag lines will be vertical, and barely visible, the edges will be sharp and straight and any oxide present will be easy to remove.

A cut surface in which the drag lines are straight up and down is called a "drop cut." In straight-line cutting where economy as well as quality is to be considered, the speed of the machine may be increased so that less oxygen and acetylene will be consumed for a given length of cut. In general, cuts having a 10 per cent drag are satisfactory for average run of shop work and are considered standard as to quality.

Shape Cutting. The principles of cutting irregular shapes are the same as for straight-line cutting, with one important exception. Speeds and pressures must be adjusted for minimum drag, otherwise the contour on the bottom of the cut will come out differently from that on top. All shape-cutting machines require some sort of a guiding templet.

Circle Cutting. Nearly all oxygen cutting machines can be adapted to circle cutting. The straight-line machines usually use a radius rod with an adjustable center point. The shape-cutting machines can use either a circular templet of the proper size or a circle-cutting attachment. It is usual to make first a short "lead in" cut through the scrap, so as to have the finished cut smooth and unmarred at the starting point. Such cuts are usually started with a pierced hole. This is possible only when the machine is equipped with a racking facility for lateral adjustment. If racking facilities are not provided, the starting hole must be pierced or drilled tangent to the line of cut.

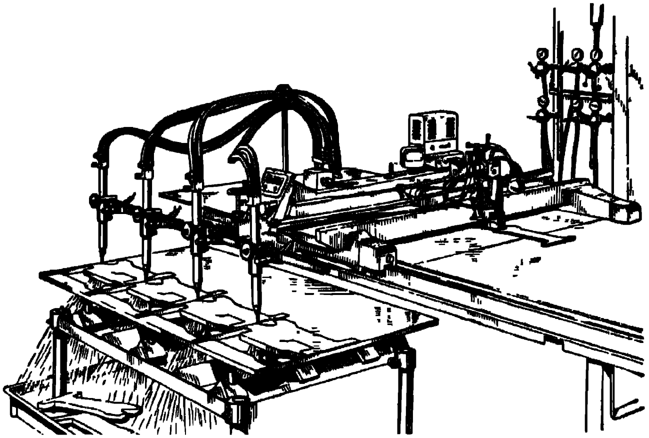

Fig. 12-46. Cutting identical parts with four blowpipes mounted on one machine.

Multiple-blowpipe Cutting. Many of the stationary shape-cutting machines are provided with means for mounting more than one blowpipe. In this way a number of identical parts can be cut at one time. This type of installation is shown by Fig. 12-46. Although only a single templet is required to guide the machine, consideration should be given to laying out the work and to reversing or reorienting the templet so as to produce a minimum of scrap.

Stack Cutting. When relatively thin plates are to be cut, the productive capacity of the machine may be increased many times over that obtained with multiple blow-pipes by piling the thin sheets one on top of the other, firmly clamping together and then cutting as a single piece of heavy material. Stacks with an over-all thickness not greater than 2 in. can be cut with an edge tolerance of $\frac{1}{32}$ in., while stacks having 2 to 4 in. thickness over-all, can be cut with an edge tolerance of $\frac{1}{16}$ in. Obviously, the thickness of the stacks must tie in with the economics of the individual application.

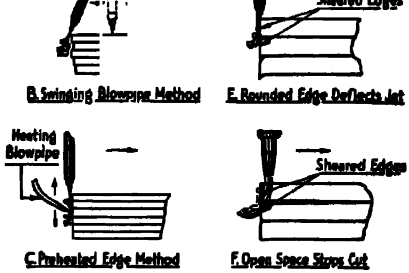

Fig. 12-47. Methods of starting the cut on stacked materials.

Starting the Cut. Starting the cut on stacked plates is the most important part of the operating technique. A faulty start usually damages the edge of several plates and makes a second start extremely difficult. To ensure against mishaps, several methods of starting have been devised as shown by Fig. 12-47, the recommended methods are shown at *A* and *C* but the methods in *B* and *D* are considered poor practice.

Plate-edge Preparation. At one time this work was done by using a single cutting nozzle and making individual cuts. The recently developed rip-and-trim method indicates the possiblity of increasing the speeds and reducing the costs of cutting while still maintaining the accuracy. This method employs two blowpipes: the first of which does the actual cutting, the second, when properly placed, follows and smooths up the cut. The speed of cutting ½-in. plate can be increased by 100 per cent with only about 10 per cent increase in cutting-oxygen consumption.

Multiface Cutting. By using two or more cutting nozzles, plates can be prepared for welding with a single bevel and nose, a double bevel, or a double bevel and nose. The success of this method depends on the positioning of the different nozzles in a manner so that one jet will not interfere with another and also so that the proper heat balance between the zones is maintained. Owing to the fact that the plate surface may be somewhat uneven with respect to the cutting machine, devices have been made to float the blowpipes on the plate surface so that the distance between nozzles and plate is automatically maintained constant.

SPECIAL CUTTING TECHNIQUES

The recently developed powder-cutting process makes it possible to cut cast iron, oxidation-resistant steels and nonferrous alloys with the results equivalent to those obtained in cutting low-carbon steels. In the oxygen cutting of stainless steels, the chromium immediately forms a highly refractory chromium-oxide on the faces of the kerf and prevents further oxidation. Powder cutting provides a means of melting or fluxing out the chromium oxide that has formed. In the reaction zone, the powder is elevated to the ignition temperature by heat from the oxyacetylene preheat flames, where it burns and creates an extremely high temperature reaction. The refractory oxides are continuously removed in a combination melting and fluxing action. The apparatus required for powder cutting consists of a dispenser in which the powder is placed and a means of introducing powder into the reaction zone. This apparatus is shown in operation by Fig. 12-48.

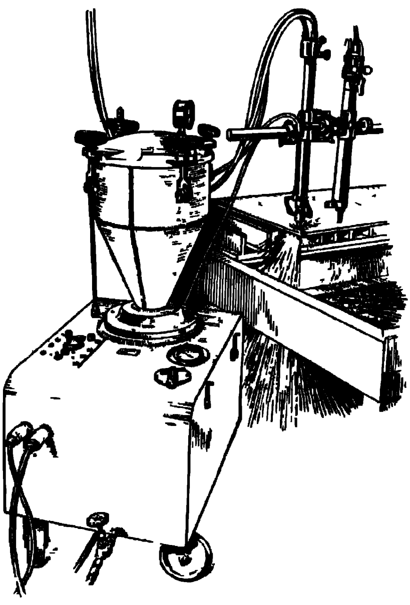

Fig. 12-48. Powder-cutting stainless steel.

Flux Cutting. With materials of low cuttability, the slag formed usually does not flow freely. Frequently, such materials can be cut easier if the slag is made more fluid. This can be done by mixing the slag with slag from materials of high cuttability, or by the addition of certain materials that will produce the same results. This process is known as flux-cutting. A common example of flux-cutting is the hand-feeding of steel welding rod into the cut during the cutting action.

Manual Cutting of Cast Iron. The technique for cutting cast iron differs from that for steel in that a high preheat is required and the cutting nozzle moves back and forth across the line of cut instead of back and forth along the line of cut as for stainless steel.

For determining the nozzle size and the recommended pressures to be used, a certain thickness of cast iron should be considered the equivalent of twice the same thickness of steel. The preheat flames should have an excess of acetylene and the inner cone should be held about 1/4 in. above the surface of the casting. Cast iron can be cut readily by the powder-cutting process.

OXYGEN GOUGING AND MACHINING

In the following applications of gouging and machining, large-volume low-velocity jets of oxygen are used. The purpose is not to cut or pierce but to remove metal from the surface of the piece. Oxygen gouging is a rapid and economical method of forming U-shaped grooves by removing the surface or edge metal from rolled, drawn, forged, or cast steel. Among the principal uses are the progressive gouging of the underside of electric arc welds, spot gouging for the removal of defective weld metal, or tack welds. Mechanized gouging can be used where joint specifications require a U- or J-edge preparation. The flame-gouging blowpipe is mounted on a mechanical device so as to provide a more uniform rate of speed than is possible with manual operation.

Oxygen Machining. With suitable nozzles, the oxygen-cutting process can be used to remove strips or linear areas from the surface of steel stock. The dimensions of the cut taken depend on the size of the cutting-oxygen orifice, the velocity of the cutting oxygen, the rate of travel of the nozzle with respect to the work, and the angle of inclination of the nozzle over the surface of the work.

Effect of Oxygen Cutting on the Cut Edge. Tests have shown that, unless proper precautions are taken, the effect of oxygen cutting on high-carbon steel and certain alloy steels is likely to be detrimental. A thin layer of hardened zone may be produced on the surface when such steels are oxygen-cut at room temperature. This brittle condition, however, is readily overcome by proper heat-treatment. If the high-carbon or alloy steels are preheated to a temperature of about 500 or 600°F before oxygen-cutting, cracking will not occur. Preheating should not be carried too far, as this may have an adverse metallurgical effect. When preheating is used before oxygen-cutting, the majority of steels do not require postheating.

Flame Softening. Flame softening employs a multiple-flame oxyacetylene head that is manipulated about the cutting zone to effect the desired heat-treatment. Flame softening furnishes a rapid and economical means of preventing or eliminating the hardening of higher carbon and alloy steels caused by flame cutting.

SECTION 13

INDUSTRIAL-PLANT POWER DISTRIBUTION AND LIGHTING

BY KENNARD PINDER

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INDUSTRIAL-PLANT POWER DISTRIBUTION AND LIGHTING

BY KENNARD PINDER

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PART 1. INDUSTRIAL-PLANT POWER DISTRIBUTION

Introduction

The production and utilization of electric power has had a very great influence in shaping the present social structure. The introduction of mechanical power brought a centralization of industry near the source of its raw material and power. As mechanical power does not lend itself readily to transmission of power over long distances with good economy, workers found it necessary to congregate in large numbers at the source of power with the result that the older industrial plants find themselves in congested areas with high land costs and taxes and labor difficulties.

Electricity has grown to be the most important form of power. It has been said that the nation which puts in the hands of its people the most abundant supply of electricity will have the highest standard of living. Owing to its flexibility and economy in industrial application and its transmission from the central generating station to the worker, electrical energy permits industry to decentralize and enjoy the advantages of abundant power. Large industries and certain types of process industries sometimes find it profitable to generate their own power, but a large percentage of industrial plants are supplied through a utility company's system with its central electrical generating stations and its radiating network of transmission and distribution lines, substations, and secondary feeders.

Modern industrial executives have come to the realization that their power plant or purchased-power substation and the associated electrical distribution system and power-utilization equipment is an important part of their manufacturing plant. One essential and important difference exists between other production departments and the electrical department. Electrical power, whether the product of the plant power plant or a utility service, is not completed until the instant it is needed, and then only in quantities exactly equal to the instantaneous demand. There is no simple way to warehouse an extra supply of kilowatthours against some future period of extraordinary demand. For this reason the industrial-plant electrical system, in general, must function as a unit and not as a collection of individual pieces of equipment. This statement is significant, as it governs the method of treatment of the subject in this section. Industrial electrical engineering, as construed here, does not include the theory and design of a generator, transformer, motor, or control equipment, but rather the selection of a manufacturer's product with a specific service in mind. There are many excellent textbooks, handbooks, and other technical literature covering the theory of design and operation of electrical equipment.

Almost all industrial plants are tailor-made, and their layout is usually the work of many engineers, designers, draftsmen, operators, etc. One of the principal functions in the design involves industrial electrical engineering, which may be defined as the art of selecting and arranging the necessary electrical equipment and wiring so that (1) a maximum of return will result from a minimum of expenditures over the working life of the plant, and (2) the operation of the completed plant will be such as to provide an economical, reliable, safe, and continuous service.

In weighing the relative merits of one arrangement of electrical equipment and wiring versus another, two items must be considered: (1) that all engineering problems, when solved, are in effect compromises—some desirable characteristics may have to be sacrificed in order to gain something else more desirable at the moment; and (2) most important, the ever-present economic problem goes hand in hand with every engineering problem. Economic justification is the principal factor governing the choice of equipment and the type of electrical layout. Construction costs have risen sharply in the last few years, the greatest increase being in the direct field labor. It is therefore very important to consider and evaluate new design and construction techniques with a view toward improving reliability and reducing costs at the same time.

The subject of "industrial-plant power distribution and lighting" is so large and the factors involved are so variable that it is possible only to cover the major items in a general manner. The scope of the work involved in the design, operation, and maintenance of the electrical system and equipment of an industrial plant is shown in Fig. 13-1.

There are many factors which govern the application of sound engineering principles and procedures in the design of the electrical system. Table 13-1 gives a list of the main factors involved and indicates the type of information that must be assembled preliminary to the design if the electrical system is to prove entirely adequate for the present and reasonably adequate for the future.

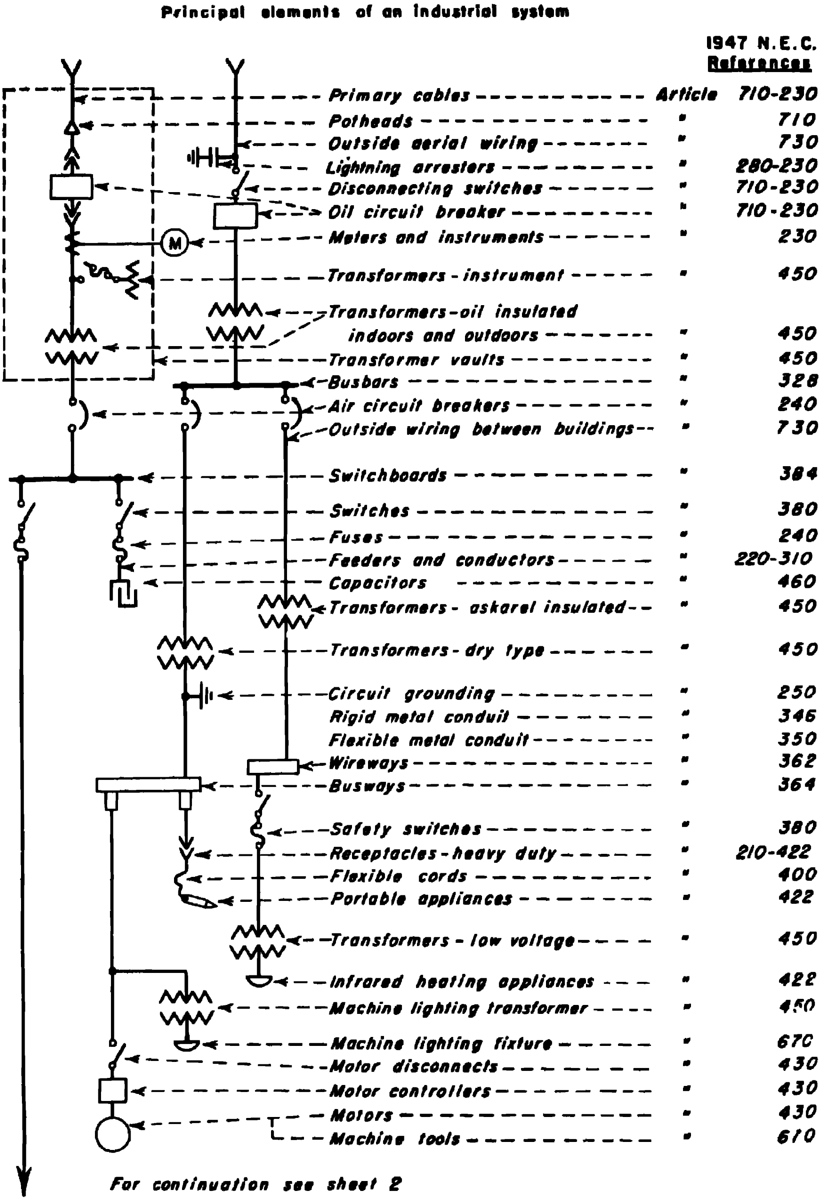
Table 13-1. Factors Governing the Design of the Electrical System for Industrial Plants

1 Character of the plant
    a Kind of product manufactured and physical arrangement and layout of plant
    b Process or material flow diagram showing the magnitude and location of the principal power-consuming equipment
    c Areas where standard type electrical equipment and construction may be used
    d Areas where dripproof, vaporproof, or explosion (vapor or dust) resisting equipment and construction are required
    e. Time allowed for design construction, and procuring materials
2 Size and characteristics of the load
    a. Total connected load—lighting, power, heating, etc.
    b. Probable future load changes and increase in load indicating approximate time when changes and additional capacity are anticipated
    c Maximum kw or kva demand
    d. Power factor and load factor of load
    e. Space available for locating lines, transformers, switchgear, controllers, etc.
3 Type of power supply to plant
    a Is power to be generated locally or purchased
    b If purchased, data are necessary on available power supply, including number and type of primary feeders, their voltage and kva ratings and short-circuit kva which can be delivered over each circuit when each is faulted at plant
4 Service requirements
    a. Importance and degree of reliability required
    b What processes in the plant must be kept operating. (This is where a power failure would ruin the product or create hazards of explosion, dangers to workmen, etc)
    c. If interruptions can be permitted to entire plant, or various types of operations, what is the maximum length of time that may elapse before the reestablishment of service?
5 Technical ability of the operating and maintenance department
    a. Large plants. May be economically staffed with men possessing adequate technical ability and maintenance and operating experience
    b. Small plants. May be necessary to consider operation by persons having scant technical knowledge and possibly but little practical experience

Fundamentals of Industrial-plant Wiring

Electric power is essential for modern industrial-plant operation. The larger the plant, the more significant becomes the selection of the wiring system to supply the energy to the machine tools, process-equipment drives and control, and the plant lighting; but, regardless of the size of the electric load, the problem requires simultaneous recognition of all the items that make up the entire electrical system from the point of entrance of the power-company service or the plant generating powerhouse to the terminals of the utilization devices.

The design of industrial-plant wiring depends primarily upon two basic factors:


Principal elements of an industrial system
1947 N.E.C. References
M
Primary cables — Article 710-230
Potheads — " 710
Outside aerial wiring — " 730
Lightning arresters — " 280-230
Disconnecting switches — " 710-230
Oil circuit breaker — " 710-230
Meters and instruments — " 230
Transformers - instrument — " 450
Transformers - oil insulated indoors and outdoors — " 450
Transformer vaults — " 450
Busbars — " 328
Air circuit breakers — " 240
Outside wiring between buildings — " 730
Switchboards — " 384
Switches — " 380
Fuses — " 240
Feeders and conductors — " 220-310
Capacitors — " 460
Transformers - askarel insulated — " 450
Transformers - dry type — " 450
Circuit grounding — " 250
Rigid metal conduit — " 346
Flexible metal conduit — " 350
Wireways — " 362
Busways — " 364
Safety switches — " 380
Receptacles - heavy duty — " 210-422
Flexible cords — " 400
Portable appliances — " 422
Transformers - low voltage — " 450
Infrared heating appliances — " 422
Machine lighting transformer — " 450
Machine lighting fixture — " 670
Motor disconnects — " 430
Motor controllers — " 430
Motors — " 430
Machine tools — " 610

For continuation see sheet 2

FIG. 13-1a. Industrial-plant distribution system with code references. Sheet 1.

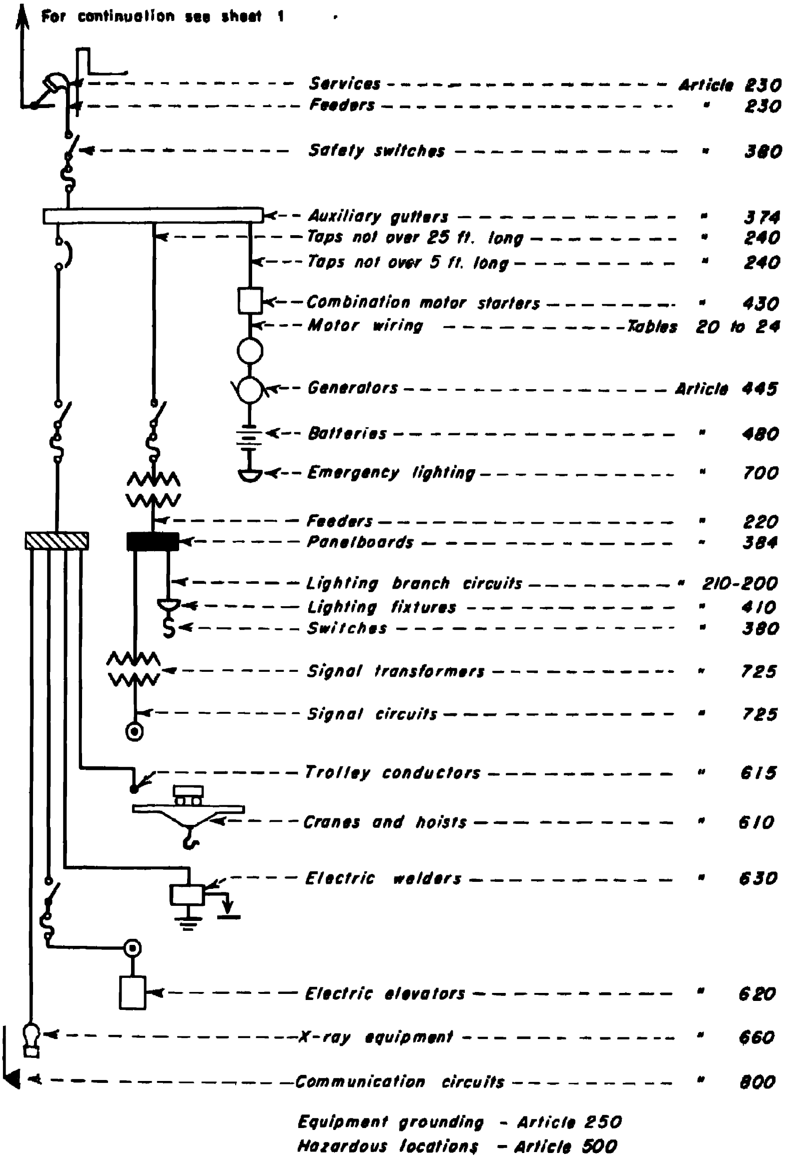

For continuation see sheet 1
Services — Article 230
Feeders — " 230
Safety switches — " 380
Auxiliary gutters — " 374
Taps not over 25 ft. long — " 240
Taps not over 5 ft. long — " 240
Combination motor starters — " 430
Motor wiring — Tables 20 to 24
Generators — Article 445
Batteries — " 480
Emergency lighting — " 700
Feeders — " 220
Panelboards — " 384
Lighting branch circuits — " 210-200
Lighting fixtures — " 410
Switches — " 380
Signal transformers — " 725
Signal circuits — " 725
Trolley conductors — " 615
Cranes and hoists — " 610
Electric welders — " 630
Electric elevators — " 620
X-ray equipment — " 660
Communication circuits — " 800
Equipment grounding - Article 250
Hazardous locations - Article 500

Fig. 13-1*b*. Industrial-plant distribution system with code references. Sheet 2.

1. Cost factor—investment, maintenance, and operating costs. This item is greatly affected by the relation of the electrical work to both the physical plant layout and the process-equipment design.
2. Operating factor—service reliability, safety, voltage regulation, working lines energized, joint use of pole lines for power, telephone, and communication circuits, etc.

The analysis of the problem is complicated by the fact that many kinds of electrical equipment, various plant layouts, and arrangements of process equipment have to be considered. The major electrical items encountered in most types of industrial plants are listed below:

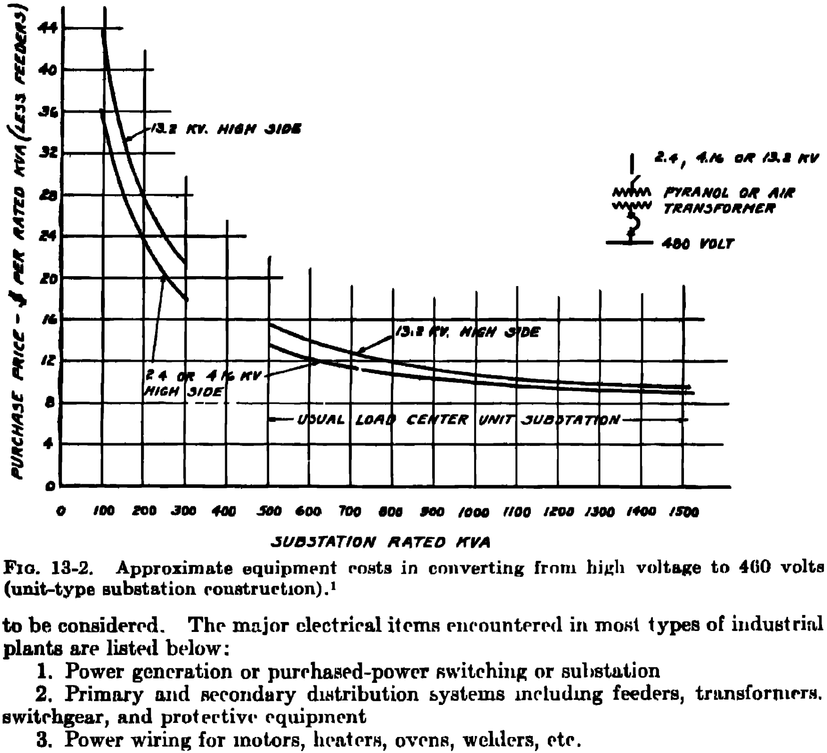


FIG. 13-2. Approximate equipment costs in converting from high voltage to 460 volts (unit-type substation construction).[1]

1. Power generation or purchased-power switching or substation
2. Primary and secondary distribution systems including feeders, transformers, switchgear, and protective equipment
3. Power wiring for motors, heaters, ovens, welders, etc.
4. Lighting equipment, feeders, and branch wiring circuits
5. Electrical and electronic control and instrumentation systems
6. Communication equipment and wiring (public telephone, paging, intercommunication, etc.)
7. Auxiliary systems (fire alarm, electric clock, burglar alarm, etc.)
8. Special items peculiar to process, such as welding, batteries, and rectifiers, electroplating apparatus, elevators and industrial trucks, conveyors, cranes and hoists, ventilation and air conditioning, etc.
9. Yard, roadway, and protective lighting equipment, feeder wiring, and control circuits

In designing electrical distribution systems for industrial plants, major savings can be effected by following certain fundamentals The three major factors influencing the over-all distribution costs are

1. High-voltage lines
2. The substation or substations
3. Low-voltage lines

How these three factors vary for specified conditions is shown in Fig. 13-2, Fig

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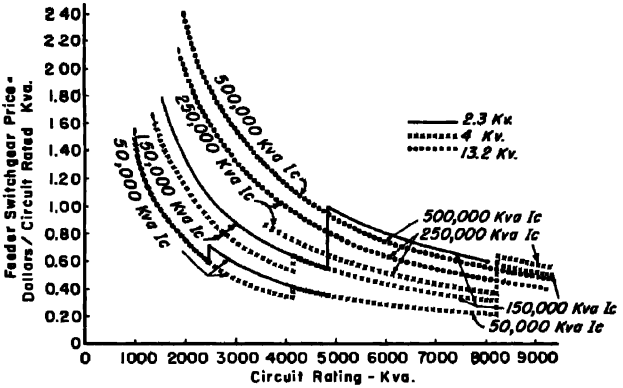
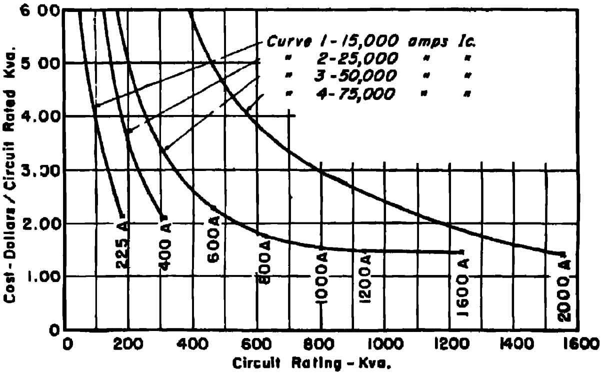
[1] Superior numbers refer to entries listed under References on p. 959.

13-3, and Fig. 13-4. Figure 13-2 shows that, from 500-kva capacity to 1,500-kva capacity, the substation costs are about the same per kva, regardless of individual substation size. Figure 13-3 shows how the feeder switchgear costs are influenced by the circuit kva rating, the required interrupting and momentary current in carrying capacity, and by the operating voltage. This figure presents a summary of feeders—switchgear-equipment device in terms of dollars per circuit-rated kva as a function of circuit-rated kva. In Fig. 13-4 the investment costs for feeders are defined



Metal-Clad Primary Switchgear Feeder Equipment Costs



460 Volt Feeder Switchgear Equipment Costs

Fig. 13-3. Metal-clad switchgear equipment costs.

in terms of dollars per kva per 1,000 ft as a function of unit circuit kva rating at nominal voltages of 460, 2,400, 4,160 and 13,800.

From an examination of these figures the following conclusions may be drawn:

1. Except in the smaller sizes the cost of power-conversion equipment is approximately the same per kva regardless of individual substation size. If this is assumed in any study of costs, the problem becomes much easier, as there remain only two variables—the high-voltage lines and the low-voltage lines. When the problem has been analyzed in terms of two variables, the other variables such as transformer costs, switchgear costs, losses, etc., may be brought in, one by one, to see how each will modify the answer obtained at the first trial.

2. Increasing the substation size increases the length and cost of the low-voltage lines. Decreasing the substation size increases the length and cost of the high-voltage

lines. Starting with a large substation, it is possible to determine quickly whether the high-voltage-line costs increase faster than the low-voltage line decreases. If they increase and decrease the same amount, the determining factor is the substation.

3. If the high-voltage lines are underground, they will increase much faster than overhead low-voltage lines decrease and will definitely favor the larger substation. Likewise, if we assume that the low-voltage lines are underground, we get the opposite result, *viz.*, the low-voltage lines decrease much faster than the high-voltage lines increase. What tends to increase the low-voltage-line cost favors the small substation, and what tends to increase the high-voltage-line cost favors the large substation.

4. If we assume that both high- and low-voltage lines are underground, we will find that the small difference which originally favored the small substation will turn out to be a more substantial saving for the small substation. Because of the greater length of low-voltage lines, the cost to put them underground favors the small substation even though the high-voltage lines are underground.

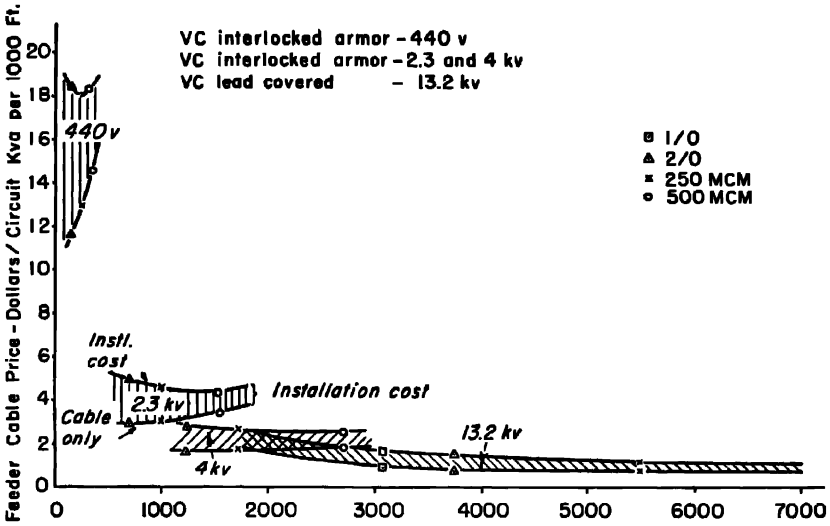


FIG. 13-4. Approximate cost of power-feeder cables.[1]

5. With respect to load density, a high density means shorter lines for both high-voltage and low-voltage lines. The higher the density, the more the substation costs alone control the economics. The lighter the density, the more the lines control the economies, with the substation capacity being larger at higher densities than at lower densities.

From the above, it will be noted that much can be done in controlling the economics of industrial power and its distribution by giving consideration to circuit arrangement. Major savings can be effected by applying the following basic rules to plant layout:

1. Cover as much distance with high voltage and as little with low voltage as possible, consistent with plant layout and process requirements. Distribute power at the highest practical voltage.

The amount of conductor required to transmit 1,000 kva at voltages ranging from 460 to 13,200 volts varies from 100 per cent for 460 volts to 1.8 per cent at 13,200 volts. This difference becomes a more staggering differential when it is stated that it requires one No. 6 copper cable per phase at 13,200 volts and six 250,000-cir mil cables per phase at 460 volts to transmit 1,000 kva over a comparable distance.

By selecting the highest practical primary (2.4 to 13.2 kv) and secondary (460 volts and below) voltages, a large percentage of copper or aluminum required for feeder cables can be saved. A practical example follows:

To carry a 150-kva load three phase at 208 volts requires three 500,000-cir mil copper conductors weighing 4.6 lb per ft; at 230 volts, three 400,000-cir mil copper conductors weighing 3.8 lb per ft; at 460 volts, three 2/0 copper conductors weighing 1.2 lb per ft.

2. Locate the substation as close as practical to the "load center" of the area. Please note the word "practical." There are many installations made to serve equipment in hazardous locations such as carbide generators, paint mixing, solvent vapors, oil and gasoline storage, and explosive dusts. It would be imprudent to locate any standard switchgear equipment within the hazardous area.

However, in systems with the substations located outside the process area, 60 to 75 per cent of the total feeder conductors by weight is required for low-voltage distribution. By locating the substation at the center of the load, the low-voltage feeder lengths are reduced an average of 33 per cent, and the total conductor requirements are reduced about 15 to 40 per cent.

Locating the substation at the center of the load area often means locating it in the working area. Compact, enclosed substation equipment of the air-cooled type or equipment utilizing askeral liquid (noninflammable) may be installed in nonhazardous working areas without vaults. The most practical method is to locate these units on balconies, along building column centers, or in similar areas of minimum value as working space.

3. Use substations of reasonable size. By using, tor example, four small substations rather than one large substation to serve a square load area, one-half the low-voltage feeder conductors can be saved. There is, however, a limit to the desirable reductions in ratings. As the size is reduced, more material is required for high-voltage feeders, and less efficient use is made of the material for constructing smaller substations.

In general, the optimum rating of a three-phase substation with 460-volt secondaries will be from 500 to 1,000 kva; for 208/120- or 230-volt secondaries, 300 to 500 kva.

4. Use cable with high-temperature insulation. The use of high-temperature insulation on wiring for main power and lighting feeders will permit a higher current rating for the same size of conductors. The smaller size wire for a certain current rating also permits the use of smaller conduits. However, in industrial-plant wiring, it is almost always desirable and economical to install services and main feeder conductors of larger sizes than are required for the initial load. If 75°C insulation is installed with initial currents according to the 60°C insulation, advantage may be taken of the difference in rating for any load increase.

5. Use as simple an electrical distribution scheme as possible consistent with justifiable service reliability and required maintenance practices. In general, it is axiomatic that, the more complicated the distribution system, the more feeder material is required to carry a given kilovolt-ampere load. It is often said that extra complication is necessary to give satisfactory service reliability. To accomplish this, we provide means for isolating possible cases of trouble, dual supply circuits, or complicated secondary networks. The processes should be analyzed to develop which building or piece of apparatus is critical and build the system to suit. It is obvious that it is seldom that the whole plant requires an uninterrupted supply of power.

6. Select the proper equipment. Construction costs have risen sharply in the last few years, the greatest increase being in direct field labor. It has been proved that, substations designed piecemeal with individual pieces of equipment, individual motor starters, safety switches, control panels, etc., ultimately cost more than factory-assembled metal-enclosed equipment. The latter results in lower first cost when properly considered with regard to

a. Substantial reduction of initial layout-engineering time

b. Reduced installation time

c. Use of manufacturers' lower priced "repetitive-manufacture" units which reflect direct saving through volume production

Another item that must be considered in considering costs of substations is the use of auxiliary cooling equipment in the form of forced air on medium-size banks and forced oil in combination with forced air on large transformers. A second item that saves material and reduces costs, except in the smaller distribution transformers, is the use of three-phase instead of three single-phase transformers for supplying three-phase loads.

Electrical Distribution Systems

The local distribution system serves the industrial plant in very much the same manner as the combined transmission and distribution systems serve the utility company. Industrial plants offer a big variation in providing a distribution system to distribute the output from the local generating station, from the purchased-power lines, or from these sources, where paralleled, to the various individual loads throughout the plant

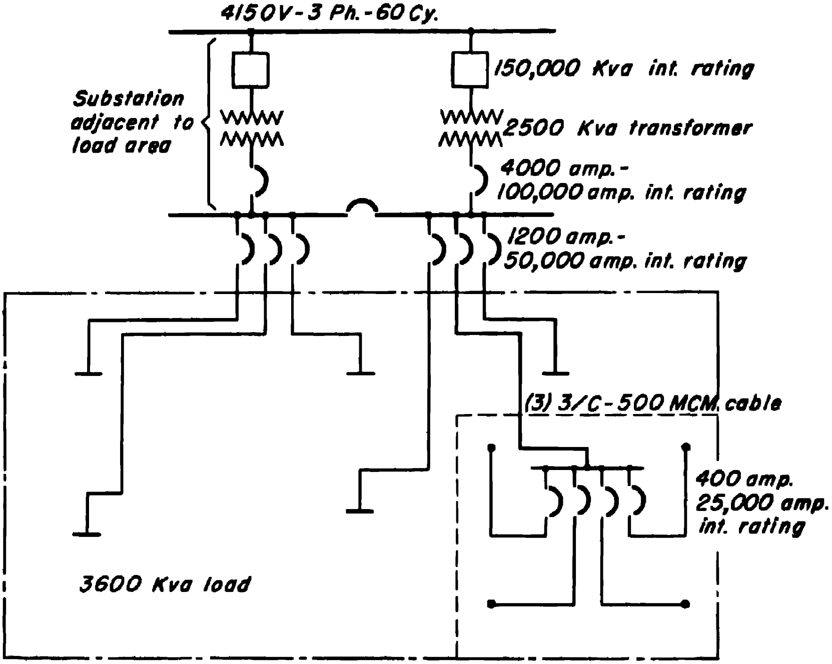


FIG. 13-5. Conventional-type distribution system (multiple low-voltage feeders).

What form of distribution system checks up as best for a plant depends entirely on how much value can be attached to dependability and flexibility in the system. For example, if power is to be used in the manufacture of a product, such as that of an automobile, where the design is changed frequently and extensively, a flexible, easily changed system is indicated. Where continuity of service is essential, such as some chemical processes where a batch could be ruined by a power failure, an extra-dependable system should be used.

Up until a few years ago, the electrical system in most industrial plants consisted of a single large-capacity substation which stepped down the supply voltage to the low-voltage utilization voltage, with 2,300-volt systems being the highest industrial-distribution voltage even for the larger plants Such an arrangement, commonly called the "conventional" type of distribution, is shown in Fig. 13-5. The distribution from the main switchboard involves large low-voltage feeder conductors which are very expensive to install and result in heavy transmission losses. Most plants today use higher voltage distribution systems (4,160, 6,900, 11,500, or 13,200) for supplying transformer banks of adequate size within the load areas "Unit-type substations" are economically employed, particularly where the total plant load is above 500 kva and somewhat uniformly distributed throughout the area The advantages and disadvantages of the "conventional" and "unit-type-substation" distribution systems are given in Table 13-2 The desirable installation and operating features of the unit-substation-type industrial-distribution system are summarized below

Table 13-2. Comparative Performance of Conventional versus Unit-type-substation Distribution Systems

Advantages (+) Disadvantages (−)

Item	Conventional	Unit-type substation
Transformer cost	(+) Less transformer capacity usually is required when a single conventional substation is used Averaging out demand for an entire plant permits taking greater advantage of load factor	(−) Two or more smaller capacity transformers cost more than one large transformer
Secondary breaker costs in small plants (500 kva or less)	No advantage	No advantage
Secondary breaker costs in large plants (over 500 kva)	(−) When conventional single-transformer system is used in large plants high kva rating transformers are required and thus large and expensive secondary breakers are inevitable	(+) Splitting a large load between smaller size transformers permits smaller and less expensive secondary breakers
Cable costs (small area plants)	No advantage	No advantage
Cable costs (large area plants)	(−) Long lengths of low voltage feeders are uneconomical	(+) Short secondary feeders from transformers located at load centers
Efficiency	(−) Higher loss due to longer cables	(+) Fewer losses from shorter secondary feeders
Voltage regulation	(−) Becomes worse the longer the secondary feeders	(+) Good and uniform
Service reliability	(−) Failure of main transformer or fault on main bus affects entire plant load	(+) Failure of any one part does not necessarily mean an interruption of the entire plant load
Interchangeability	(−) Breakers only	(+) Breakers transformers and entire substations can be planned to be interchangeable
Flexibility	(−) Requires entire plant-load interruption when certain changes are in progress	(+) May be designed for high degree of flexibility and permit arrangement changes with minimum service interruption
Expandability	(−) Single transformer limits possible plant additions and load increases	(+) New substation can be readily added when, as and if new load develops

1 They are installed at centers of load demand where cable runs are short

2 Because extensive engineering and planning are not necessary and the equipment is self-contained, little time is required for installation

3 Beginning with small units, substation capacity can be added in small increments as required, saving fixed charges on the unused capacity of larger units or on large substations located away from load centers

4 One cannot always foresee exact factory load conditions, and here the unit substation is a help It can be moved from place to place if changes in load conditions become sufficiently great to require it

5 A number of small unit substations offer a high degree of over-all service reliability Since each serves a relatively small load, outages of any of its elements involve the dropping of a relatively small amount of load

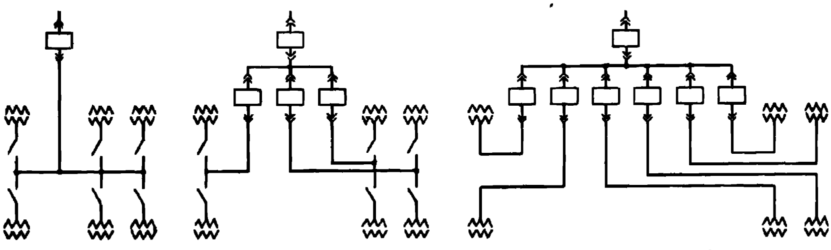
6 Excellent voltage regulation can be obtained by means of the ratio-control equipment on the transformer, ensuring the greatest economy in regulating equipment.

7. The metal enclosure provides a feature particularly desirable in the industrial plant, with the elimination of accidental contact with live parts, the assurance of safety to personnel, and protection to the enclosed equipment.

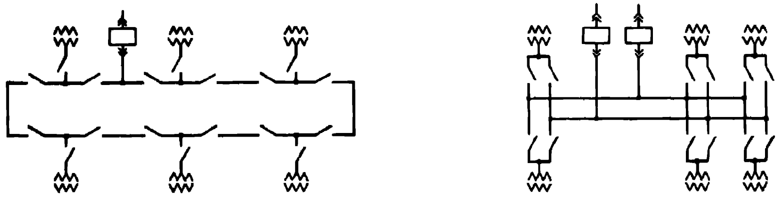
8. Because of standardization of units, additional unit substations can be purchased economically in simple transactions based on definite costs without involved engineering studies, plant layouts, or additional building structures.

There are a large number of plants which are satisfied with a single purchased-power feeder supply, particularly where the feeder is overhead and faults on it can be located and repaired in a relatively short time. The larger plants, or plants where continuity of service is very important, will usually require two or more primary feeders to serve them.

Today, for all plant loads over 500 kva, the unit-type substation construction should be considered and its advantages and disadvantages compared with the conventional-type distribution. Table 13-2 shows such a comparison.



Simple Radial (a) Simple Radial (b) Simple Radial (c)



Single Loop Radial (d) Primary Selective (Double Radial) (e)

FIG. 13-6. Basic primary voltage-system arrangements.

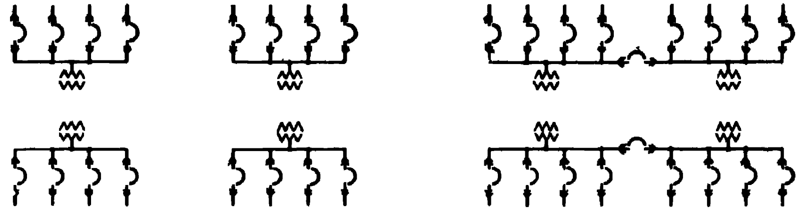
The primary distribution systems in industrial plants have been, in the past, in almost all cases, some form of radial system similar to that shown in Fig. 13-5. Figure 13-6 shows three basic primary voltage-system arrangements, each of which can be used with any type of low-voltage system.

The simple radial system *a* has one primary-feeder breaker for all transformers. This arrangement is inferior to the conventional-type system in service reliability, since a number of the transformers are connected to the primary feeder, and a fault in the feeder or disconnect switch and trouble in any transformer will result in an outage to the area. This outage may be of long duration since service cannot be restored until the trouble is located and repairs are made. As the breaker must carry the full-load current of all transformers simultaneously, short-circuit current of one transformer will not look so big to *the breaker* if the other transformers are highly loaded, and the breaker may be slow in opening.

The simple radial system *b* shows that, by increasing the number of primary feeders and breakers and connecting only two transformers to each feeder, the extent of the outage can be reduced when a primary-feeder fault occurs. While this represents an increase in cable and breaker expense over the simple radial system *a*, better transformer protection and greater system dependability are obtained. For most plants, this would be a wiser investment than system *a*.

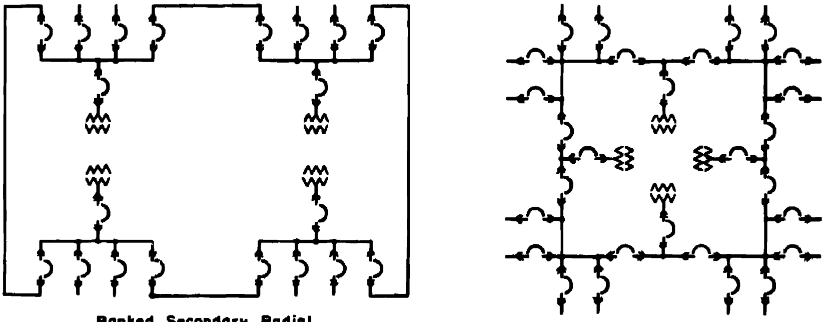
Simple radial system *c* shows an increase in the number of feeders carried to the point where there is only one transformer per primary feeder. This is both the costliest and the most dependable of the three versions of the simple radial high-voltage system. This form of simple radial system will usually be very expensive and can seldom be justified.

The single-loop radial system *d* employs a sectionalized loop controlled by a single primary-feeder breaker. This system requires more primary cable and additional disconnect switches and is only slightly more expensive to install than the simple radial system *a*. The advantages are that (1) transformers and cable difficulties can be deenergized for inspection and repairs without further interruptions to the rest of the system; and (2) regulation and efficiency are improved over the simple radial system.



Conventional Secondary Radial

Secondary Selective Radial



Banked Secondary Radial

Loop Tie Radial

Fig. 13-7. Basic secondary voltage-system arrangements.

The primary selective radial *e* is, in effect, one simple radial system imposed upon another, each having sufficient capacity to carry the essential plant load. The addition of a second primary-feeder breaker and extra primary cable makes this the most reliable and flexible of the three basic high-voltage systems and also the most expensive. Failure of one line means dropping of only half the load, and only for as long as it takes an operator to throw the loads interrupted over to the line still in service. Repairs and changes can be made in one line while the other carries the load. The benefits derived from the reduction in the amount of load dropped when a fault in any section of the primary occurs, plus the quick restoration of the service to all or most of the system, are of considerable importance.

Figure 13-7 shows four basic secondary voltage-system arrangements. Any one of the systems can be used with any one of the high-voltage systems previously considered. These systems are alike in that power is fed to the loads through radial circuits. The chief difference between them lies in the extent to which power is maintained on the low-voltage side of the transformer in case of primary-system failure.

The conventional secondary radial system is the one most commonly used in industrial plants because of its lower cost. While it is essentially a scaled-down, low-voltage version of the conventional primary system, the shortcomings of this system are less serious on the low-voltage side because cable runs are shorter.

However, the conventional secondary radial system has a minimum of dependability. If the transformer fails in any distribution unit, that entire unit must be taken out of service until repair or replacement can be made. To overcome this disadvantage, tie connections can be made between individual units. Tie connections can group the unit into pairs resulting in a secondary selective radial system, or a complete secondary loop can be formed resulting in a banked secondary radial system.

The secondary selective radial system has two load-center units grouped to form a single substation. This secondary system is used mainly with the corresponding primary selective radial system, in which one transformer is fed from one of two lines and the second transformer from the other.

Tie breaker between the units is normally open. The two main breakers and the tie breaker are so interlocked that only two of them can be closed at a time. Each transformer may be large enough to carry either the combined load or only its own load plus emergency load from the other unit. In either case, transformer investment is greater than in previously considered systems. Because the tie breaker is normally operated open, short-circuit currents are no larger than with a conventional radial system.

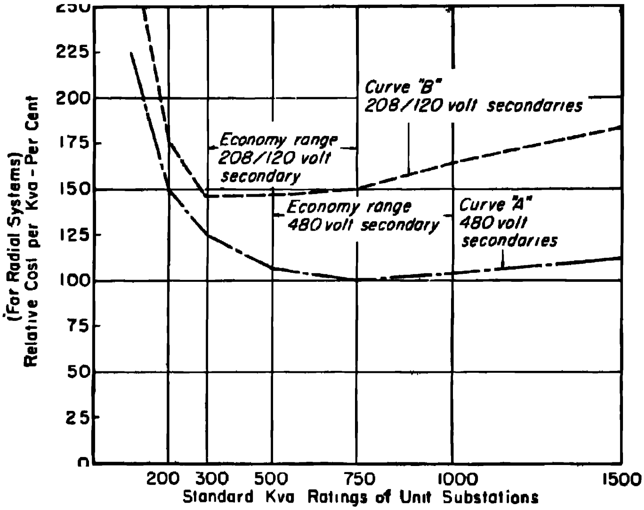


FIG. 13-8. Unit substation sized within "economic range."

Cost of this system is slightly higher than that of the conventional secondary radial system. Dependability is much greater. This system can be used in a modified form to pick up the load supplied from a single transformer substation by extending the tie circuit and connecting it to the substations through tie-line breakers.

The banked secondary radial system is a form of secondary network scheme. Power is supplied to the load centers throughout the plant over two or more primary feeders. Each primary feeder and its associated transformers function as a unit, and the network must be so designed that the entire plant can be carried when one primary feeder and its associated transformers are out of service. This system is not justified for most small plants, but it appears to offer some economic promise where there are two or more primary feeders available for serving the plant and where the load density in the plant and the total plant load are large.

A tie connection may be a solid connection but should have breakers at each end, as shown above. Should one unit be overloaded, power will be fed automatically from the adjacent unit; more efficient transformer loading is obtained.

The tie connection increases the short-circuit currents possible on feeder circuits so that larger feeder breakers may be required.

The so-called "secondary network" for industrial plants is a banked secondary radial system where the transformer secondary breaker has been replaced by a network protector. Flexibility, regulation, and reliability are unchanged, but the cost is increased considerably.

The loaded-tie radial system differs from the banked secondary radial system in the location of the feeder breakers. In the banked secondary system the tie cables must be capable of carrying from 60 to 70 per cent of the current from one transformer bank. On a balanced system little current flows through these tie cables except in emergencies.

Here all the power flows through the tie cables, thus requiring them to be larger, but there is a net saving in copper because the tie breakers are tapped off the tie as near as possible to the load. This shortening of the cable from feeder breaker to load not only saves copper and reduces installation cost but also improves regulation and efficiency. The system has better characteristics in emergency operation than the banked secondary radial system.

Figure 13-8 shows the relative cost of various substation sizes for 460-volt and 208/120-volt secondary distribution with the economical range of capacity noted.

Table 13-3 shows the comparative performance of various groupings of basic primary and secondary voltage-system arrangements.

Table 13-3. Comparative Performance of Distribution Systems

Item, based on 10 va/sq ft and plant building area of 750 ft (length) by 500 ft (width)	Unit-type-substation arrangements					Conventional multiple low-voltage feeders
	Simple radial primary and conventional secondary	Primary selective and conventional secondary	Double simple radial primary and secondary selective radial	Triple simple radial primary and banked secondary radial	Primary selective and banked secondary radial	
Number of substation units	5	5	4	6	5	1
Number of transformer units.	5	5	8	6	5	2
Size of transformer units....	750	750	750	750	750	2,500
Total transformer kva .	3,750	3,750	6,000	4,500	3,750	5,000
Firm load capacity for high-tension feeder outage only	.....	3,750	3,750	3,750	3,750	2,500
Relative cost, per cent	100	125-130	140-145	175-200	145 150	160-170
Voltage regulation, normal ..	C	C	A	B	A	D
Service reliability	D	C	B	A	B	D
Ease of maintenance . ..	A	C	B	C	D	A
Simplicity of operation ..	A	B	B	C	B	A
Safety.	A	C	A	B	C	A

Voltage of the Primary Distribution System

The distribution-system voltage or voltages is one of the first factors to be considered in laying out the distribution system in an industrial plant. The distribution-system voltage, or voltages, is materially affected by the size of the motors or other electrical equipment which are to be operated, the total kva load, the area over which the power is to be distributed, and, to a greater or lesser extent, by the generated or incoming voltage.

With respect to the primary distribution system, if the voltage of the incoming utility lines is 13.2 kv or below (generated power is always within this range), primary distribution at this level will generally be most economical. Higher voltage will be stepped down by a master substation unit before plant primary to local substation units and other high-voltage-utilization equipment.

Satisfactory and convenient distribution must follow standard practice as to frequency, number of phases, and nominal system voltage. Sixty-cycle systems predominate in this country and are superseding twenty-five-cycle systems for practically all applications with the exception of heavy-duty steel-mill work. In general, three-phase supply is most desirable. This leaves the question as to which primary voltage is most economical.

The fact that standard voltages are in steps, *i.e.*, 2,400, 4,160, 6,900, 11,500, or 13,200, indicates that the problem is one of comparing the cost for any one condition with that for the others, rather than attempting to determine a theoretically most economical voltage. A general cost equation may be written for each voltage classification as follows:

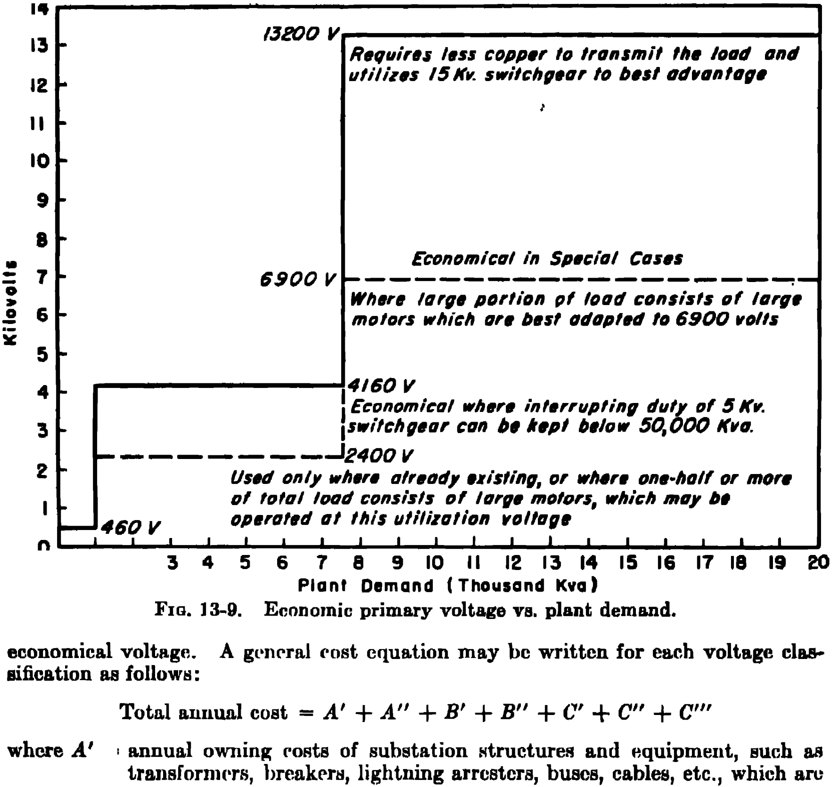


Fig. 13-9. Economic primary voltage vs. plant demand.

$$\text{Total annual cost} = A' + A'' + B' + B'' + C' + C'' + C'''$$

where A' = annual owning costs of substation structures and equipment, such as transformers, breakers, lightning arresters, buses, cables, etc., which are affected by the voltage

A'' = annual costs of energy losses in substation which vary with the voltage

B' = annual owning costs of primary lines in place with consideration given to conductor size, voltage drop, insulators, pole-line construction, etc.

B'' = annual costs of energy losses on primary lines

C' = annual owning cost of distribution transformers in place

C'' = annual cost of core losses on distribution transformers

C''' = annual cost of copper losses on distribution transformers

The above factors show that the choice of voltage depends not only upon the total kva to be supplied but also upon the area over which it is distributed.

A study of the distribution-systems costs for medium-sized plants is summarized in Fig. 13-9. In nearly all cases, the 4,160-volt supply is more economical than the 2,400-volt supply. This is true because the 5-kv switchgear is used more effectively than at 2.4 kv; smaller cables are required at the higher voltage; and motors of less than 200 hp can be operated at 460 volts (stepped down from 4.16 kv) at a lower over-all investment cost per horsepower than directly at 2.4 kv because of the higher starter and wiring costs for the latter. Also, the extra cost of motors larger than 200 hp rated at 4,000 volts instead of 2,300 volts is less than the saving of 4,160-volt over 2,400-volt transformers and distribution. Comparative curves of investment for industrial power systems and motors and controls are shown in Fig. 13-10.

In general, while the higher voltages offer the double advantage of lower line losses and lighter and cheaper cables leading to each substation, experience proves the practicability of choosing a primary voltage of 4,160 for industrial-plant demands up to 7,500 kva and 13.2 kv for larger plant demands. The terminal apparatus is approximately proportional to the voltage. The line costs, however, vary only moderately with the voltage for a given amount of power to be transmitted.

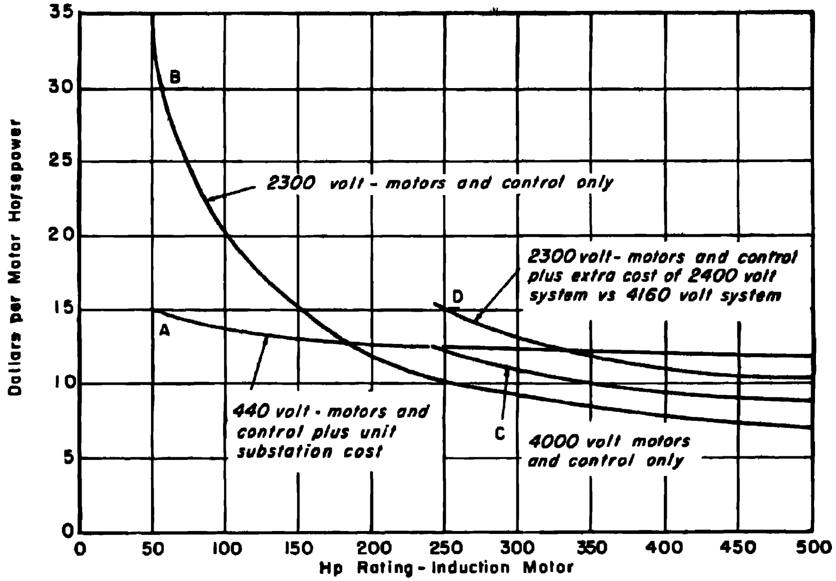


FIG. 13-10. Comparative curves of investment for industrial power systems and motors and controls.[7]

Voltage Best Suited to Utilization Apparatus

Most industrial-plant equipment has a majority of relatively low-rated utilization apparatus, usually not in excess of 25 kw or 25 hp. Economy on feeder and branch wiring, together with controllers and switching, indicates that, in general, the systems shown in Table 13-4 should be used to supply the equipment noted.

For economical location, not only the substation but also the motor control centers, power and lighting panels, and other distribution boards should be located as near as possible to the center of the load. By representing the normal rating of each piece of utilization apparatus as a weight and its distance from a fixed reference or base line as its moment arm, the economical weight of the wiring system can be determined. Locating the load centers as closely as practical to the center of the load results in the most economical layout.

Another of the most important factors to be considered in determining the economical distribution system is the selection of the voltage. The best example of the advantages may be illustrated by considering the factors or constants used to determine the current and wire size for a known load. Figure 13-11 shows the commonly

Table 13-4. Equipment-utilization Voltages

Nominal voltage	Phase	Number of conductors	Equipment
120	1	2	Fractional-horse power motors up to and including ½ hp Heaters, ovens, etc , up to and including 1½ kw Incandescent and fluorescent lighting
208 220	3 and 3	3 3	Motors—¾ to 25 hp—three phase Heaters, ovens, etc , from 2 to 3 kw, single phase at 208 volts or 220 volts Heaters, ovens, etc., from 4 to 50 kw, three phase Fluorescent and mercury lighting Portable power equipment
460	3	3	Motors—¾ to 150 hp Heaters, ovens, etc , from 5 kw up
460/265	3	4	Same as for 460 volts—three phase noted above, plus fluorescent and mercury lighting in industrial-plant areas where the fixtures are mounted at least 8 ft above floor
4,160	3	4	Motors—200 to 4,000 hp (current-limiting fuse protection limit—1,000 hp; larger motors require oil circuit breakers)
6,900	3	3	Motors—above 500 hp
13,800	3	3	Motors—induction, above 800 hp, synchronous, 2,000 hp and larger

used utilization-wiring systems, and Table 13-5 gives a comparison of the essential data which should be considered in determining the best utilization voltage.

The 575-volt power system was formerly used in some sections of this country, but this voltage may now be considered obsolete except for extensions to existing systems. The 575-volt system is slightly more economical than the 460-volt system, but 550-volt utilization equipment is not so readily available as equipment rated at 440 volts. Also, as 600-volt insulated equipment and wiring are used on both systems, the 460-volt system has the greatest safety factor.

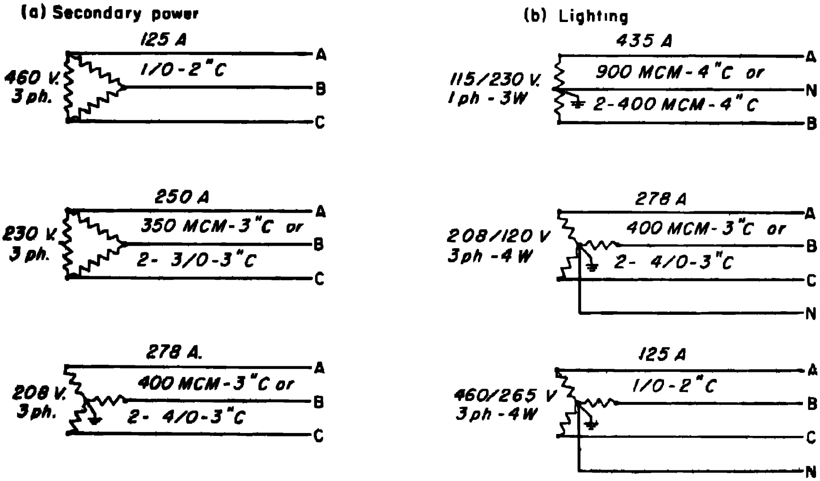


FIG. 13-11. Comparison of wiring systems (100-kva base).

Raising the plant voltage or changing over an existing system cannot usually be justified on a power-savings basis alone. It is a worth-while consideration toward defraying the cost should increased capacity be required. In most cases a change to higher voltage can be accomplished at a lower cost than the installation of heavier feeders should the minimum additional load be 30 to 40 per cent of the existing load.

Table 13-5. Comparison of Various Voltages for Equal Loads

Low-voltage power, three phase

Voltage	460	230	208
Current/kva	1.25	2.50	2.77
Current, per cent	100	200	221.6
Wire size, per cent	100	335	380
Number of wires	3	3	3
Comparative copper, per cent	100	335	380
Comparative cost, 460-volt base	100	230	270
208-volt base	37	88	100

Lighting

Voltage	208/120	115/230	460/265
Phase	Three	Single	Three
Current/kva	2.77	4.35	1.25
Current, per cent	100	156	45
Wire size, per cent	100	225	25
Number of wires	4	3	4
Comparative copper, per cent	100	169	25
Comparative cost, per cent, 208-volt base, 3 φ 4 wire base	100	150	40
115/230-volt base, 1 φ 3 wire base	67	100	27
460/265-volt base, 3 φ 4 wire base	250	375	100

Voltage Variation

Electrical engineers have a very special interest in voltage standardization because all electrical equipment and construction material is designed and manufactured for a definite voltage range. Construction material is rated generally in terms of the maximum voltage for which it should be used. For example, wire insulated for 600 volts may be used on any circuit in which the applied voltage does not exceed 600.

Utilization equipment, such as lamps, motors, electronic tubes, and heating devices, is manufactured for operation on a relatively small variation from the rated voltage of the equipment. Nearly all such equipment is rated at a particular voltage, which must be provided to obtain the designated operating characteristics or to acquire the guaranteed results. However, the manufacturers recognize that there will be voltage variations on the circuits supplying the equipment and so design their products that satisfactory operation will be obtained over a range of voltage which will vary with the equipment. One, two, five, or ten per cent voltage variation may be allowable depending upon the equipment and the use to which it is put.

The light output and life of incandescent lamps vary markedly with the voltage. An increase of 5 per cent (6 volts on a 120-volt system) will increase the light output of an incandescent lamp 18.5 per cent and will decrease the life to about 55 per cent of its rated life. A decrease of 5 per cent in the rated voltage will decrease the light output 16 per cent and will double the rated life.

The characteristics of fluorescent lamps change also with voltage variations but not to such a degree as do the incandescent lamps.

Variations of voltage above and below the design voltage also affect the life of electronic tubes and their operating characteristics at a much higher rate than the fluctuation of the voltage.

The operation characteristics of motors, especially induction motors, change with the voltage. The starting torque of induction motors varies as the square of the voltage. The temperature rise increases with the decrease in rated voltage and decreases with an increase in voltage within certain limits.

For satisfactory operation of industrial electrical apparatus it is important to have and maintain a proper zone of utilization voltage. For further discussion and information on the results of voltage variation upon the operating characteristics of utilization equipment, the reader is referred to AIEE *Miscellaneous Paper* No. 47-157 entitled "Report on Industrial Voltage Requirements" issued in May, 1947.

The electrical engineer designing the wiring system for an industrial plant has no control over the voltage variations of the electricity supply where the supply is received at utilization voltage. Where the electricity supply is received at a voltage higher than the utilization voltage and where transformers are to be installed or where the electricity is generated as a part of the project, the engineer can control the voltage variations by means of voltage regulators and tap changers on the transformers or by voltage regulators on the generators.

From the service switch or from the secondary terminals of the transformers, if they are installed, the fluctuations in voltage *caused by the load* are within the control of the designer of the system. In any circuit, as the load increases, the current and the voltage drop increase in direct proportion. This voltage drop can be controlled within the desired limits by the installation of wires of ample size and by reducing the load on each circuit. For branch circuits, it is desirable generally to limit the wattage rather than to install larger wires. For feeders, ample cross section should be used in the wires to limit the voltage drop in order to obtain the proper voltage at the points of utilization.

The National Electrical Code (Sec. 2202) recommends that the voltage drop in feeders for lighting circuits be limited to 1 per cent and in power circuits to 3 per cent from the service switch, or transformer secondary terminals, to the farthest distribution panel board. Governing authorities may require limitation of voltage drop to other than these figures or may not require in their codes at all the limitation of voltage drop.

The National Electrical Code has limited the current to be carried in each size of wire depending upon the conductor material and the type of installation. These limitations are based upon temperature rise only and have been determined by test. Voltage drop in the circuit does not enter into the determination at all. However, it may be that, for a given current, a No. 14 AWG wire may be used, yet a No. 10 AWG is necessary in order to maintain a low drop in voltage in the circuit

Consider a branch circuit 75 ft in length with a 12-amp load concentrated at the end of the circuit. According to the Code a No. 14 AWG copper wire will satisfy the requirements. The voltage drop in this circuit will be

$$VD = \frac{12 \times 2 \times 75 \times 2.63}{1{,}000} = 4.73 \text{ volts}$$

where 2.63 is the resistance per 1,000 ft of No. 14 wire

A voltage drop of this magnitude is high and generally too high for good operation of electrical equipment. If it is desirable to limit this drop to 2 volts, the wire size should be

$$\text{Wire size} = \frac{11 \times 2 \times 75 \times 12}{2} = 9{,}890 \text{ cir mils}$$

This size is slightly less than a No. 10 AWG copper wire, which is the proper size to use to limit the voltage drop to the desired amount.

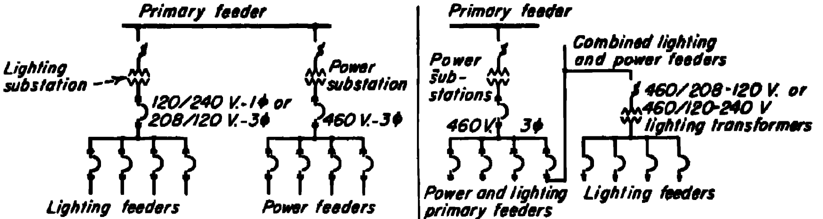
The calculation above is based on the following formula:

$$\text{Circular mils} = \frac{R \times L \times I}{VD}$$

where L = length of wire, in feet
a. For single-phase circuits
For d-c circuits
For each leg of a two-phase four-wire circuit } twice the length of the circuit
b. For three-phase circuits 1.73 times the length of circuit
I = current per conductor
VD = allowable or desired drop in voltage in circuit
R = resistance per mil-foot of conductor: for commercial copper wire, 11; for commercial aluminum wire, 17.25

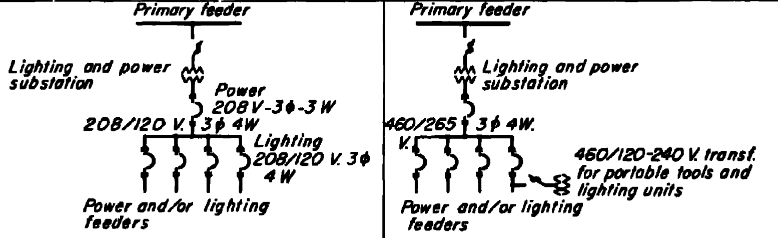
It is recommended that the maximum voltage drop under full-load conditions from the service switch or the distribution transformer (depending upon the installation conditions) to the center load of the farther branch circuit (to the first outlet on the branch circuit if the outlets are reasonably close together) be limited to 3 per cent for lighting circuits and 5 per cent for power circuits. In calculating the voltage drop, the full-load current of lighting circuits should be used and also the summation of the full-load running currents for motors on power circuits. This latter current will be somewhat less than the current required by the Code for the determination of the proper size of wire which includes an additional 25 per cent of the current of the largest motor on the circuit.

If the total drop from service switch to the point of utilization is limited to 3 per cent for lighting circuits and 5 per cent for power circuits and if the drop in the feeders is limited to 1 per cent and 3 per cent as recommended by the National Electrical Code, the voltage drop in branch circuits is then limited to 2 per cent for both types of circuits.


Primary feeder
Lighting substation
120/240 V.-1φ or 208/120 V.-3φ
Power substation
460 V.-3φ
Lighting feeders
Power feeders
Primary feeder
Power sub-stations
460 V. 3φ
Combined lighting and power feeders
460/208-120 V. or 460/120-240 V lighting transformers
Power and lighting primary feeders
Lighting feeders

(a) Separate substations and feeders for lighting and power

(c) Combined power and light substations with air-cooled 1φ or 3φ 460 volt primary lighting transformer


Primary feeder
Lighting and power substation
Power 208 V-3φ-3 W
208/120 V. 3φ 4W
Lighting 208/120 V. 3φ 4 W
Power and/or lighting feeders
Primary feeder
Lighting and power substation
460/265 V. 3φ 4W.
460/120-240 V. transf. for portable tools and lighting units
Power and/or lighting feeders

(b) 208/120 volt substation supplying power and/or lighting feeders.

(d) 460/265 volt substation supplying power and/or lighting feeders

FIG. 13-12. Industrial-plant power and lighting systems.

The figures given above are the maximum drops that should be allowed in any electrical wiring system. It is good practice to limit the drops, especially in the branch circuits, to narrower limits.

Comparison of Methods for Supplying Power and Lighting. Figure 13-12 shows the methods usually used in the new low-voltage distribution systems of industrial plants.

Method *a* shows a layout with separate substations and feeders for lighting and power. This system is usually the highest in first cost and operating cost and requires the largest space.

Method *b* shows a layout with a 208/120-volt substation, or substations, and feeders with power supplied at 208 volts, three phase, three wire, and lighting supplied over a 208/120-volt, three-phase, four-wire system. This system is generally most economical for small compact plants where feeders are not over 100 to 125 ft in length and the motors are small in size.

Method *c* shows a layout of a power substation operating at 460 volts, three phase, with power supplied directly at that voltage and the lighting supplied with 460-volt primary air-cooled transformers with either single-phase or three-phase secondary distribution. The system is particularly advantageous where the power load is three times the size of the lighting load or more. The advantages of this system are (1) the elimination of one complete distribution system and the concentration of substation capacity in a smaller number of larger units and (2) savings due to the reduction of copper in the distribution of lighting load at power voltage because of the greater radius of distribution at the higher voltage.

Method *d* shows a layout with a 460/265-volt substation, or substations, and feeders with power supplied at 460 volts, three phase, three wire, and lighting supplied over a 460/265-volt, three-phase, four-wire system. The 460/265-volt system is limited to light sources of 85-watt fluorescent, 40-watt 5-ft-long fluorescent-lamp units or other types with mogul lamp sockets and the 3,000-watt type-AH or the 400-watt type-H mercury lamps with fixtures mounted at least 8 ft above the floor.

This design will, generally, result in a greater saving than the other schemes in designs where the above units can be used. The mercury lamps have such a high intrinsic brightness that mounting heights under 25 ft are objectionable even for the 400-watt type-H mercury lamp. Also, the four-tube 85-watt fluorescent unit can be used only where mounting heights of 18 to 20 ft are possible. For lower mountings, uniform light intensity cannot be secured except at unusually high intensities.

Another limitation of this system is that extension cords for hand lamps, hand tools, and portable and semiportable equipment are not suitable and permitted by Code requirements to be operated above 150 volts to ground. For this equipment, it will be necessary to install 460/120- or 460/120-240-volt single-phase or 460/208-120-volt three-phase transformers similar to the scheme shown as method *c*.

Distribution of the lighting load is important in planning the distribution system. Regarding light and power supply in industrial plants, the following general conclusions may be stated:

1. Where the lighting load is heavy—comparable to the motor load—a separate distribution system may be worth while for its dependability.

EXAMPLE: An airplane assembly plant where there are a large number of employees, a great amount of handwork, and a large number of small power tools.

2. Where the lighting load is not heavy, the combined light and power systems are less expensive than systems using separate substations for light and power. If it is assumed that, under present-day costs, the total cost of method *a*, including substations, secondary feeders, panel boards, and branch wiring, averages $70 per kva, method *b* will average about $58, method *c* about $47, and method *d* about $36.

3. High-voltage lighting circuits (460/265 volts) are generally most economical where all or a large portion of the lighting is by means of mercury lamps or 85-watt fluorescent lamps.

4. The 208/120-volt system is most economical for incandescent and 40-watt fluorescent lamps. Almost all the incandescent lamps are rated at 120 volts since higher voltage incandescent lamps are fragile and generally not satisfactory. This system will be especially advantageous for combination power and lighting loads where power load consists of small three-phase motors (25 hp and less).

Service Distribution Methods

There are two general methods of running primary and some of the secondary feeders in industrial plants—overhead and underground.

If run overhead, the cables may be (1) bare or weatherproof installed on pole lines or towers with service feeder spans bracketed to the buildings; (2) self-supporting preassembled insulated cables, aerial type; and (3) catenary suspended conduit, weatherproof wireways, or busways carried between buildings.

If run underground, the feeders may be (1) armored or synthetic-covered cable placed in trench and trench refilled, (2) underground raceways with manholes at suitable locations, and (3) exposed cables in tunnels or enclosed in raceways.

The advantages of the overhead type of distribution are

1. Cheapest initial installation
2. Cost of repairs and changes lower for the same number of repairs or changes
3. Repairs and changes easily made

The disadvantages of the overhead type of distribution are

1. Crippling of service due to severe weather conditions; damages by sleet or windstorms and lightning may result in heavy maintenance charges.
2. Poles and wires may be an impediment in traffic.
3. Poor appearance
4. May present danger to life

The advantages of the underground type of distribution are

1. Not affected by severe weather conditions and lightning disturbances
2. No impediment to traffic after installation

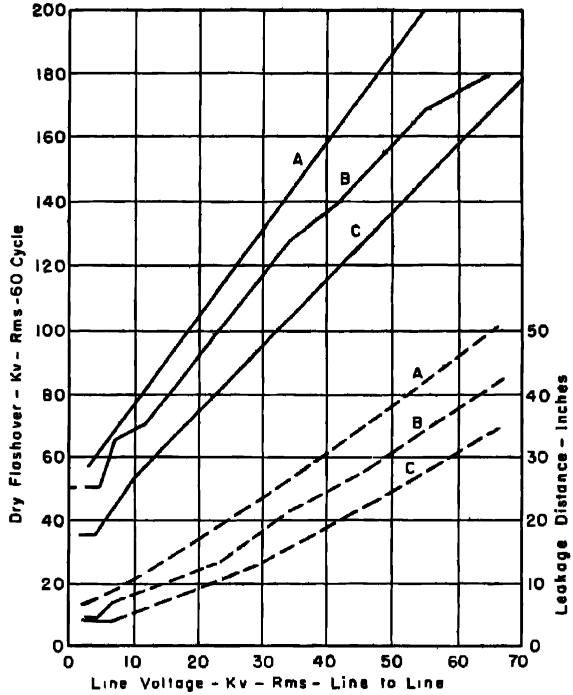


FIG. 13-13. Pin-type insulator. Dry 60-cycle flashover and leakage distance vs. service voltage. Three-phase systems.[3] Curve *A*—largest insulator commonly used on service voltage shown unless special conditions are involved. Curve *B*—approximate average-usage insulators. Curve *C*—minimum insulator levels.

3. No wires or cables visible to produce unsightly appearance
4. No danger to life except to maintenance personnel under certain conditions

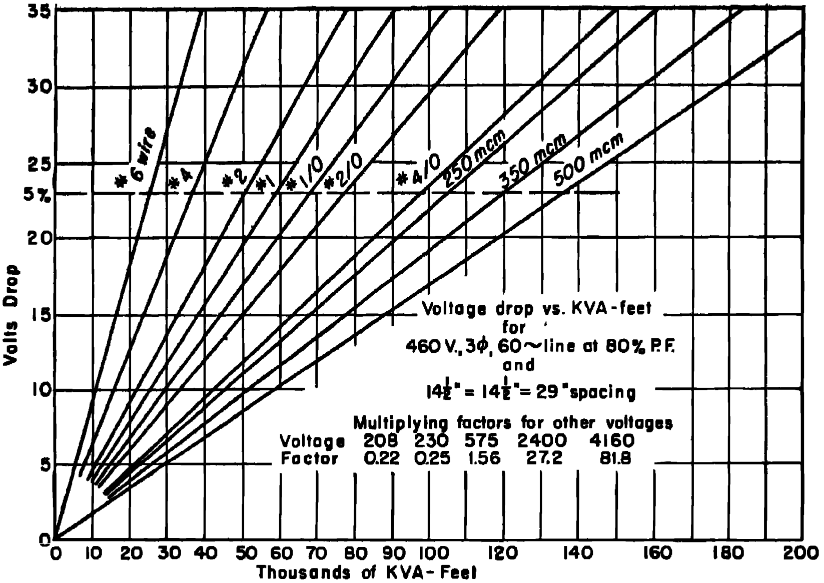
The disadvantages of the underground type of distribution are

1. High first cost (three to five times the cost of overhead construction)
2. Inconvenient to make repairs or changes
3. High cost of making repairs or changes

Recommended Applications. Overhead distribution is recommended for long distances (above 1 mile) and also for short distances with the number of circuits is small (four or less) unless underground is economically justified according to the advantages listed above.

Underground distribution is recommended for short distances, congested plant areas, and where maximum reliability is economically justified.

Overhead construction is also recommended for plants which are operating on a temporary basis and which are more or less continually in a state of change. In

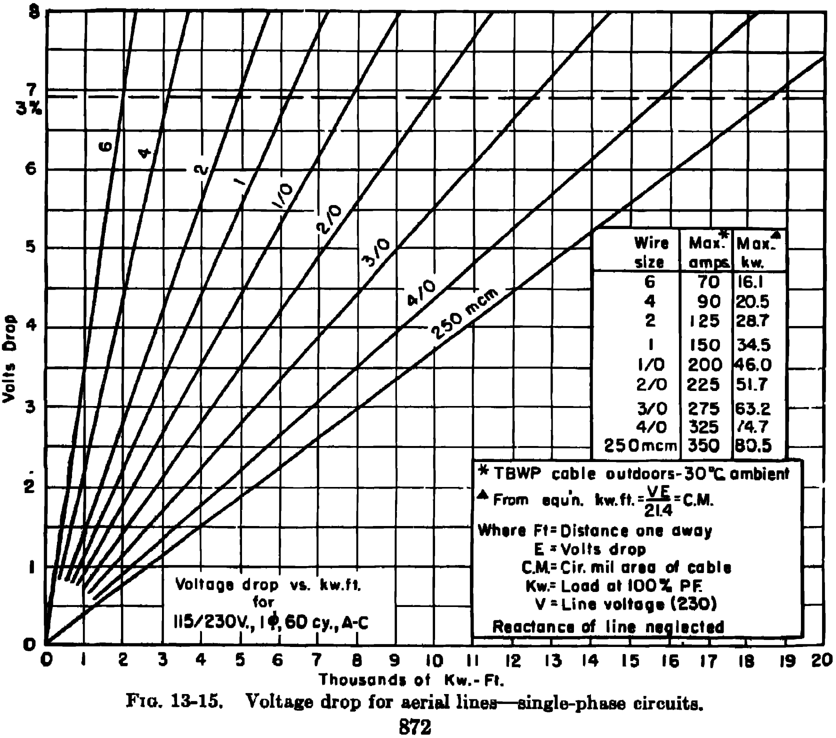


Voltage drop vs. KVA-feet
for
460 V., 3φ, 60~line at 80% P.F.
and
14½" = 14½" = 29" spacing

Multiplying factors for other voltages

Voltage	208	230	575	2400	4160
Factor	0.22	0.25	1.56	27.2	81.8

FIG. 13-14. Voltage drop for aerial lines – three-phase circuits.



Wire size	Max.* amps.	Max.▲ kw.
6	70	16.1
4	90	20.5
2	125	28.7
1	150	34.5
1/0	200	46.0
2/0	225	51.7
3/0	275	63.2
4/0	325	74.7
250mcm	350	80.5

* TBWP cable outdoors-30°C ambient

▲ From equ'n. $\text{kw.ft.} = \frac{VE}{21.4} = \text{C.M.}$

Where Ft = Distance one away
E = Volts drop
C.M. = Cir. mil area of cable
Kw. = Load at 100% PF.
V = Line voltage (230)

Reactance of line neglected

Voltage drop vs. kw.ft.
for
115/230V., 1φ, 60 cy., A-C

FIG. 13-15. Voltage drop for aerial lines—single-phase circuits.

general, an investment for these types of plant would be expected to pay for itself in a relatively few years.

In the more stable industries like textile plants which expect to operate in a similar manner for a number of years, higher construction costs can usually be justified.

Aerial cable of the preassembled type is now being used for power distribution in many plants where the areas are congested and where neat appearance is required. This type of installation provides increased reliability and requires much less time to install than open-type wiring construction. The net cost of installation is fairly comparable with the unsightly open-wire lines. The safety and reliability of nonmetallic cables have been increased by improved methods of shielding, including proper methods of bonding and grounding the shields.

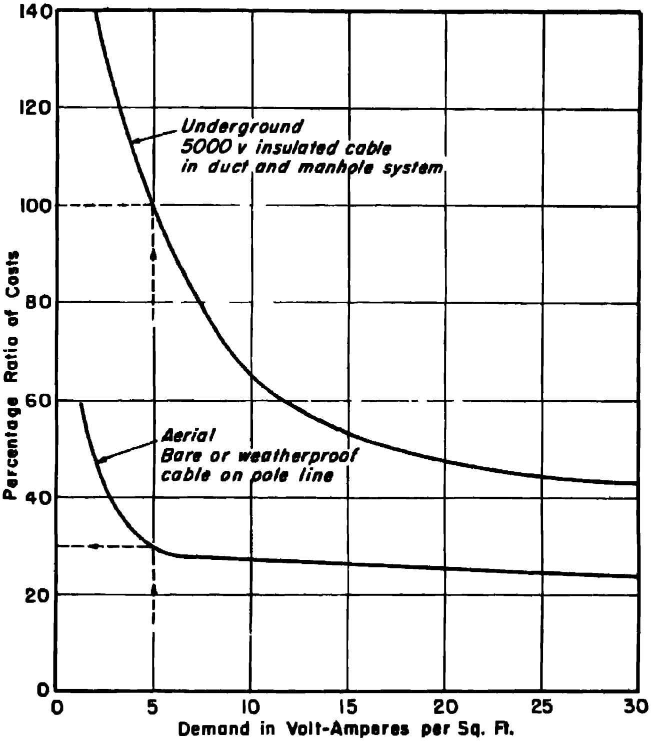


FIG. 13-16. Ratios of comparative installation cost of underground vs. aerial lines (based on a 5-va load per sq ft, equal to 100 per cent for underground lines).

With respect to aerial lines, there is a question regarding the almost universal practice of using weatherproof wire and cable for line conductors. Except in extremely corrosive atmospheric conditions there seems to be little justification in using weatherproof cable. This type of wire is not made to any voltage specification, and its insulating value is doubtful. Much weatherproof wire is soft drawn, so its strength is less than hard-drawn wire. The larger diameter of covered wire makes it more vulnerable to wind and sleet. It will burn down quicker than bare wire once an arc has started owing to the concentration of the arc at the point of covering failure. Finally, its cost installed is higher than bare hard-drawn wire.

Figure 13-13 shows a graph of the characteristics of line insulators, and Figs. 13-14 and 13-15 are graphs of the voltage drop for industrial-plant aerial lines.

The relative cost of plant distribution systems installed underground vs. those run aerially depends upon several factors—the principal ones being

1. The density of the load to be served
2. The type of electrical system used to give the quality of service required

Other factors are probable load increases; locations of loads, aesthetic reasons such as appearance of plant; safety to life and property; cost of operating and maintenance; space requirements for lines and equipment; and others.

Figure 13-16 shows comparative curves of the initial installation costs of underground vs. aerial methods of distribution for a plant consisting of several buildings spaced somewhat uniformly within the plant area having various load densities per unit area. The initial installation cost of an underground system is always higher at all load densities than for the aerial system, but the annual over-all carrying charges are another matter

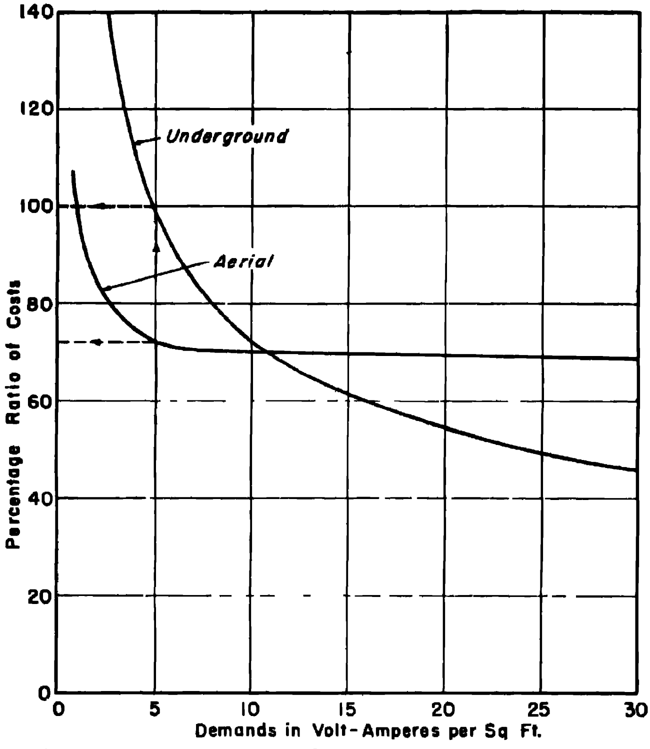


FIG. 13-17. Ratios of comparative annual owning cost of underground vs. aerial lines (based on same conditions as noted under Fig 13-16 plus a 30-year owning period including maintenance and replacement cost of aerial lines).

Wooden pole lines have a life expectancy of 15 to 25 years. Replacement costs would be nearly twice the cost of the original installation as extra labor is required to replace poles and wire spans a few at a time. Electrical losses are also less with underground construction. Figure 13-17 shows comparative curves of the annual owning cost of underground vs aerial methods taking into account depreciation, taxes, sinking fund, maintenance charges, and electrical losses. This curve indicates that underground distribution is justified where the demand is sufficiently high and the loads supplied are relatively stable for a long period of time.

Characteristics of Different Cables for Primary-feeder Installations

Of necessity cables used for underground distribution in industrial plants fall into two general categories—(1) metal-covered, meaning lead-covered, and (2) nonmetallic. Considering first lead-covered cables, the dielectric most frequently used is impregnated paper, chiefly because it is suitable for practically all voltages and because it is reasonable in price. Some varnished cambric has been used, and considerable rubber likewise has been used under the lead, the latter particularly for secondaries.

The object in placing a lead sheath over the dielectric is to prevent moisture, acids, and alkalies from reaching the dielectric and causing failure as those particular dielectrics were never intended to be proof against moisture, acids, or alkalies. Following this line of reasoning a lead-covered cable may be said to be only as good as its sheath and that, should the efficacy of the sheath be destroyed either because of mechanical damage, imperfections in manufacture, or chemical reactions, the efficacy of the cable itself is destroyed.

Nonmetallic cables are confined pretty well to rubber compounds and synthetics. Because of the nature of rubber, it is necessary to make a subdivision and differentiate between primaries and secondaries.

For primaries it is necessary to take into consideration and provide for corona. Some industrial-plant primaries operate at 11.5 and 13 2 kv. In such cases it is necessary to provide an electrostatic shield properly grounded. At 5 kv, through the use of some types of synthetic jacket, shielding is not necessary. In all cases utmost care must be exercised in the correct choice of dielectric. Oxygen is one of the greatest deteriorators to a rubber compound, and a perfect bonding between the dielectric and the synthetic, thus eliminating the oxygen, makes the combination of a good dielectric and the synthetics ideal from the standpoint of longevity. It is essential to eliminate corona and hence ozone in manholes because, even if the primaries have been designed to withstand it, there have been cases known where the ozone generated by the primaries attacked and caused failure of the secondaries in the same manholes.

Primaries with synthetics are much to be preferred to the fibrous covered or braided types. As for the secondaries, voltage being nominal, the characteristics which should be sought are a 75°C-temperature compound, ability to resist normal acid and alkali solutions and residual grease or oil in ducts, together with high mechanical strength, in the order of 3,600 psi.

With every engineering problem the economic problem exists. What is logical for one industrial plant is not necessarily feasible for another. For example, trained lead-cable splicers may reside and be available in one locality, while in others those specialists might have to be imported.

Before weighing any relative merits of one type versus another, consideration should be given to the following characteristics:

1. A lead sheath is ideal for excluding moisture or oils from the dielectric.
2. A 60 per cent jacket is good for moisture, not good in the presence of oil, yet is noninductive and unaffected by electrolysis.
3. At 11.5 or 13.2 kv a compound must have high dielectric strength and ability to resist corona.
4. For 600 volts sufficient electrical strength and a high degree of mechanical strength and ability to age well are desirable

The whole question being, therefore, one of compromise, all the known operating conditions must be set up on one side of the ledger, and on the other side the exact prescription must be written which will fit those particular conditions.

An inventory of the advantages and disadvantages, the opportunities and limitations, of the various types reveals the following:

Leaded Cables. *Advantages.* (1) Under normal conditions, a lead sheath offers the finest and most permanent protection to a cable. Aside from alkalies and organic acids, it is very resistant to corrosion. (2) High-temperature operation is permissible. (3) The lead sheath provides a good grounding means.

Disadvantages. (1) The lead sheath should not be used where electrolysis is present either chemically or in the form of stray currents. (2) It is highly susceptible to failure under vibratory conditions. (3) It requires potheading at all voltages and wiped sleeve joints. (4) Inductive and sheath losses are prevalent in some cases. (5) The lead sheath should not be used on high pole risers because of hydrostatic head and migration of oil.

Nonmetallic Cables. *Advantages.* (1) An advantage of nonmetallic cables is their lower first cost. They are easier to handle and install, being lighter in weight, and are constructed with utmost simplicity. (2) They are not so susceptible to mechanical damage as lead and (3) do not require wiped joints or potheading at 5 kv or under, thereby reducing installed costs. (4) Where pH content of residual water is high, this cable will be unaffected.

Disadvantages. (1) Rubber should not be used in the presence of oils or be subjected to direct rays unless protected by a suitable synthetic covering. (2) Operating temperature of the copper is limited to 75°C for insulations to 8 kv and to 70°C above, although this in turn is compensated by better thermal characteristics and no sheath losses. (3) The chief disadvantage of nonmetallic cables generally is the fact that there is no minimum standard of quality or design, with the result that in the search for economy there have been cases of improper designs being used.

It is impossible to give a specific recommendation general enough in character to cover all cases. In conclusion, one observation that should be emphasized is that no plant engineer in his search for reduced costs should jeopardize the permanency and reliability of a system by any compromise with inherent quality of materials used. Special conditions in any given installation may, of course, introduce factors that could effect a change in the selection, such as corrosion conditions, skill of splicers, load requirements, etc.

The most difficult handicap to overcome in the layout of aerial and underground lines and the reduction of hazards involved is the space factor.

In industrial plants it appears desirable to provide utility alleys for outside service lines and sectionalize these alleys for electric lines, communication circuits, yard lighting, underground fire mains, domestic and process sewer lines, overhead steam and process pipe lines, etc. Through this method it is possible to prevent conflicts and compromises between the various trades that are making use of the utility alleys. These alleys are usually located along the plant roads.

The same idea can be advantageously used in the process buildings in the form of utility shafts or areas reserved for service facilities.

The National Bureau of Standards Handbook H32 contains basic safety rules dealing with the installation and maintenance of electric supply and communication lines. Any designer or construction engineer dealing with the design and construction of both aerial and underground lines should be thoroughly familiar with these rules.

Interior Wiring Systems

A carefully planned lighting and power installation requires a well-planned wiring system. The types of loads served by the distribution system are usually very diversified, and it is not uncommon to see the full range of switch or breaker sizes installed on the same board.

Naturally the voltage should be the highest at which the major part of the power equipment can operate. This is usually the nominal 460-volt, three-phase system.

In selecting the interior feeder wiring system in industrial plants, many factors must be considered. Among these factors are the floor area, height and type of building, the type of occupancy, probable future changes in load requirements, the distribution of load, whether generally uniformly distributed or whether there are large concentrations of load, whether hazards are present such as moisture, corrosive or explosive fumes, vapors and dusts, and many others. The actual construction may be by means of one or a combination of the following methods. The important thing to note is that practically all the methods have two things in common—steel and copper. Steel comprises the raceways, wiring enclosures, outlet boxes, junction boxes, cabinets, and various fittings. Copper constitutes the conductors, switch devices, overload protective devices, and various equipments for current utilization.

1. Open work on insulators
2. Interlocked armor cables installed exposed on racks, walls, or ceiling
3. Enclosed wiring in conduit or tubing which may be exposed or concealed in the building structure
4. Metal wireways
5. Underfloor raceways (steel and fiber)
6. Busways or guarded busbar racks
7. Concentric tubular or polarized tubular conductors as recently developed for heavy welding loads

Branch circuits may be installed by one or a combination of the following methods.

1. Open work on insulators
2. Rigid or flexible raceways
3. Armored metallic or nonmetallic cables
4. Wireways and busways
5. Plug-in bus systems
6. Trolley bus systems
7. Underfloor raceways (steel and fiber)
8. Cellular-steel floors (type of Q flooring)

The open-type cable or bus wiring is acceptable under certain conditions, but other types of wiring methods are so practical for protection against mechanical injury and moisture, acid and alkali fumes and vapors, and safety considerations, that very little, if any, open-type construction should be used in industrial plants, except where the conductors can be placed out of the way, such as along girders and tunnels. Consideration must be given to voltage regulation as the greater conductor spacing required for open-type construction results in increased impedance.

The National Electrical Code, now in its 1947 edition and in its fifty-third year of use, has become the recognized minimum standard on all matters relating to electrical installations and electrical-wiring methods and equipment It was for the purpose of preventing loss of life or injury to person and loss by fire starting from electrical systems that the present Code was developed. All the above-listed methods of wiring are covered in sections of the Code. The designer should keep in mind that the installation must be made as safe and foolproof as practicable.

Table 13-6. Comparison of Interior Wiring Systems

System	Installed cost	Protection against mechanical injury	Protection against moisture	Fire risk	Appearance	Installation during building construction	Ease of installing in finished building	Ease of changing or adding loads
Open work on insulators	1	4	—	4	3	No	1	2
Armored metallic	2	2	—	3	2	No	1	3
Flexible conduit	3	2	3	3	2	No	1	3
Metallic tubing	4	2	2	2	1	Yes	3	4
Rigid conduit	5	1	1	1	1	Yes	4	5
Busways	6	3	—	2	1	No	2	2
Plug-in bus system	7	3	—	2	1	No	2	1

"1" represents the best or in case of costs the least expensive, "2," next best, etc

— indicates the system should not be used

Underfloor duct and cellular-steel floors are not included in this comparison as they are usually used in industrial plants to serve special conditions

Many of the wiring systems noted above cannot be used in certain types of occupancy. For example, rigid metal conduit with threaded explosion-proof joints, approved flexible couplings, and explosion-proof boxes and fittings are required for all Class I, Division 1 hazardous locations; either rigid metal conduit or electrical metallic tubing are approved for both Classes I and II, Division 2 hazardous locations. In

Class II, Division 1 locations, the wiring shall be in rigid metal conduit with threaded boxes and fittings. In wet and damp locations, the wiring raceway systems, including all boxes and fittings, shall be made watertight. Conduits, electrical metallic tubing, flexible metal conduit including all boxes and fittings used therewith should be of corrosion-resistant material suitable for the conditions. In addition, the conduit and equipment should be mounted so there is at least ½ in. air space between the conduit and equipment and the wall or other supporting surface.

Table 13-7. Comparison of Conduit Work

	Percentage
Conduits concealed on concrete forms or installed in masonry walls	100
Conduits run exposed on racks, comprising groups of four or more not over 10 ft above floor*	110
Conduits run exposed on walls or ceilings not over 10 ft above floor*	120
Conduits fitted to machines, benches, etc.	140

* For 11- to 15-ft ceiling run add 10 per cent; for 16- to 20-ft ceiling run add 30 per cent; above 20 ft add 3 per cent additional for each foot.

The rigid metal conduit is used to a great extent in both industrial plants and office buildings as it provides maximum protection for the conductors and a dependable protective grounding path for the apparatus. The conduit can be installed either exposed or concealed and is universally used under all atmospheric conditions. However, Everdur or aluminum conduits are used for certain acid or alkali corrosive conditions.

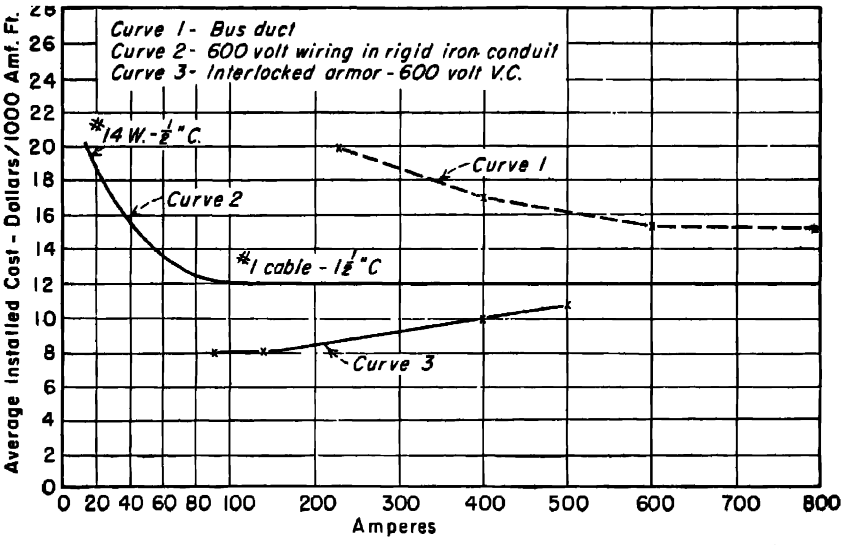


FIG. 13-18. Comparative costs of various types of feeder systems; 600 volts and lower, three-phase, three-wire.

A trolley bus system is a special form of busway constructed so it can receive both stationary and trolley-type devices. Portable tools or other loads are connected with brushes or contact rollers which make contact with the bus bars. Trolley busways are also being used for lighting distribution where it is desirable to provide readily facilities for rearrangement of the lighting fixtures.

The busway provides a safe, flexible, easy-to-erect installation with nearly 100 per cent salvage value. It is possible to provide a current range up to 4,000 amp. The maximum voltage rating is 600 volts, but at that this represents a lot of horsepower, kilowatts, and lumens. This type of system is particularly advantageous for serving groups of small- and medium-size motor-driven machines such as drills, lathes, shapers, etc. It also provides flexibility for future power applications and is particularly adaptable to high-current loads of 400 amp and larger. The installed unit cost per ampere-feet is about 50 per cent higher than for equivalent cable and conduit installations.

Table 13-8. Duct-line Reactance Drops

(Three 600-volt type R and T wires in conduit)

Size, awg	Average conductor spacing, center to center*	Reactance at this spacing for 1,000 ft†	Resistance of commercially pure copper for 1,000 ft‡	Effective impedance at various power factors, volts difference/amp/1,000 ft, three phase			
				1.0 power factor	0.9 power factor	0.8 power factor	0.7 power factor
10 solid	0.380	0.0550	1.000	1.732	1.619	1.445	1.341
8 solid	0.483	0.0535	0.627	1.086	1.024	0.940	0.850
6 strand	0.633	0.0525	0.398	0.689	0.664	0.612	0.556
4	0.632	0.0460	0.250	0.433	0.424	0.396	0.363
2	0.736	0.0427	0.157	0.272	0.276	0.263	0.244
1/0	0.941	0.0427	0.100	0.1732	0.1867	0.1830	0.1739
2/0	0.946	0.0412	0.0794	0.1375	0.1544	0.1529	0.1471
4/0	1.127	0.0396	0.0500	0.0866	0.1070	0.1104	0.1093
350M	1.399	0.0378	0.0306SE	0.0530	0.0755	0.0817	0.0836
			0.0303	0.0525	0.0750	0.0813	0.0832
500M	1.410	0.0347	0.0217SE	0.0385	0.0601	0.0669	0.0699
			0.0212	0.0369	0.0584	0.0654	0.0682
750M	1.631	0.0327	0.0149SE	0.0258	0.0471	0.0547	0.0582
			0.0141	0.0244	0.0459	0.0535	0.0572
1,000M	1.875	0.0370	0.0116SE	0.0211	0.0450	0.0521	0.0535
1,000M			0.0106	0.0184	0.0445	0.0500	0.0526

* The spacing between wires is assumed to be the average between the minimum possible spacing, three wires in contact, and the maximum, which is an equilateral configuration with each wire in contact with the interior surface of the conduit. Based on appropriate sizes of conduit as specified in National Electrical Code.

† The single-phase reactance is figured for the average conductor spacing.

‡ SE is the skin effect included in resistance figure. Resistance at 20°C. Three-phase impedance figures for 1.0 power factor include only the resistance component of drop, the reactance component being so small it is negligible. Since the column headed 1.0 power factor is resistance drop only, a comparison with values for other power factors shows the error occurring if the reactive component is ignored, *i.e.*, at 0.7 power factor, for example, the drop is 23 per cent less for No. 10 wire and 125 per cent more for 750,000-cir mils cable. The divergence from figures based on resistance drop only is entirely too great for even approximate results.

Example: What is the drop in 400 ft of No. 4 wire in conduit feeding a 40-hp motor taking 48 amp at 440 volts full load, three phase, 90 per cent power factor?

$$E = IZ$$

where E = volts drop, I = amperes per phase, and Z = impedance per 1,000 ft

$$E = 48 \times 0.424 \times (400/1{,}000) = 8.13 \text{ volts drop}$$

For single-phase or two-phase load, multiply impedance by (1/0.866) = 1.155.

Example: What is the drop in 200 ft of No. 8 wire in conduit supplying a 29-amp load at 220 volts, single phase, 70 per cent power factor?

$$E = 29 \times 0.85 \times (1/0.866) \times (200/1{,}000) = 5.7 \text{ volts drop}$$

Wireways may be advantageously used where it is desirable to provide a flexible and accessible means to install quickly additional circuits and to move machines from one location to another. With its allowance of 20 per cent of area or 30 conductors maximum it has a great many possibilities. The ground continuity of this system of wiring is rather doubtful but is easily remedied by installing a bare grounding conductor looped to grounding bushings on each conduit or bolted to each section of the wireway. Wireways may not only be used advantageously to distribute power loads to groups of small motors but also to facilitate the distribution of lighting branch circuits and to eliminate the large groups of conduits necessary to route them to their various centers.

Varnished-cambric-insulated, interlocked armor cable may be advantageously used in industrial plants where there are dry installation conditions. It is a low-cost general-purpose power wiring cable rated at either 600 or 5,000 volts. It does not require the use of conduit as the positively interlocked armor provides proper mechanical protection for the cable. It can be installed in racks or straps on the ceiling or along the walls. As compared with the usual rigid conduit system, this cable has approximately the same material costs, but there is a great saving in labor costs (see Fig. 13-18).

The tasks that electricity is being called on to perform in all types of occupancies are expanding rapidly. No longer is it used solely as a source of light and power. Provisions must also frequently be made for telephone and telegraph system, signaling and other communication systems, such as fire alarm, sprinkler alarm, watchman circuits, burglar alarm, etc. In interiors where there are few or no permanently located dividing partitions, underfloor raceways may be advantageously used. The number and location of raceways comprising the system depend upon service requirements. Underfloor raceway systems often fall short of their designed function owing to insufficient home-run conduits or connections between the feeder cabinets and the underfloor system for the requirements of wiring for signal and power circuits.

Power distribution in large areas to groups of small machines is greatly simplified when cellular-steel floors are used. The cells which are on 6-in centers provide a series of continuous raceways the full width of the building and are ideally adapted for multipurpose electrical distribution circuits. The steel cells of the floor are crossed over by headers carrying wires for telephone, power, and other types of electrical service.

Equipment and Wiring for Hazardous Locations. Chemical processes are now employed to some extent in almost every industry. Many chemicals create fumes that when mixed with air are highly explosive and/or dust-laden atmospheres that are dangerously flammable. Explosion hazards are classified in the National Electrical Code into three groups (Art. 500, 1947 edition) as follows:

Class I hazards embrace air-fume contamination separated in two divisions, *viz*, (1) locations where flammable gases or vapors may exist under normal operating conditions, frequent repair or maintenance operations, and where breakdown or faulty operation of process equipment might also cause simultaneous failure of electrical equipment; (2) locations where flammable gases, vapors, or volatile liquids are handled in a "closed system" (confined within suitable enclosures), or where hazardous concentrations are normally prevented by positive mechanical ventilation, or areas adjacent to the locations noted in (1) above where the gases or vapors might occasionally be communicated.

Class II hazards cover combustible dust-laden atmospheres. Again this class is separated in two divisions similar to those noted above for Class I.

Class III hazards include easily ignitible fibers or flyings. Division 1 covers manufacturing-process areas where ignitible fibers or materials produce combustible flying. Division 2 covers storage areas for such fibers and materials.

Considerable skill and judgment are required to determine the class of hazard and the extent of the hazardous area in the plant. Many marginal cases will be found due to the nature and relative concentration of the vapors, gases, dust, or other highly combustible matter. Installation conditions for hazardous areas are considerably more expensive than for standard work. In general, higher costs arise when installing conduit, conduit fittings, motors, controllers, lighting panels and equipment, and wiring. Motors weigh more; fittings are heavier and clumsier; and, in Class I wiring, varying quantities of sealing fittings must be installed and filled with compound.

Refer to Art. 500, 1947 edition of the National Electrical Code, for rules and regulations applying to the installation of electrical wiring and equipment in classified locations. Figure 13-19 is a diagram illustrating the requirements for Class I installations.

No motors or control equipment are available at the present time for hazardous atmospheres which contain acetylene, hydrogen, carbon disulfide, or similar gases.

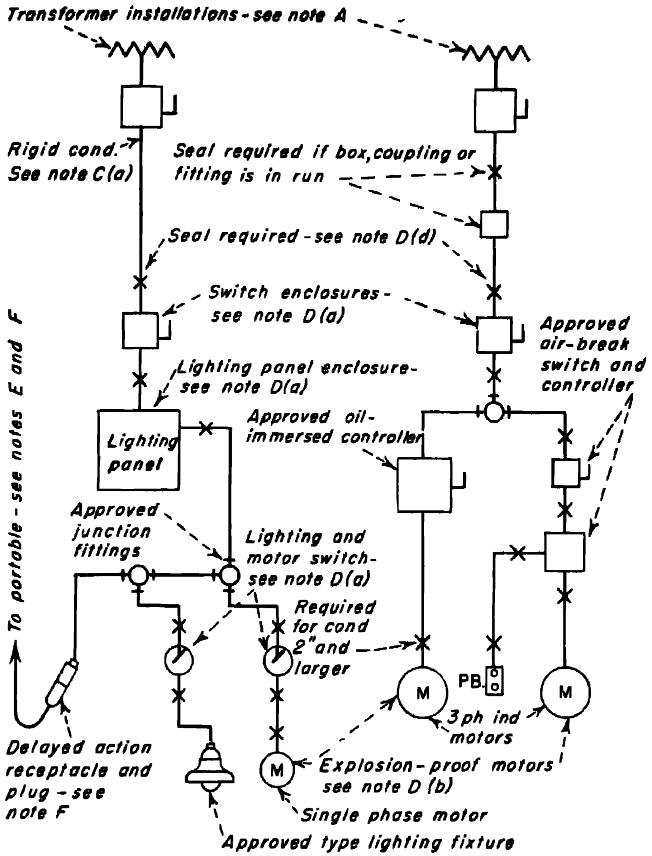

Transformer installations - see note A
Rigid cond. See note C(a)
Seal required if box, coupling or fitting is in run
Seal required - see note D(d)
Switch enclosures - see note D(a)
Approved air-break switch and controller
Lighting panel enclosure - see note D(a)
Lighting panel
Approved oil-immersed controller
To portable - see notes E and F
Approved junction fittings
Lighting and motor switch - see note D(a)
Required for cond 2" and larger
M
PB.
3 ph ind motors
Delayed action receptacle and plug - see note F
Explosion-proof motors see note D (b)
Single phase motor
Approved type lighting fixture
Class I - Division I

FIG. 13-19. Diagram illustrating requirements of Class I, Division 1 installations.

For these gases and other applications where special precautions are necessary (as in the manufacture of explosives) one of the following methods may be used:

1. *Remote Location.* Remote location means location outside hazardous area with extended shafts to transmit power to the driven machine. Lighting units outside area with light directed into area through skylights and windows.

2. *Isolation of Equipment in Special Rooms.* Isolation of equipment in special rooms is very closely related to remote location, but the equipment is set up in a special room within the hazardous area. The room is tightly constructed and equipped with positive ventilation.

3. *Use of Inert-gas-filled Equipment.* The design of the electrical equipment must provide an enclosure tight enough to confine inert gas so that only a small supply of "make-up" gas will be required. A pressure of 1 to 2 lb is usually sufficient to exclude the hazardous gas from the equipment enclosures.

Successful operation in hazardous areas depends on careful selection, application, installation, and maintenance of the electrical equipment with adequate consideration given to ventilation and the "closed system" of process operation.

Notes on the Requirements of Class 1 Installations (See Fig. 13-19)

Construction features	Division 1	Division 2
A. Transformer and capacitor vault requirements		
1. Containing a liquid that will burn		
a. Located within building	Yes—NEC 5012	Yes—NEC 5012
b. Located outdoors	Yes*	No
2. Containing a liquid that will not burn		
a. Located within building	Yes, or be of explosion-proof type	Yes, fire-resistant construction 4521 to −4. Suitable separation satisfactory for Class B insulation
b. Located outdoors	Yes*	No
B. Meters, instruments, and relays	Explosion-proof enclosure	With make or break contact, explosion-proof enclosure, or if general-purpose enclosures to have oil-immersed or hermetically sealed contacts
C. Wiring methods		
1. Conduit type	Threaded rigid	Rigid or EMT Threaded or nonthreaded
2. Boxes and fittings	Explosion-proof approved	Vapor-proof
3. Flexible connections	Explosion-proof approved	Flexible metallic
4. Sealing of conduit required	Yes, in approved manner within 18 in. of enclosures	Only where explosion-proof equipment is required
D. Type enclosures		
1. For switches, breakers, motor starters, push buttons, relays, etc.	Explosion-proof approved	Explosion-proof, general-purpose with oil-immersed or hermetically sealed contacts
2. For motors	Explosion-proof approved	Totally enclosed type when no sliding contacts are employed. Sliding contacts, switching devices, etc., shall be installed in explosion-proof enclosure
3. For lighting fixtures	Explosion-proof approved type with threaded rigid conduit stems	Vapor-proof
E. Appliances, fixed and portable	Explosion-proof approved	Explosion-proof approved
F. Special items	Receptacles and plugs of polarized type Flexible cords—hard-usage type with grounding conductor No exposed live parts Bonding jumpers with proper fittings or other approved means Flexible conduit shall be jumpered Service conductors shall be protected with lightning arresters No motors or control equipment are available at present time for hazardous atmospheres which contain acetylene, hydrogen, carbon disulfide, or similar gases	

* Not specifically specified in NEC.

Metering and Instrumentation. A minor but important item in the supply of power services to the various processes or departments is that of metering. The power requirement for each department, building, process, or section of the manufacturing plant, together with the total for each service from its point of origin, should be metered. The sum of the individual loads should be balanced daily against the total generated output or purchased power and any discrepancy corrected immediately. Any losses noted may be found more easily, increases in consumption by any division noted and explained, and controversies over service charges backed up with documentary evidence of actual consumption. Thus meters of the recording type are a worth-while investment because of their aid in detecting losses from the distribution system and in allocating equitably service costs for accounting purposes. The meter usually used for this purpose on feeder circuits is the kilowatthour meter, and if used in conjunction with an ammeter for each circuit and a voltmeter for a group of circuits, the power factor of the load may be determined. The feeder circuit and also the generator capacity are dependent on the ampere requirements and not upon the kilowatt load.

Successful plant operation depends as much upon the ability to control the process as it does on knowing how to design and construct the plant. Figure 13-20 shows the interrelation of processes in chemical plants, practically all of which lend themselves to a varying degree of process control The most important process variables are (1) time, (2) speed, (3) temperature, (4) pressure, (5) fluid flow, (6) level, (7) humidity, (8) gas analysis, and (9) substance composition.

Process industries	Unit processes																					
	Absorption	Agitating and mixing	Centrifuging	Chemical processes	Crushing and grinding	Dissolving, leeching	Distillation	Drying	Dust and fume handling	Evaporation	Extraction	Filtration	Gas and air handling	Heat technology	Liquid transferring	Material handling	Pressing-hydraulic	Refrigerating	Sifting and screening	Solvent recovery	Thickening	Weighing
Cement and lime		●		●	●			●	●				●	●	●	●			●		●	●
Ceramics and glass		●		●	●			●	●			●	●	●	●	●	●		●		●	●
Chemicals	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●		●	●	●	●	●
Coal products	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●
Electrochemical products	●	●		●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●		●
Explosives - cellulose	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●
Fertilizers		●	●	●	●	●		●	●	●			●	●	●	●	●		●			●
Glue and gelatine		●		●	●	●		●		●	●	●	●	●	●	●		●	●	●	●	●
Leather tanning		●		●	●	●		●	●				●	●	●	●	●	●	●		●	●
Paint and varnish	●	●	●	●	●	●	●	●	●	●		●	●	●	●	●			●		●	●
Petroleum refining	●	●	●	●		●	●		●	●	●	●	●	●	●	●	●	●	●		●	●
Pulp and paper	●	●	●	●	●	●	●	●		●	●	●	●	●	●	●	●		●	●	●	●
Rayon-cellulose and acetate	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●		●	●	●
Rubber	●	●		●	●	●		●	●				●	●	●	●	●		●			●
Soap		●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●	●		●	●
Sugar refining		●	●	●	●	●	●	●	●	●	●	●	●	●	●	●			●		●	●
Steam power plant					●				●	●	●	●	●	●	●	●						●

Fig. 13-20. Interrelation of processes in chemical plants.

There are, in general, three main operating principles of measurements, *viz.*, electrical and electronic, mechanical, or chemical. Instruments operating under these three principles have their own particular advantages or disadvantages. In general, except in the field of ion-concentration measurement, which is dominately chemical, and in the field of pressure, which is dominately mechanical, the instruments that are operated on the electrical or electronic principle are used more than any other type.

Electrical and electronic instruments, as a whole, are usually more accurate and more easily adapted to remote operation than other types. It is necessary only to lengthen the lead wires connecting the detector and the indicating or control mechanism In a process that is equipped with adequate measurement and control instrumentation, the man power required to follow and supervise the operation is kept to a minimum and used at maximum efficiency.

Determination of Fault Currents and Interrupting Capacity of Protective Equipment*

In an electrical circuit it is necessary to employ some form of protective interrupting device since there is always the possibility of an electrical fault. The protective devices must have not only adequate current-carrying capacity and proper voltage rating but also (1) ample interrupting capacity so as to prevent damaging effects of continued short circuit on apparatus and wiring and (2) the ability to withstand without injury the high momentary current liable to exist before interruption.

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* Superior numbers in this section refer to the entries listed under References on p. 959.

It has been common belief that short circuits of 10,000, 15,000, or 20,000 amp are the maximum that can occur in low-voltage circuits. However, greater faults can and do occur. The value of fault currents in any electric circuit at any given voltage depends upon the impedance of the circuit. The magnitude of the short circuit is equal to the quotient of the voltage divided by the impedance.

The exact computation of the maximum possible short-circuit current in a given circuit is very complicated, involving intricate mathematical analyses and complete system-layout studies. For industrial-plant systems, we are fortunate that there is a relatively simple manner of arriving at approximate values that are close enough to actual values for all practical purposes.

This method involves simply the calculation of the maximum symmetrical short-circuit current which is then multiplied by established factors to obtain the total short circuit to be interrupted and the momentary current which the device must withstand.

In a-c circuits, all generators, synchronous motors, and induction motors are sources of short-circuit current. The resistance of these machines and transformers is usually negligible compared to the reactance, and, consequently, the reactance and the impedance are considered to be the same. Short-circuit currents are dependent upon the subtransient of the machines. The initial heavy surge diminishes rapidly, in a matter of cycles, and in induction machines dies out. In synchronous machines the subtransient current is followed by a current dependent upon the transient reactance. The final sustained short-circuit current is dependent upon the synchronous reactance of the machines.

Since the maximum interrupting duty of low-voltage air circuit breakers is assumed to occur ½ cycle (60-cycle basis) after the fault occurs, it is, therefore, necessary only to consider the effects of the subtransient current values.

In addition to the initial subtransient current, there is also present a d-c component which tends to displace the a-c component, increasing this initial value. This is illustrated by Fig. 13-21 which shows a typical curve of a short-circuit current in one phase of a low-voltage three-phase system.

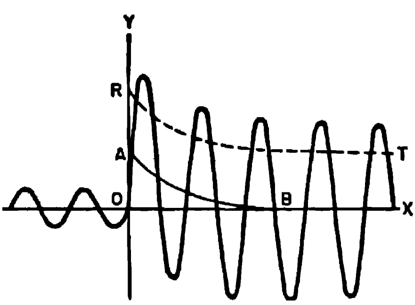


Fig. 13-21. Typical curve of a short-circuit current.

The wave to the left of OY represents the current previous to the fault, and the wave to the right represents the short-circuit current. The curve AB represents the d-c component, which has the effect of displacing the total wave by the amount shown by AB. After a few cycles, the d-c component dies down and leaves only the a-c component, which is symmetrical with respect to the X axis. During the existence of the d-c displacement, the total rms value is represented by the curve RT.

The rate of decay of the d-c component is a function of the ratio of the total of the machine subtransient reactance plus the external reactance, to the total circuit resistance X/R. The higher the value of the resistance, relative to the reactance, the faster the d-c component decays The initial value of the d-c component is dependent upon the point of the voltage wave at which the short circuit is established. It is maximum when this occurs at the zero point on the voltage wave, and its theoretical maximum is equal to 1.732 times the symmetrical current. A wave with this maximum d-c component, however, requires half a cycle to reach its crest value, and during this time, sufficient decay will take place so that a lower factor can safely be used. The currents in the other two phases are less than this maximum.

Air circuit breakers are applied on the basis of the average current in a three-phase system. In low-voltage systems, the X/R ratios are invariably low. On the basis of a thorough study, a multiplying factor of 1.25 has been adopted as representing average conditions at an instant ½ cycle after the short circuit occurs. This means that the total average current at this time, under maximum conditions, will be 1.25 times the initial three-phase symmetrical current, and it is this value we use for the interrupting and momentary ratings.

This represents the worst condition possible with the exception of a phase-to-neutral short circuit at the terminals of a delta-wye grounded-neutral transformer. However, this condition is so rare and unusual that its consideration can safely be omitted.

The reactance or impedance of a generator or transformer is generally expressed in per cent values based upon the rated kva rather than in ohms. Five per cent reactance in a circuit means that, when rated current is flowing, the voltage drop due to the reactance is 5 per cent of the rated voltage of the circuit.

Any ohmic value of resistance, reactance, or impedance in a circuit may be expressed in per cent as follows:

$$\text{Per cent on base kva} = (100 \times \text{base kva} \times \text{ohms} \times 1{,}000)/\text{volts}^2 \tag{1}$$

Any per cent value expressed in terms of one base can be expressed in terms of another base as follows:

$$\frac{\text{Per cent base 2}}{\text{Per cent base 1}} = \frac{\text{kva base 2}}{\text{kva base 1}} \tag{2}$$

In order to avoid the use of the value 100 when using per cent, values are sometimes expressed in per-unit values. The relationship is as follows:

$$\text{Per cent values} = 100 \times \text{per-unit value}$$

Systems with Primary Sources of Power. In low-voltage systems the impedances of the machines, conductors, and other equipment may not be negligible; and if necessary to consider them, the following formula can be used:

$$I = \frac{\text{kva} \times 1{,}000}{\sqrt{3} \times \text{voltage}} \tag{3}$$

where I = the normal current corresponding to the kva base used

$$I_i = (100 \times 1.25 \times I)/Z_T \tag{4}$$

where I_i = interrupting capacity or the symmetrical short-circuit current times the 1.25 factor for the d-c component at ½ cycle, and Z_T = the total impedance of the circuit in per cent

$$Z_T = \sqrt{(r_m + r_t)^2 + (X_m + X_t)^2} \tag{5}$$

where r_m = the resistance of the rotating machines
r_t = the sum of the resistance of the conductors and other equipment in the circuit to the point of short circuit
X_m = the subtransient reactances of the rotating machines
X_t = the sum of the reactances of the conductors and other equipment in the circuit to the point of short circuit

All the above are in per cent values on the same kva base.

In the general equation $Z = \sqrt{r^2 + x^2}$, the resistance combines with the reactance in quadrature. Unless r has values equal to one-third or more of the value of x, the resistance does not have an appreciable effect upon the value of Z. If resistance may be neglected, Eq. (4) reduces to

$$I_{sc} = \frac{100 \times 1.25 \times I}{X_m + X_e} \tag{6}$$

Now if the external reactance X_e is of a negligible value, Eq. (6) can be further reduced to

$$I_{sc} = (100 \times 1.25 \times I)/X_m \tag{7}$$

It is obvious that Eq. (7) will give a higher value of short-circuit current than either Eq. (4) or Eq. (6), but it is sufficiently close for all practical purposes.

Three-phase Transformers—Unlimited Primary Supply. The foregoing applied to systems where primary sources of power were present. A common application of low-voltage air circuit breakers occurs in systems that are served by power transformers. In such applications, the characteristics of the system back of the transformer are usually not known, and it is frequently the practice to consider the transformer as the source, in which case the primary voltage is then maintained constant even during a short circuit in the secondary. This is based on unlimited power available at the primary terminals and results in the maximum possible short-circuit current in the transformer secondary circuit. In this case Eq. (8) can be used:

$$I_{sc} = (100 \times 1.25 \times I)/Z_t \tag{8}$$

where Z_t = impedance of the transformer in per cent.

Three-phase Transformers—Limited Primary Supply. If the total circuit impedance of the system back through the transformer is known, the value of I_{sc} is calculated from Eq. (4), but if these data are not available, it may be assumed that the interrupting rating which is required for the power circuit breaker in the primary of the transformer is indicative of the short-circuit capacity available, and this provides an approximate means for determining the magnitude of the secondary short-circuit current. When a kva base limit is thus chosen, the impedance of this base is 100 per cent.

For calculating the short-circuit current of the transformer in such a case, the primary kva per cent impedance is converted to the transformer kva base [Eq. (2)]. The following formula can then be used:

$$I_{sc} = \frac{100 \times 1.25 \times I}{Z_p + Z_t} \tag{9}$$

where Z_p = impedance of the primary kva based on transformer kva in per cent.

A more conservative estimate will result when the maximum momentary instead of the interrupting rating of the primary circuit breaker is considered. This is due to the fact that modern low-voltage air circuit breakers are faster opening than older air circuit breakers. The kva rating at opening would therefore be higher than that indicated by the kva interrupting rating of the primary breaker.

Contribution by Motors to Short-circuit Current. Motors make up a considerable part of the load of a transformer. Because they generate at the time of a fault and thus feed energy into the fault, their contribution to the short-circuit current is important.

Induction motors as well as synchronous motors must be included. At the instant of short circuit there is magnetic flux in the core of the secondary of the induction motor. This flux cannot vanish immediately but is maintained by the current flowing in the secondary and will decrease in keeping with the resistance and inductance characteristics of the motor circuit. This decreases very rapidly but must be included because of the high opening speed of the circuit breaker. (This generator action of the induction motor is different from that of the induction generator.)

Synchronous motors continue to receive their normal excitation at the time of the short circuit, run as synchronous generators by inertia, and act like synchronous generators in delivering energy into the fault. The current from synchronous motors decreases less rapidly than that from induction motors.

For estimating purposes the approximate symmetrical values of short-circuit current contributed by the motors at an instant ½ cycle after the short circuit happens are 3.6 times full-load current for induction motors and 4.8 times full-load current for synchronous motors.

In practice it can be assumed that, for a 208/120-volt installation, the motor load will be 50 per cent of the transformer kilovolt-ampere, and for 240-, 480-, and 600-volt installations the motor load is 100 per cent of the transformer kva. It can be further assumed that the motor load will be 75 per cent induction and 25 per cent synchronous.

Using all these assumptions, the total motor short-circuit current to be added to the short-circuit current of the transformer for a given kva transformer will be as follows:

For a 208/120-volt transformer

$$I_{scm} = 2 \times I_t$$

For a 240-, 480-, or 600-volt transformer

$$I_{scm} = 4 \times I_t$$

where I_t = full-load current of the transformer, and I_{scm} = symmetrical short-circuit current contributed by motors.

These values will then be multiplied by 1.25 to include the d-c component.

For a 208/120-volt transformer

$$I_{icm} = 2.5 \times I_t \tag{10}$$

For a 240-, 480-, or 600-volt transformer

$$I_{icm} = 5.0 \times I_t \tag{11}$$

where I_{icm} = interrupting capacity for motors or symmetrical short-circuit current contributed by motors times the 1.25 factor for the d-c component at ½ cycle.

EXAMPLES: 1. Three-phase transformer—unlimited primary supply, 1,000 kva, 480-volt secondary, 5½ per cent impedance.

Required: The interrupting capacity (fault located at X).

$$I = \frac{1{,}000 \times 1{,}000}{\sqrt{3} \times 480} \tag{3}$$

= 1,200 amp, normal full-load current of transformer

$$I_{ic} = (100 \times 1.25 \times 1{,}200)/5.5 \tag{8}$$

= 27,300 amp, interrupting capacity (required from transformer)

$$I_{icm} = 5 \times 1{,}200 \tag{11}$$

= 6,000 amp, interrupting capacity required from 100 per cent motor load

The combined interrupting capacity is therefore

$$27{,}300 + 6{,}000 = 33{,}300 \text{ amp}$$

A 50,000-ampere-interrupting-capacity circuit breaker would be selected.

2. Three-phase transformer—limited primary supply, 25,000 kva, interrupting capacity of primary circuit breaker, 1,000 kva, 240-volt secondary, 5½ per cent impedance.

Required: The interrupting capacity (fault located at X).

$$I = \frac{1{,}000 \times 1{,}000}{\sqrt{3} \times 240} \tag{3}$$

= 2,400 amp, normal full-load current of transformer

$$\text{Per cent} = 100 \times 1{,}000/25{,}000 \tag{2}$$

= 4.0 per cent primary impedance based on transformer kva

$$I_{ic} = \frac{100 \times 1.25 \times 2{,}400}{4.0 + 5.5} \tag{9}$$

= 31,600 amp, interrupting capacity required from transformer

$$I_{icm} = 5 \times 2{,}400 \tag{11}$$

= 12,000 amp, interrupting capacity required from 100 per cent motor load

The combined interrupting capacity is therefore

$$31{,}600 + 12{,}000 = 43{,}600 \text{ amp} \tag{12}$$

A 50,000-ampere-interrupting-capacity breaker would be adequate, but the full-load current of the transformer is 2,400 amp. A breaker of this continuous current rating will have a 75,000-amp interrupting rating and would therefore be selected.

In the two preceding examples we have assumed the presence of a motor load and the fault so located that the motors contribute to the short-circuit current. This gives us the worst possible condition, considering the switchgear and connected load as a whole. Applications are sometimes encountered where a short circuit can occur at such a place that the motor load will not contribute energy to it. The circuit breaker should then be chosen on the basis of the energy supplied by the transformer alone, omitting the energy flowing from the motors. In example 1 the short-circuit current without motors would then be 27,300 amp, and in example 2 the short-circuit current would be 31,600 amp.

Direct-current Systems. The fault current to be considered in d-c systems is the maximum current obtainable and not some lower value during the initial rise. When a short circuit occurs, the current rises rapidly, and the final value depends on the resistance in the circuit. The rate of current rise depends on the resistance and the inductance in the circuit

For calculating the short-circuit current, it is necessary to have information concerning the maximum short-circuit current obtainable at the terminals of the machines.

For estimating purposes, the data in Table 13-9 may be used to obtain the approximate maximum short-circuit currents at the terminals of d-c machines

Table 13-9

Type of machine	*Times K = normal current*
Shunt-wound	10
Compound wound	10
Compound-wound with compensating field	15
Rotary converters	15

Using Table 13-9 the short-circuit current for a fault at the machine terminals will be

$$I_{sc} = K \times I \tag{13}$$

where I = full-load current of the machine

If the air circuit breakers are remote from the machine, the resistance of the external circuit should be included in the calculations, and the short-circuit current is

$$I_{sc} = E/(R_m + R_e) \tag{14}$$

where E = the voltage of the circuit

R_m = the equivalent machine resistance or

$= E/(K \times I)$

R_e = the external resistance to the point of the fault

Substituting the value of R_m in Eq. (14), the value for short circuit can be computed from the following formula

$$I_{sc} = \frac{K \times E \times I}{E + (K \times R_e \times I)} \qquad (15$$

In a d-c generator the final current depends upon the winding and upon the demagnetizing effect of the current flowing The rate of current rise depends upon the voltage and capacity of the machine, being lower in higher voltage machines The rate of rise of a standard 1,000-kw machine may be approximately 1,500,000 to 1,800,000 amp per sec

Storage batteries have low values of internal resistance and inductance and give rise to high currents and high rates of rise of current.

Mercury-arc rectifiers also have inherently low resistance and inductance values which produce very high values of rate of rise and final short-circuit currents. The actual value cannot be approximated without knowing the characteristics of the particular rectifier involved. A number of installations have shown a rate of rise of from 6½ to 8 million amp per sec.

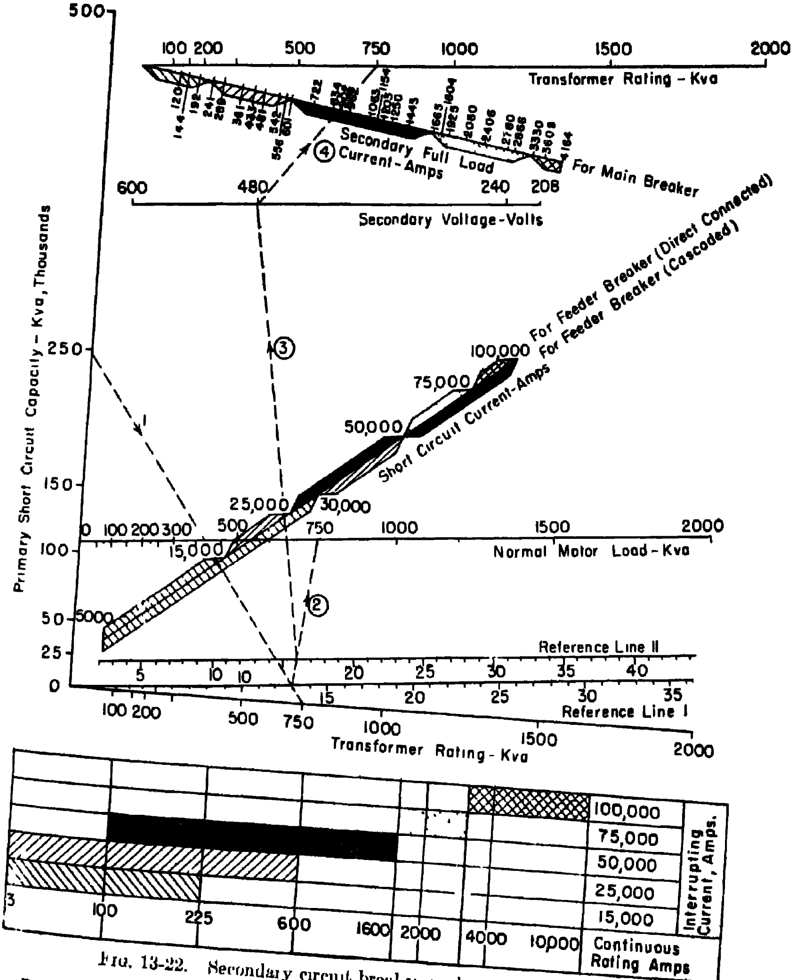


FIG. 13-22. Secondary circuit breaker application nomogram.[6]

By using Fig. 13-22, the interrupting capacity and short-circuit currents in a-c circuits can readily be determined by three settings of a straightedge as follows:

1. Place a straightedge on the value of primary short-circuit capacity and on the transformer rating on the abscissa, finding a point on reference line I. If an infinite system is assumed, place the straightedge on the value of the transformer kva and perpendicular to the axis of transformer kva.

2. Using this point and the kva of normally connected motor load, find a point on reference line II. Motor load may vary from 0 kva to the full transformer rating

(100 per cent motor load is usually assumed for 240-, 480-, and 600-volt systems and 50 per cent motor load for 208Y/120-volt systems).

3. Using this point on reference line II and the value of the secondary voltage, the rms value of the secondary short-circuit current will be found as well as the breaker with the correct interrupting capacity for direct-connected or cascaded feeder breakers.

4. Check continuous current ratings given on chart of breaker sizes to be certain that breakers with necessary interrupting capacity as determined by nomogram have adequate continuous current-carrying capacity. The load on each feeder circuit determined this current.

Cascading of Breakers. Low-voltage air circuit breakers are normally applied on the basis of adequate interrupting capacity and current-carrying ability, both for normal loads and fault currents for short-time intervals. Under these conditions, standard interrupting-duty cycles apply. Selective tripping based on time discrimination can easily be arranged to isolate faults and not interrupt power to normal circuits

It is not always economical, however, to apply circuit breakers in the conventional method. If the load consists of an assembly line or other related processes which make it necessary to shut down all other circuits, breakers may be applied on the cascade principle. Under these conditions, certain breakers may be subjected to currents above their published interrupting ratings, and a restricted interrupting-duty cycle results.

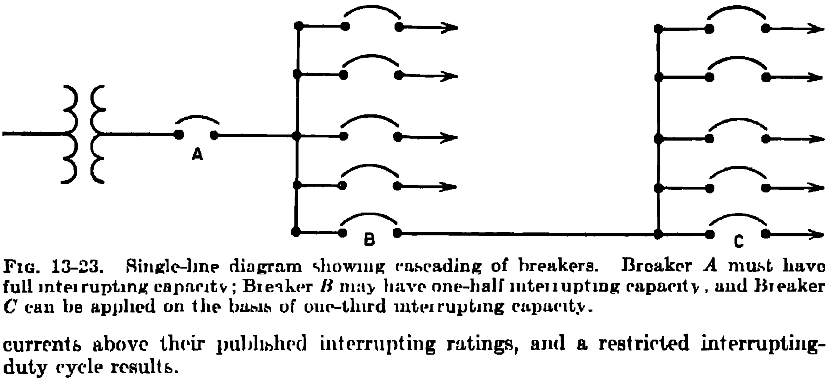


Fig. 13-23. Single-line diagram showing cascading of breakers. Breaker *A* must have full interrupting capacity; Breaker *B* may have one-half interrupting capacity, and Breaker *C* can be applied on the basis of one-third interrupting capacity.

It is evident that, if two or more breakers operate in series to clear a fault, the stresses on them are less than if they opened individually. This condition is recognized, and breakers farthest from the source of power in cascade arrangements may be applied with interrupting ratings less than the fault currents to which they will be subjected Each breaker in these cascade circuits must be free to trip instantaneously when the breaker next lower in the cascade is subjected to currents above its published interrupting rating. This means that selective isolation of faults is not possible on currents exceeding the interrupting ability of the smallest breakers.

Breakers that interrupt currents above their rating, as evidenced by the opening of the breaker next higher in the cascade, must not be closed even once without inspection. They may be damaged so that normal-load current cannot be safely carried, or their interrupting ability may be impaired.

Cascading is limited to three steps of interrupting capacity, as shown in Fig. 13-23. The breaker nearest the source of power must have adequate interrupting capacity for the circuits in which they are applied. The breakers next lower in the cascade may have one-half the interrupting rating required by the maximum fault currents, and those farthest from the source may have one-third this rating.

For any application requiring full selective isolation of faults for maximum continuity of service, breakers with full interrupting capacity must be chosen for all positions. The reduced selectivity, restricted interrupting-duty cycle, and some additional maintenance which can be expected with cascaded breakers are often compensated for, however, by large savings in the initial cost of the switchgear. Under these conditions, cascaded arrangements may be preferred.

Fuses. In some cases, fuses and switches are suitable for circuits, the fuse supplying the protection and the switch the control. The operation of any fuse (both high and low voltage) under short-circuit conditions is so fast that it clears during the subtransient conditions and before the d-c component has had time to die out. The capacity must therefore be based on the total current including the transient offset. In the general case this is 1.6 times the symmetrical short-circuit current, but for fuses rated 15 kv and below, not applied directly to a generator bus of high capacity, the resistance introduced by lines and intervening apparatus is so great that the total current will seldom exceed 1.2 times the calculated symmetrical short-circuit current. Application on this basis is considered safe. However, the contribution from motors must be added to the current from the power source.

Demand Factors

Demand factor is the ratio of the maximum use of power to the total load connected expressed as a percentage. In the average plant, the demand factor usually falls between 60 and 80 per cent.

In the design of an electrical system of an industrial plant, one of the most difficult problems is to determine how much electrical power will be required for different load groupings and the various types of operations for the entire plant. The loads occurring coincidently between various drives, departments, or areas of the plant always lead to loads that are less than the total connected load. Also the total over-all plant load will be less than the peak loads of the various departments or areas combined. The load grouping and determination of the diversity and demand factor is the key to a more practical and economical design.

To determine the demand factor with a fair degree of accuracy, it is necessary to investigate many items. Some of the most important are listed below:

1 What equipment will be required to operate simultaneously and under what loads.

2 What pieces of apparatus will require starting together and which machines will be running at that time and under what loads.

3 The load densities commonly encountered, including both power and load, usually range from about 8 to 25 va per sq ft.

4. Lighting loads for up-to-date lighting-intensity levels range from 4 to 6 va per sq ft for incandescent lamps and 2 to 4 va per sq ft for fluorescent and mercury lamps (see Fig 13-43).

In converting all ratings, such as horsepower, into kva, it is usually sufficiently accurate to allow for loads in accordance with Table 13-10.

Table 13-10

Kva	*Type of load*
1.0	Each induction motor hp
0.9	Each 100 per cent synchronous motor hp
1.0	Each 80 per cent synchronous motor hp
1.0	1,000 watts incandescent lighting
1.25	1,000 watts fluorescent lamps equipped with standard starters
1.0	Each 1 kw of resistance heaters, ovens, etc.

Table 13-11 gives values of typical demand factors for over-all loads of various types of industrial plants and represents the average maximum for a group of medium- and large-size plants in the same industry. In general, the more nearly an industry may be classified as a "continuous process," the higher will be both the demand factor and the annual load factor.

Lightning Protection

All electrical equipment connected directly or through cables to overhead lines is exposed to lightning. Switchgear and other apparatus may also have additional exposure through power transformers that are connected to aerial lines. Both manufacturers and users have for years been interested in finding means of preventing, so far as economically feasible, damage from lightning.

An especially important contribution toward this objective has been a better understanding of the nature of lightning. Through the use of high-speed cameras, oscillographs, curve voltage records, and surge-crest ammeters, information on the formation of lightning strokes and the magnitude, front, and length of current and voltage waves resulting from lightning discharges has been recorded.

As a result of the better understanding of the actual character of lightning, it is possible to evaluate the effectiveness of protective equipment and to set up protection rules so that troubles from lightning will be minimized.

Table 13-11. Typical Demand Factors for Various Types of Industrial Plants

Industry (manufacturing processes)	*Demand factor ratio of max demand / connected load*
Agricultural implements	0.65
Aluminum manufacturing	0.40
Asbestos products	0.50
Bakery (gas fired ovens)	0.45
Cement	0.65
Chemicals	0.50
Copper and brass products	0.50
Dyeing and finishing textiles	0.75
Electrical machinery, apparatus, and supplies	0.65
Engines, turbines, and water wheels	0.50
Explosives manufacturing	0.55
Flour mill and elevator	0.65
Foundry and machine shop products	0.65
Gas manufacturing	0.60
Iron and steel (blast furnaces, rolling mills)	0.45
Knitting mills	0.85
Laundry	0.80
Lumber and timber products	0.65
Machine shops, miscellaneous stampings, machine parts, welded assemblies, etc.	0.25
Marble and other stone products	0.65
Paints and varnishes	0.70
Paper and pulp	0.75
Petroleum and oil refining	0.60
Pigment manufacture	0.40
Printing and publishing	0.80
Pipe manufacture (wrought iron)	0.35
Refrigerators (mechanical)	0.70
Rubber goods, tires, and tubes	0.75
Rayon manufacture, chemical house excepted	0.80
chemical house included	0.65
Slaughtering and packing	0.70
Shell manufacture	0.40
Soap manufacture	0.60
Sugar refining	0.65
Tin cans and other tinware manufacture	0.70
Wood distillation and charcoal manufacturing	0.60

Surge voltages on lines are limited in maximum values to the flashover of the insulation to ground. The current in these surges is also limited by the surge impedance of the line. For example, the surge impedance may be on the order of 500 ohms. For a 1,000-kv surge, the current would be approximately 2,000 amp. While this current is not excessive, the voltage would be dangerous to equipment if protective equipment were not used.

Direct strokes of lightning may involve currents as high as 200,000 amp, rising at the rate of thousands of amperes per microsecond. However, only a small percentage of direct strokes will be higher than 20,000 amp.

Figure 13-24*a* shows curves of a surge on a 13.2-kv circuit and the voltage reduction by the discharge of the lightning arrester. It is seen that, with good lightning arrester grounds, the voltage during a surge will be below the basic insulation level of the circuit. However, if the arrester ground has a high resistance, safe voltage may be exceeded. The conditions that must be met by lightning arresters in application are as follows:

1. Voltage across the arrester during a discharge must not exceed a certain predetermined value which is safe for the insulation to be protected.

2. The circuit should not be disturbed by the arrester's action. Little or no system current should follow the impulse.

3. The arrester should be so constructed that it has a factor of safety against failure such that the arrester will not fail under usual service conditions.

4. The characteristics of the arrester should remain practically constant during the life of the arrester, and under normal service its life should be at least as long as the apparatus which it protects.

The most vulnerable part of the electrical system is the overhead aerial lines. In cases where the circuits are highly exposed to lightning of appreciable frequency and intensity, system outages can be very largely prevented through the use of (1) overhead ground wires, (2) counterpoises, (3) expulsion tube protectors, (4) ground-fault neutralizer coils, and (5) high-speed relay. Of these, only the overhead ground-wire scheme is generally used on overhead lines in industrial-plant systems of 13.2 kv and lower.

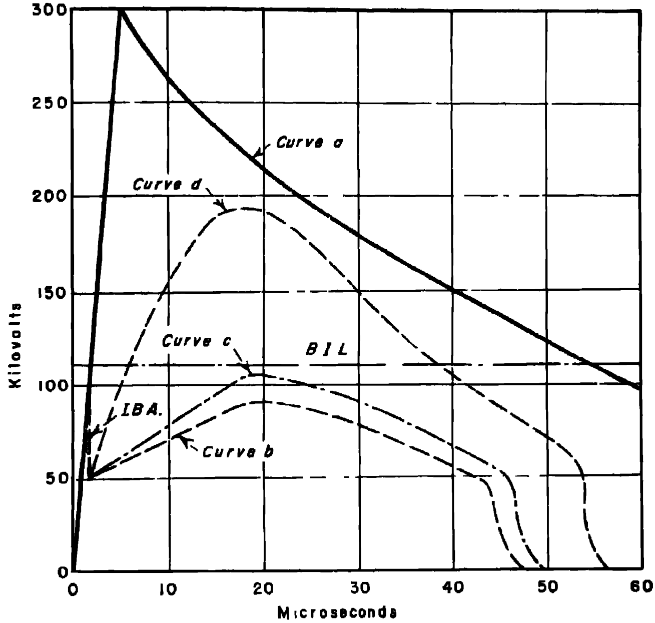


Fig. 13-24a Voltage reduction by arrester discharge on 13.2-kv circuit Curve a—surge 5-40 wave on circuit. Curve b voltage across arrester only. Curve c voltage from arrester to ground (0 ohm ground resistance and 40-ft ground lead). B.I.L basic-impulse insulation level, 15-kv class. I.B.A impulse breakdown of arrester.

The effectiveness of the overhead ground-wire protection in preventing arc-over of insulator strings depends on its resistance to ground. For adequate protection of motors when they are supplied with power from overhead lines, the lines should be shielded against direct strokes.

Virtual immunity from oil-insulated-type transformer failures is being obtained through the use of the most modern valve-type arresters, particularly where interconnection is made between the ground terminal of the primary arrester and the ground connection of the secondary, thereby shunting the transformer insulation to be protected. Air-cooled transformers have a BIL (basic impulse level) considerably lower than oil-insulated transformers (see Table 13-12). If the former are subject to surge voltages, they should be protected by special tube-type arresters located as close as possible to the transformer terminals with a second set of standard line-type arresters connected on the line about 500 ft ahead of the transformer.

When aerial lines of any voltage enter cables, the terminals should have adequate protection. This protection is usually secured by an arrester located at the junction of the cable and aerial line with the arrester ground connected to the cable sheath.

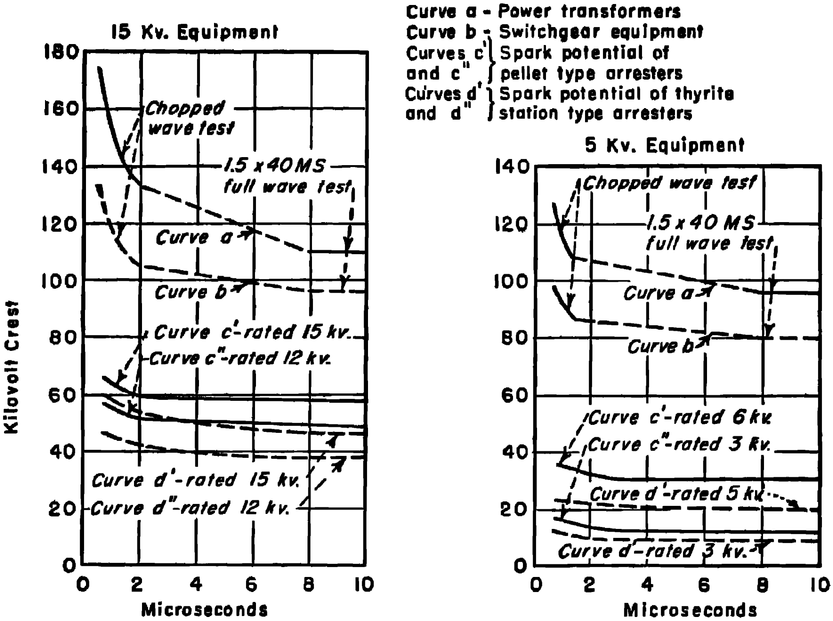


FIG. 13-24*b*. Impulse strength and protective characteristics for industrial-type equipment.[7]

The cable sheath should also be grounded at the other end as a ground at only one end will not be effective in keeping the sheath at the opposite end effectively grounded.

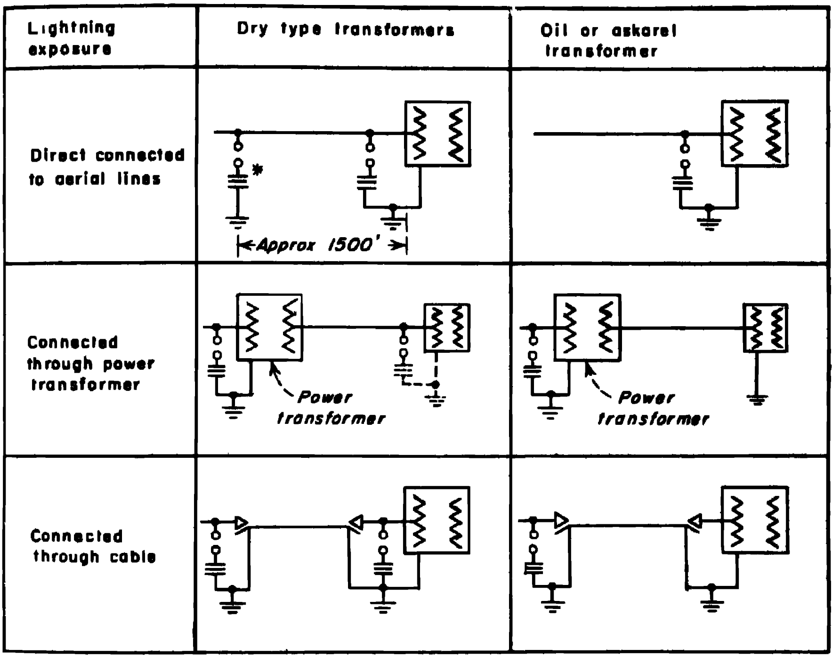
Table 13-12. Performance Characteristics of Thyrite Station-type Lightning Arresters*†

Arrester rating, kv (rms)	Station voltage class		Transformer insulation class, kv	Transformer impulse levels, 1.5 × 40 ms kv crest		Arrester impulse sparkover in kv crest		Arrester discharge voltage on 10 × 20 ms current wave	
	Ungrounded neutral	Grounded neutral		Oil or askarel	Dry type	Front of wave	1.5 × 40 ms wave	1,500 amp	3,000 amp
3.0	2,400	4,160	5.0	60	25	12	10	9	10
4.5	4,160	4,800	5.0	60	25	16	14	15	16
6.0	4,800	7,200	5.0–8.66	60–75	25–35	21	19	18	19
7.5	7,200	.....	8.66‡	75	35	26	23	22	24
9.0	.....	11,500	15.0	95	50	31	27	27	29
12.0	11,500	13,200	15.0	95	50	40	35	36	38
15.0	13,200	.....	15.0	95	50	52	45	45	48

* HUNTER, E. M., and T. E RODHOUSE, Indoor Transformers, *Gen. Elec. Rev.*, November, 1947.
† These arresters are also used for protection of a-c rotating machines.
‡ Applies only to three-phase transformers.

Lightning protection on transformers alone will not ensure protection to plant equipment.

A most effective scheme for protecting a-c rotating machines has been in use for a number of years for use with generators, motors, or synchronous condensers which may be subjected to impulses either from directly exposed lines or transmitted through transformers. The usual lightning arrester is here supplemented with protective capacitors, connected from bus or machine terminals to ground, thereby reducing the steepness of the traveling-wave voltage entering the machine. The amplitude of the impulse wave is limited by the use of line arresters 1,000 to 1,500 ft out on the circuit, together with station-type arresters in parallel with the capacitors.



*May be distribution type - all other type arresters indicated should be station type

FIG. 13-25. Arrester-protection for different types of exposures

Buildings, tanks, etc., containing explosives and inflammable liquids and vapors may be adequately protected from being struck by lightning by properly placed mast or overhead shielding wires connected to low-resistance grounds.

In evaluating the effectiveness of lightning protection equipment it is necessary to plot the impulse strength and protective characteristics of the various types of apparatus under consideration. Representative volt-time curves for 5- and 15-kv switchgear equipment, power transformers, and applicable arresters are shown in Figs. 13-24b and 13-25.

Industrial-system Grounding

The term "grounding," as applied to electrical circuits and equipment, has many complex ramifications. Grounding the circuits in industrial plants acts a double safety check: (1) Faults in electrical circuits are giving a low-resistance path to ground to operate protective devices quickly; and (2) the plant personnel is protected against dangerous shock hazards because the voltage to ground of all parts of the wiring system is limited. In both these safety checks the grounding can be effective only if it is properly installed according to good engineering practice with a full understanding of its purpose.

The best insulated and most carefully installed electrical system does not always stay perfectly insulated. There is always the possibility of accidental grounds. There are two general types of ground: equipment grounds where the non-current-carrying metal parts (or frames) of electrical equipment and wiring enclosures are grounded; and system grounds, where the neutral or one conductor of the system is permanently connected to ground.

The metallic parts that must be grounded include conduits, sheaths or armor of cables, frames or cases of switches, motor starters, controllers, fuse boxes, junction boxes, switchgear, transformers, generators, motors, and similar items. The National Electrical Code (Art. 250, 1947) specifies the methods and requirements for grounding equipment. The Code, however, requires that the resistance of the ground shall not exceed 25 ohms. The current flowing in the grounding conductor can be calculated by Ohm's law.

If we have a lighting circuit with 120 volts between the live wire and ground, a ground resistance of 10 ohms, and a short circuit between the live and ground wires, the current flowing in the grounding circuit will be 12 amp, which can be continuously on a 15-amp fuse. Hence, the fuse protecting the grounded circuit will not "blow" unless and until other load is placed upon that circuit. If the grounding resistance is reduced to 5 ohms, the current will be 24 amp, and the protecting fuse or circuit breaker will open in a few seconds. It is obvious that, the lower the resistance to the flow of electricity through the grounding conductors, the greater will be the protection. It is recommended that the resistance of all the grounds installed be no greater than 5 ohms.

Electrical circuits are grounded in order to (1) limit the voltage upon a circuit upon breakdown of insulation and accidental contact with wires of circuit of higher voltage, (2) protect against transient overvoltage due to a switching or lightning surge, (3) prevent the establishment of a voltage upon equipment which is normally assumed to be at ground potential, and (4) possible improvement of lightning-arrester protection for overhead transmission circuits.

The largest single factor in grounding of the electrical system of industrial plants is that accidental faults are either immediately cleared or otherwise detected before they develop into a double line-to-ground fault, tripping out a considerable portion of the plant load. Ground detectors can of course be used with an ungrounded system but will not indicate the location of the fault. Accidental grounds in an ungrounded system which are not detected or immediately corrected constitute a hazard for the following reasons: (1) A ground on another conductor puts a fault on the system, which may occur on a circuit other than the first, shutting down both circuits or, in some cases, the entire plant system; (2) a line-charging current flows into the fault, and even though small, it may, in time, cause considerable damage, particularly if the fault is internal to apparatus, such as a generator or motor; and (3) arcing faults can produce surge voltages as high as five or six times normal line-

Table 13-13. Low-frequency Test Voltage*

(Voltage expressed in number of times system line-to-neutral voltage)

Nominal system voltage	Oil-insulated transformers	Dry-type transformers	Motors	Usual values for wiring
600 and below	28.8	11.5	6.1	2.5 6.0
2,400	10.8	7.2	4.1	4.0–5.0
4,160	7.9	5.0	3.7	3.0–4.0
4,800	9.4	6.9	3.5	3.0–4.0
6,900	6.5	4.8	3.6	2.0–2.5
13,800	4.3	3.9	3.4	2.0

* Witzke, R. L., Neutral Grounding Limits—Industrial System Faults, *Elec. World*, May 10, 1947.

to-ground voltage. Table 13-13 shows that this level of surge voltage may exceed the apparatus test voltage for all systems above 600 volts.

Some utility companies object to grounding systems proposed by industrial plants. Of course, if the industrial system is served directly connected at the utility company's operating voltage, the scheme of grounding requires proper coordination with that of the utility system.

The two systems are usually interconnected by means of a transformer bank with at least one winding of the bank normally connected in delta. This delta-connected winding will make grounding of each system independent of grounding of the other with the result that the grounding of the low-voltage system will have no effect on the high-voltage system or vice versa.

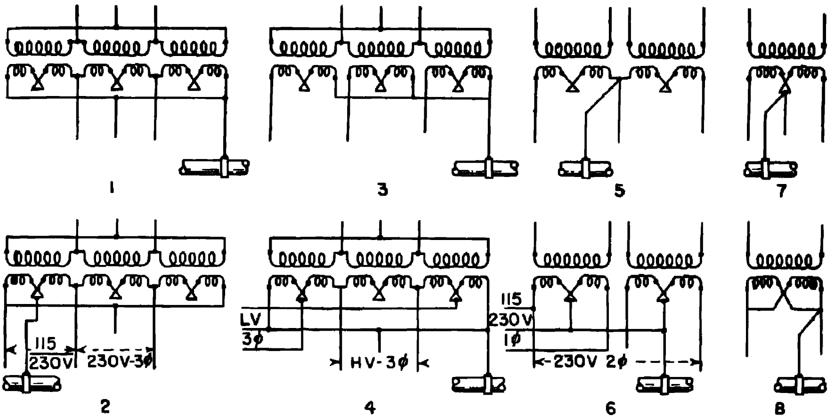


Fig. 13-26 Grounding connections for secondary distribution circuits. (1) Delta-connected power transformers—secondary for three-phase three-wire installations. (2) Delta-connected power transformers—secondary for three-phase 230-volt power and 115/230-volt single-phase lighting installations. (3) Wye-connected power transformers—secondary for three-phase three-wire or four-wire installations. (4) Delta-connected power transformers—secondary for three-phase three-wire half-voltage employing common-phase ground. (5) Two transformers—secondary for two-phase three-wire installations or three-phase transformers—secondary for three-phase three-wire open-delta installations. (6) Two-phase transformers—secondary for 230 volts, two-phase four-wire and 115/230-volt single-phase three-wire installations. (7) Single-phase transformer—secondary for 115/230-volt three-wire installations. (8) Single-phase transformer—secondary for single-phase two-wire installations.

Systems at 600 Volts and Below. The National Electrical Code (1947) specifies that secondary a-c systems supplying interior wiring and interior a-c wiring systems shall be grounded if they can be so grounded that the maximum voltage to ground does not exceed 150 volts. It also recommends that a-c systems with voltage to ground that does not exceed 300 volts be grounded and states that higher voltage circuits may be grounded. Figure 13-26 shows diagrams where such grounds are required.

The great majority of 460- and 575-volt three-phase three-wire power systems are operated with the neutral or one-phase conductor ungrounded. The reasons for not grounding these low-voltage power systems is that the apparatus at these voltages usually has considerably more margin in insulation than apparatus for higher voltages. The principal advantage of an ungrounded system is its ability in some cases to clear ground faults without an interruption to the service. Also, the line-to-ground voltage is 57.8 per cent of the phase-to-phase voltage, but with the possibility of accidental grounds, it is doubtful whether the ungrounded system is less hazardous than the grounded-phase system. The problem in these low-voltage systems is not the protection of equipment but the possible personnel hazard due to large ground fault currents.

Figure 13-27*a* shows what may happen when faults occur on ungrounded systems with high-resistance-equipment grounds.

Figure 13-27*b* shows that personnel hazards are removed from electrical systems by proper and adequate equipment and circuit grounds. The grounded-phase circuit shown in Fig. 13-26 possesses some distinct operating advantages:

1. The piece of apparatus in fault is immediately removed from the system without affecting any other part of the system.

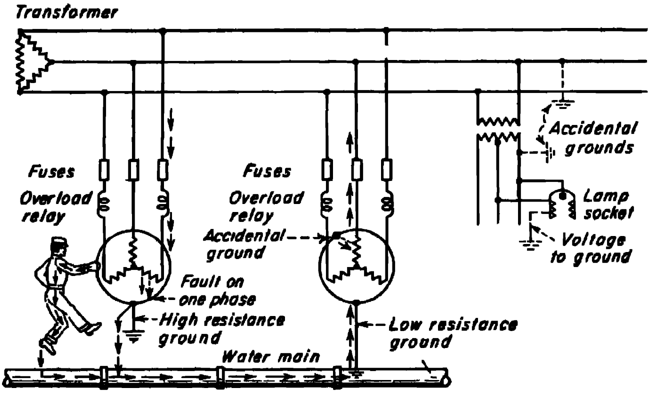


FIG. 13-27*a*. Ground characteristics of electrical systems - possible personnel hazards due to ground-fault current.

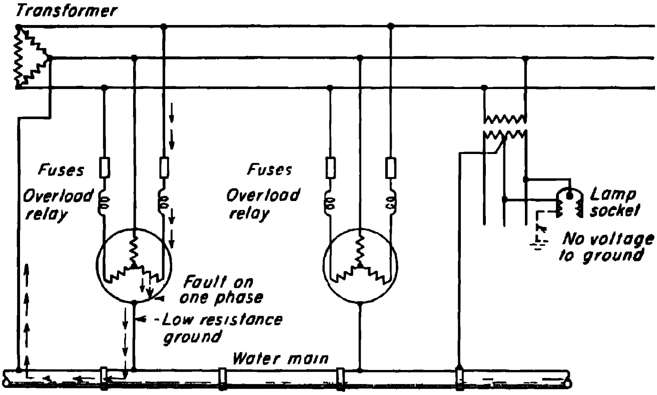


FIG. 13-27*b*. Ground characteristics of electrical systems - personnel hazard removed with proper equipment and circuit grounds.

2. The maximum voltage that can exist on the system is the nominal phase-to-phase voltage.

3. Many of the existing low-voltage industrial systems are equipped only with two overload relays on the feeders. In the grounded-phase system only two overload relays on trip coils are necessary for adequate protection.

In the 208/120- and the 460/265-volt three-phase four-wire systems, the neutrals are usually connected solidly to ground since the maximum possible voltage which can appear across any portion of the ground path is low (265 volts in the case of the 460-volt system), and the impedance of the ground path is nearly always sufficient to limit the current to a safe value.

Higher Voltage Systems (2,400 Volts and Above). In the higher voltage systems it is necessary to limit the ground fault current to the lowest value which is satisfactory for relay operation. The reasons for limiting the ground fault current are

1. To reduce the burning and melting effects of the fault current on equipment and wiring.
2. To reduce the momentary line-voltage disturbances to at least 2½ to 3 times the normal line-to-ground voltage. This value is below the test level for apparatus with the result that the number of equipment failures attributed to surge voltages will be decreased.

Depending upon the kva rating and voltage rating and the inherent impedance of the system, some transformers can have their neutral solidly grounded. The approximate maximum size of transformer banks that can be solidly grounded without exceeding 2,500 amp is shown in Fig. 13-28.

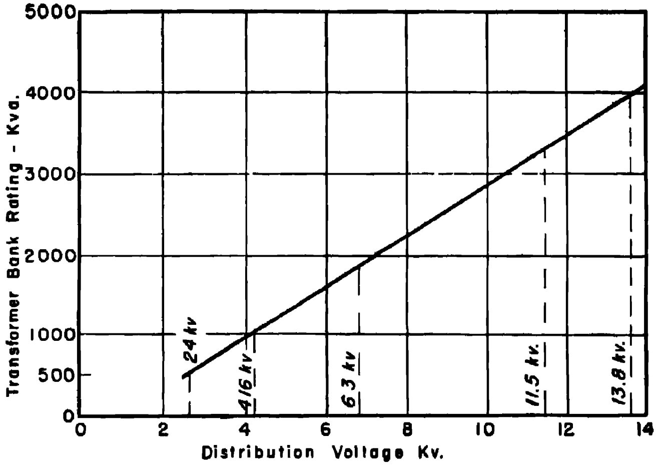


Fig. 13-28. Approximate maximum size of transformer banks that can be solidly grounded without exceeding 2,500-amp ground-fault current.

For larger size banks and practically all generators, it is necessary to limit the ground fault current. The methods used to limit the current are

1. Resistance grounding
2. Reactance grounding
3. Combination of both resistance and reactance
4. Ground fault neutralizer

The common connections for the neutral grounding methods are shown in Fig. 13-29.

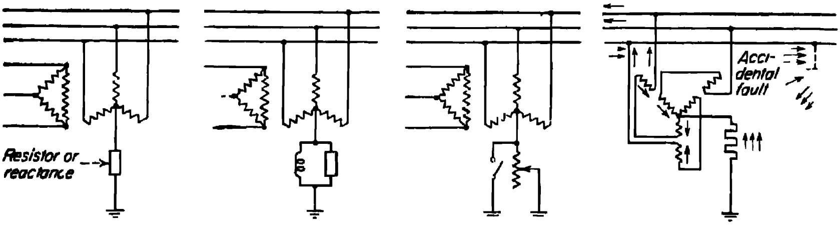
Resistor grounding is best suited for a large percentage of industrial systems as it provides a practical means of limiting ground fault currents to desired levels. Resistors are rated in volts equal to the line-to-neutral voltage and in amperes equal to the current which flows when this voltage is applied to the resistor. For example, the normal line-to-neutral voltage in a 13.8-kv system is 8,000 volts. If a 4-ohm resistor is used, the ground fault current will be approximately 2,000 amp. A range of 1,000- to 3,000-amp ground fault current is usually required for satisfactory relay operation.

Reactance grounding has application advantages in certain types of industrial systems where high values of ground fault current make the system suitable for a more effective lightning-arrester application. The solidly grounded neutral system is a form of reactance grounding. For most reactance-grounded systems, the ground fault current must exceed 25 per cent of the system three-phase fault current in order to avoid serious transient overvoltages.

Combination of resistance and reactance grounding is less expensive for outdoor applications than grounding resistors since the latter, if applied outdoors, should preferably be made of stainless steel. Outdoor grounding reactors are less expensive than stainless-steel grounding resistors of the same rating. A properly selected parallel combination of outdoor reactor and resistor will combine the advantages of each.

Ground fault neutralizer grounding is most commonly used in the higher voltage systems (13.8 kv and above) using overhead lines where there are frequent faults in the line such as insulator flashovers. The use of a neutralizer results in practically no ground current being available to maintain the arc, and hence, the fault is self-clearing.

Where no suitable wye-connected transformer bank or generator is available for providing a neutral connection, a grounding transformer is required. The type of transformer most commonly used is a three-phase zigzag unit with no secondary winding. It is intended solely for the purpose of establishing a neutral ground connection. Each wye of such a transformer has two separate windings which are interconnected between phases. In case one of the lines to which the transformer is connected becomes grounded, the ground fault current divides equally among all the windings. The ground current is limited by impedance of the two sets of coils which are on the same core. The resistor impedance should be at least twice the value of the transformer reactance. The impedance of the transformer to three-phase currents should be such that, when there is no fault on the system, only a small magnetizing current flows in the transformer windings.



Grounding system by resistor connection to neutral lead of Y-Δ transformer

Grounding system by combination resistor and reactance connection to neutral lead of Y-Δ transformer

Grounding system by means of ground fault neutralizer to neutral lead of Y-Δ transformer

Grounding system by means of a zig-zag neutral grounding transformer

FIG. 13-29. Common methods of grounding to limit ground-fault current.

A wye-delta connected transformer may also be used as a grounding transformer, but a zigzag type of transformer is usually less expensive than a comparable two-winding transformer.

It is recommended that some form of grounding be used for all industrial-plant systems. However, the expense involved in conditioning existing ungrounded 460-575-, or 2,400-volt systems for grounding may prove so excessive that grounding means cannot be warranted. As the plant system expands and loads increase, it becomes more difficult to arrange interruptions to the circuits in order to detect and clear ungrounded systems of faults.

Important Factors in Selecting Wires and Cables for Industrial Plants

Industry has many applications requiring specific types of wires and cables for efficient plant operation. Cables are required to stand up where exposed to attacks from oils, acids, or alkalies, extreme corrosive conditions, high ambient temperatures and other severe operating conditions. As the requirements of industry grow more complicated, a multiplicity of types of cables has been made available. New basic insulation materials were developed and adopted wherever it was believed they would do a better job.

Many of the types of cables are needed only for very special applications. Careful selection of a few types for each plant having a wide variety of applications can keep the number of different cables required small. It is the purpose of the following to analyze typical application problems which are encountered and outline suitable cables for the job.

Many factors jointly influence the proper selection of industrial wires and cables The primary ones are listed below:

1. Operating conditions
   a. Voltage and frequency
   b. Installation method—conduits, ducts, free air, air or earth ambient, conduit grouping, etc.
   c. Special conditions—corrosive problems (water and chemicals), high temperature, severe exposures, etc.
2. Insulation of conductors (see Table 13-14 and Table 13-15)

Table 13-14. Insulated Cable Specifications

- *A* Conductors
  - 1 Material copper aluminum
  - 2 Type of conductors solid, stranded, extra flexible
  - 3 Number and size of conductors
- *B* Insulation (see Table 13-15)
  - 1 Rubber, code grade, types R, RW, RU, RH
  - 2 Thermoplastic, code grades T, TW, TA
  - 3 Varnished cambric code grades V, ACV
  - 4 Asbestos and varnished cambric code grades AVA, AVB, AVL
  - 5 Asbestos code grades A, AA AI AIA
  - 6 Slow burning, code grades SB SWB
  - 7 Weatherproof, code grade WP
  - 8 Glyptal cloth, resistant to oils and alkalies
  - 9 Flamenol (code grade I) Severe applications for 600 volts when exposed to acids, alkalies, and oils (Special applications from plug-in-type bus to machines, control cables, installed aerially in conduit or ducts or buried directly in ground)
- *C* Finishes
  - 1 Metallic
    - *a* Lead or alloy-lead sheath (L)
    - *b* Armor—interlocked (I), flat-band (I), round-wire (W), basket weave
  - 2 Nonmetallic
    - *a* Braid flame and moisture resisting (weatherproof), seine twine (B), glyptal treated
    - *b* Rubber-hose jacket for extra weatherproofing (RHJ)
    - *c* Tellurium compound jacket for unusual rough usage such as encountered in portable welders, mining machines, electric shovels, etc
    - *d.* Neoprene compound for resisting attacks by oil, gasoline, acids, and alkalies
    - *e.* Flamenol—same as neoprene but it is not so flexible as neoprene (I I)
    - *f* Treated jute (sometimes asphalt finish)
    - *g* Semiconducting corona-resisting type for shielding of cables up to 15 000 volts
    - *h.* Treated—duct tape for use over lead to give extra protection against corrosion, electrolysis, and possible damage

3. Number and type of conductors. Single-conductor cables, three conductors in a conduit, are most frequently used for industrial-plant distributions as making of taps and splices is simplified.

4 Relative importance of circuits Reliability is generally required in the following order: main supply circuits, main feeders to local load centers, subfeeders, and branch circuits. Relative importance of circuits in each class is dependent on load; and frequently, premium-type wires and cables are selected for certain operating conditions and types of exposure.

5 Size of conductor to be selected so that the current will not heat it to a temperature in excess of the rating of the insulation Make necessary corrections for grouping (in air or in conduit) and for the ambient where the ambient is not standard (20°C earth or 30°C air). Where National Electrical Code regulations apply, check the conductor size against National Electrical Code (1947) Tables 1 and 2

6 Size of conductor to be checked to determine that it will carry the electrical energy to the point where it is to be used without excessive voltage drop. If voltage regulation is excessive, select the conductor size on the basis of voltage drop.

7 On certain feeders it may be desirable to consider and check that the cost of energy lost in the distribution circuits will not be excessive.

A conductor may satisfy one of the three latter items and yet not satisfy the

Table 13-15. Properties and Usage of Standard Conductors for Interior Wiring

Trade name	Type letter	Maximum operating temperature	Insulation	Conductor sizes	Voltage range	Outer covering	Application
Code	R	60°C (140°F)	Code-grade rubber	14–2 000 M cir mils	0–5 000	Moisture-resistant, flame-retardant, Fibrous covering	General use, dry locations
Moisture resistant	RW	60°C (140°F)	Moisture-resistant rubber	14–2,000 M cir mils	0–5,000	Moisture-resistant, flame-retardant, Fibrous covering	General use, wet locations
Heat resistant	RH	75°C (167°F)	Heat resistant grade rubber	14–2 000 M cir mils	0–5 000	Moisture-resistant, flame-retardant, Fibrous covering	General use, dry locations
Latex	RU	60°C (140°F)	90% unmilled grainless rubber	14–6	0–600	Moisture-resistant, flame retardant, Fibrous covering	General use, dry locations
Thermoplastic	T	60°C (140°F)	Flame-retardant thermoplastic compound	14–2 000 M cir mils	0–600	None	General use dry, locations
Moisture-resistant thermoplastic	TW	60°C (140°F)	Flame-retardant moisture resistant thermoplastic	14–2 000 M cir mils	0–600	None	General use, wet locations
Thermoplastic and asbestos	TA		Thermoplastic and asbestos	14–4/0	0–600	Flame-retardant cotton braid	
Varnished cambric	V	85°C (185°F)	Varnished cambric	14–2 000 M cir mils	0–28 000	Plastic covering or lead sheath	Dry locations only unless lead-sheathed
Asbestos and varnished cambric	AVA	110°C (230°F)	Impregnated asbestos and varnished cambric	14–2 000 M cir mils	0–5 000	Asbestos braid	General use, dry locations
Asbestos and varnished cambric	AVL	110°C (230°F)	Impregnated asbestos and varnished cambric	14–2 000 M cir mils	0–5,000	Lead sheath	General use, wet locations
Asbestos and varnished cambric	AVB	90°C (194°F)	Impregnated asbestos and varnished cambric	14–2 000 M cir mils	0–5,000	Flame retardant, cotton braid	General use, dry locations
Asbestos	A	200°C (392°F)	Felted asbestos	14–8	0–300	None	Dry locations only, not for use in general conduit installations; in raceways only as leads to or within operations
Asbestos	AA	200°C (392°F)	Felted asbestos	14–4/0	0–600	Asbestos braid	
Asbestos	AI	125°C (257°F)	Impregnated felted asbestos	14–8	0–300	None	
Asbestos	AIA	125°C (257°F)	Impregnated felted asbestos	14–1 000 M cir mils	0–600	Asbestos braid	
Paper		85°C (185°F)	Paper		All voltage	Lead sheath	Service entrance and special use
Slow burning	SB	90°C (194°F)	Three braids impregnated fire resistant cotton thread	14–2 000 M cir mils	0–600	Outer covering finished smooth and hard	Dry locations only
Slow burning weatherproof	SWB	90°C (194°F)	Two layers impregnated cotton thread	14–2 000 M cir mils	0–600	Outer fire-retardant covering	Dry locations only
Weatherproof	WP	80°C (176°F)	At least three layers impregnated cotton braid or equivalent			None	May not be used
Ozone resistant		Manufacturer's recommendations	Ozone resistant rubber		0–28 000	Braid or lead	General use as type R, RW or RH

remaining two. A conductor may have ample cross-sectional area to carry a given current, but the length of the circuit to the load center may be such that the allowable voltage drop or the energy loss or energy dissipated in overcoming the line resistance will be excessive. Manufacturers' data and National Electrical Code give tables for current-carrying capacities of the various types of wires and cables.

With reference to voltage drop for low- and medium-voltage a-c circuits, it is necessary to take into account the resistance, reactance, and power factor of the circuit and use the following formula for calculation:

$$E = \sqrt{(E^1 \cos \phi + IR)^2 + (E^1 \sin \phi + IX)^2} - E^1$$

where E = voltage drop to neutral, volts per total length of each conductor

E^1 = receiver, end voltage to neutral, volts

I = load current, amp

R = a-c resistance, ohms per total length of each conductor (For d-c resistance of copper conductors and the a-c/d-c resistance ratio refer to tables Sec. 15-126 and 15-128 "Standard Handbook for Electrical Engineers" 1941 edition.)

X = reactance, ohms per total length of each conductor to neutral (See table, Sec. 15-126, "Standard Handbook for Electrical Engineers," 1941 edition.)

$\cos \phi$ = power factor of load

(For direct-reading voltage, drop calculating charts for three-phase power wiring in iron conduit; for voltages of 460, 2,300, and 4,160, see General Electric Company Wire and Cable Catalog, Sec. 1200.)

When consideration is given to initial cost, it will always be found cheaper to run a few large conductors rather than a number of smaller ones. In a-c wiring, no conductors larger than 750,000 cir mils should be used in order to avoid otherwise detrimental skin effect. For economic reasons, parallel conductors for circuits are preferred for circuits of 300 to 400 amp or more.

Table 13-14 shows an analysis of types of cables for general use in industrial plants. For most plants only a limited number of types are required to do the job. As noted, the same type of cable may be used satisfactorily for a wide variety of applications. For example, a cable with a neoprene outer covering or sheath resists flame, acids, alkalies, oils, heat, sunlight, and abrasives. It is good for long service life, for installation aerially, direct earth burial, or in ducts or conduits.

Comparison of Purchased and Locally Generated Power[4]

The estimated total annual electric power generated in the United States and imported in 1948 is 300 billion kwhr. On the average, industries consume about 65 per cent of the total power generated in the United States by utilities and industries combined (after deducting the power-plant and transmission and distribution losses). Of this amount, industrial power plants are used to generate about one-third, and the balance is purchased from utilities The majority of industrial customers, particularly those having small power requirements, purchase their power from public utilities. This is the logical result when the rates are favorable

Conditions may exist, however, which make it more economical for some industrial plants to generate a part or all of their power locally in spite of reasonable and justifiable rates for purchased power. Important factors in determining the desirability of industrial-plant generation are

1 Proximity to existing utility power lines
2. Magnitude of power requirements
3 Available by-product fuel or heat
4. An adequate supply of cheap fuel or water supply
5. Large demand for low-pressure steam for processing and heating

The factors listed above will be discussed in that order. They may exist singly or in combination.

1 The plant location may be so remote from satisfactory public-utility lines that the power rates may be so high and the service so unreliable as to warrant local generation

2 Conditions may exist which make it possible and economical for the plant to generate a part of its power locally in spite of reasonable and justifiable rates for purchased power The amount of power which it is economical to generate will depend upon the rates obtainable and other factors discussed below.

Table 13-16. Wires and Cables for Industrial-plant Layout

Maximum voltage class	Installation method	Installation conditions	Maximum allowable copper temp deg C	Recommended type of cable
600	Conduit	Dry	60	Code type R RU and T
	Conduit	Wet	60	Code type RW and TW
	Conduit	Dry	75	Code type RH
	Conduit or ducts or direct burial	Wet or dry	75	Type RH or RW with neoprene jacket or lead covered
	No conduit	Dry	60	Armored cable code type R multiconductor
	No conduit	Dry	85	Interlocked armor varnished cambric multiconductor type ACV
	Direct burial	Wet	75	Type RH with neoprene jacket or lead covered
	Conduit	Dry exposed to high temperatures	85 110	Types V and AVA
	In open air	Wet or dry exposed to sunlight	75	Type RH or RW with neoprene jacket (preassembled aerial cable)
	In conduit or open air	Oil and heat resistance		Glyptal cloth with water proof braid
	In conduit or open air	Oil acid alkali moisture resistance		Flamenol type I
2 000–5 000	Conduit or duct system	Wet or dry	75	Type 50 RH or 50 RW with neoprene jacket or lead covered
	Conduit or duct system	Wet or dry	85	Type 50 VL
	No conduit	Dry	85	Interlocked armor varnished cambric multiconductor type ACV
	Direct burial	Wet	75	Type 50 RH or 50 RW with neoprene jacket
	In open air	Wet or dry exposed to sunlight and corrosive fumes	75	Type 50 RH or 50 RW with neoprene jacket (preassembled aerial cable)
5 000–15 000	Conduit or duct system	Wet or dry	75	Ozone resistant rubber with shielding tape
	Conduit or duct system	Wet or dry not subject to electrolytic action	85	Varnished cambric lead covered
	In open air	Wet or dry exposed to sunlight and corrosive fumes	75	Ozone resistant rubber with shielding tape (preassembled aerial cabl

3 Many industrial plants, such as cement plants, smelters, paper mills, sawmills blast furnaces, coke ovens, and refineries have available by-product heat or fuel as result of the type of process carried on The forms of energy available vary in the different industries hot air in cement mills and smelters chemical by-products which may be burned in paper mills bark, sawdust and waste lumber in sawmills, and combustible gas from blast furnaces, coke ovens, and refineries

In plants using by-products fuel or heat, there is no fuel cost other than handling

4. A supply of easily utilized water power or of relatively cheap fuel, such as is present in the form of oil and gas in the oil fields, presents a problem very similar to the previous one. In some locations, especially in certain sections of Canada, there is an abundance of hydroelectric power available. With a more or less limited industrial use for electrical power, the surplus is often used in electric boilers to produce process steam.

5. Large demands for low-pressure steam for processing and heating constitute another factor. Steam is of essential value in some industries. Amount, pressure, and temperature of steam used vary widely even among the same processes carried on by different manufacturers. In general, so far as the low-temperature heating operations are concerned, steam is the most economical medium to employ. In plants that require process steam in large and fairly constant quantities, many opportunities exist for the development of inexpensive power services.

Through the use of high-pressure boilers and pressure-reducing prime movers, the process steam required may be obtained, either as bled steam or high-pressure exhaust steam, while at the same time considerable quantities of electric energy can be produced very cheaply—sometimes for as low as 2 mills per kilowatt. A most interesting problem develops in the process industry by permitting the extraction of steam for electric-power generation from the steam generated by high-pressure superheating boilers before the steam is turned over at process pressure to the chemical processes. In the past, there was a tendency to treat the problems of an industrial power plant the same as those of a public-utility plant. Both have much in common, but because the industrial power plant must be necessarily considerably smaller than a public-utility plant and, therefore, less efficient, it is amiss to assume that purchased power will be cheaper. This might be true if the supply of power did not have connected with it the supply of heat. The supply of steam and electric power to a process does not constitute two problems, but a combined single problem. The efficiencies of a public-utility power plant and an industrial power plant cannot be compared as a guide in determining which source will be most economical. The true criterion of a power plant's performance is the amount of heat put to useful work, or, in other words, the amount of preventable waste that can and should be avoided.

It is the primary business of public utilities to sell power and services. It behooves them to render the best possible service without interruptions at the lowest possible costs within economic reason. The problems of an industrial power plant of the kind here being discussed are peculiar. The "by-product" power is incidental to furnishing heat. "By-product" power from pressure differences is in no way related to power from waste fuel, from the waste heat of process combustion, or from waste exhaust steam condensed through turbines. It is limited to that power which an industry can produce by generating steam at pressures within present ranges of practice and then passing this steam through a turbine or engine and exhausting it at a pressure high enough to supply a definite process requirement for heat deliverable as steam The power plant of a process industry is more of a heating plant than a power plant; the problems are heat-engineering problems rather than power problems. It is necessary only to determine what degree of perfection is warranted to secure the best possible economy for one industrial plant, rather than for a group of industries as the central station must necessarily supply. The balancing of the quality of power service against its cost is of fundamental importance in the process industries. These are the greatest differences between an industrial power plant and a utility power plant, the single purpose of which is to produce power. A misunderstanding of the heat-power problem is a barrier to potential savings.

Wherein does the economy of "by-product" power generation lie? It is not difficult to prove that, in general, "by-product" is the cheapest power available to an industrial plant, provided operating conditions requiring electric power and steam heat are favorably balanced. In the typical steam plant there are three general steps:

1. Heat stored in fuel is delivered to a plant.
2. The fuel is burned, and the stored heat is liberated and transferred to water and steam in the boilers.
3. The accumulation of heat in the boilers is transferred into electric power by a

turbogenerator or an engine-generator set, or it is carried by the water and steam to the various processes.

If there were no waste of heat and the boilers, turbines, and generators were 100 per cent efficient, a quantity of fuel containing 3412 Btu could generate 1 kwhr. It is impossible, however, to transform all of a given amount of heat supplied to a working substance into work. Modern boiler units operate at efficiencies of 80 to 85 per cent, the latter figure not being unusual; turbines operate at 95 to 98 per cent efficiency; and generators, 90 to 96 per cent. With these efficiencies, it is possible to produce 1 kwhr of by-product power at a fuel-consumption rate equivalent to 4,300 to 5,000 Btu per kwhr or 0.3 to 0.4 lb of average coal. The heat that cannot be transformed into work, and that which must be rejected by the turbine can be used in the processes. In the central-station plant, the heat exhausted from the turbine is practically useless because of its low temperature and lack of pressure; such a condensing plant, even when operating at 1,400 lb steam pressure and employing both the generative and reheat cycles must have a fuel consumption equivalent to 11,000 to 14,000 Btu per kwhr, or 0.8 to 1.0 lb of average coal.

In the by-product power plant, the heat may be extracted from a noncondensing unit at any desired pressure and temperature (this, of course, depends upon the initial pressure and temperature) and put to practical uses in the process. Temperature difference is necessary for heat transfer, and pressure difference is necessary for steam velocity in the unloading stages of the turbine to the processes. For this reason, considerably less power will be produced in the by-product plant than in the central station for the same fuel consumption, but the proportion of heat usefully applied in by-product generation to central-station generation is about in the ratio of 3:1 By-product energy need be charged only for the heat actually used in passing through the turbine. This is not true for a central station because in a condensing steam turbine the heat of vaporization of the steam at the exhaust pressure is an unusable end product which must be carried away by the condenser circulating water.

In the matter of selecting turbines for a plant, no general statement can be made because the conditions of every problem are different. However, in general, the following is correct:

1. Where the electrical load is predominant, the turbine should be a condensing-extraction type.
2. Where the steam load is predominant, the turbine should be a noncondensing-extraction type.
3. Where the steam demand balances the electrical

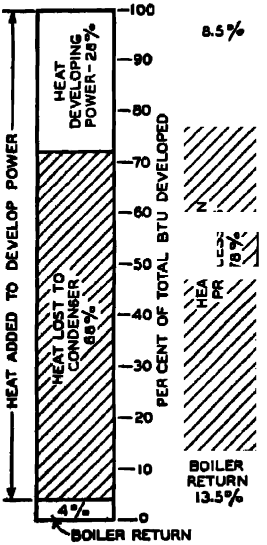


Fig. 13-30. Proportion of heat used in condenser-turbine (left) and "by-product" (right) power generation Throttle steam for both condensing-turbine and by-product generation is at 400 lb gauge pressure and 200°F superheat. Condenser turbine exhausts to a 29-in. vacuum. "By-product" generation exhausts at 80 lb back pressure and 60°F superheat. If the heat content of one pound of throttle steam is 1330 Btu, the mechanical efficiency 97 per cent, the electrical efficiency 96 per cent, and all items are the same for each turbine, then the following is a comparison of the two systems:

Condensing plant:

$$\text{Lb steam per kwhr at turbine throttle} = \frac{3{,}412}{0.28 \times 0.97 \times 0.96 \times 1{,}330} = 9.8$$

$$\text{Btu per kwhr at turbine throttle} = 9.8[1330(0.28 + 0.68)] = 12{,}800$$

"By-product" plant:

$$\text{Lb steam per kwhr at turbine throttle} = \frac{3{,}412}{0.085 \times 0.97 \times 0.96 \times 1{,}330} = 32.8$$

$$\text{Btu per kwhr at turbine throttle} = 32.8 \times 0.084 \times 1330 = 3060$$

demand, the turbine should be a straight noncondensing type. This type is seldom used because the balanced conditions are seldom obtained; however, the noncondensing turbine may be used in conjunction with a condensing turbine, and the steam may then be bled off the noncondensing turbine at process pressure. If the quantity of steam bled off is not all required for the process, it may be passed through the lower stages of the condensing turbine and used to produce power.

Figure 13-30 shows the magnitude of the heat quantities in the steam cycle of a condensing plant and of a by-product plant. As shown in the figure, a larger quantity of steam is required to develop 1 kwhr with by-product generation, but the heat energy (Btu) chargeable to power is considerably less than that required by condensing-turbine generation. The percentages given are for a specific problem but indicate relative proportions of uses of the generated steam.

Figure 13-31 gives the power available from steam-process reductions for process use at various turbine-inlet pressures and various extraction pressures. This chart is valuable for making preliminary estimates of power possibilities.

There are many schemes of connecting equipment and apparatus for the generation of by-product power from steam used later for process and heating. Figure 13-32 shows an arrangement of a turbine expanding some steam from boiler pressure to a process bleed point and the remainder passing through the lower stages of the turbine to the condenser. Figure 13-33 shows an arrangement using high-pressure boilers which supply two high-pressure turbines; one high-pressure turbine exhausts to a low-pressure condensing turbine and the second to an evaporator supplying the process steam.

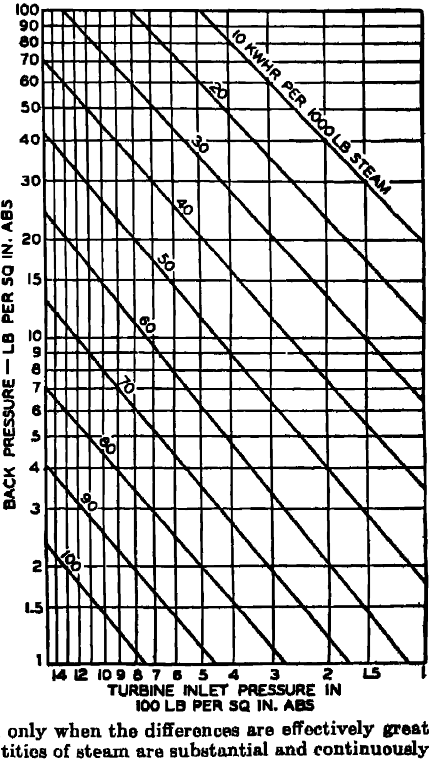


Fig. 13-31. Power available from steam-process reductions for process use. Graph is laid out for 100 per cent turbine efficiency so that conversion may be direct for substantial quantities. Actual yield in power may be taken at 70 per cent of that for the ranges plotted. Pressures are shown in pounds absolute (14.7 lb higher than plus gauge pressures). Following vertically from any generated pressure and horizontally from any exhaust or process pressure, the axis will intersect at a value indicating the kilowatthours that would be produced by passing 1,000 lb of steam from the generated pressure to the pressure used in the process. These values are plotted for a fixed superheat of 150°F above the temperature at the generated pressure. Superheat variation has a very marked effect on the available power. The graph, however, is useful for preliminary estimates of power possibilities, as the curves permit approximation from figures on steam-pressure drop.

Example: Dissimilarity of conditions will produce identical yields in power per unit weight of steam passed. On the curve for 40 kwhr per 1,000 lb, note that this power will be produced by from 375 lb to absolute, and also the same production is possible from passing from 1,200 to 55 lb. "By-product" power from pressure differences is practical only when the differences are effectively great as shown by the curves, and when the quantities of steam are substantial and continuously used.

Commercial development of high-pressure steam (1,000 lb and above) has opened a field of unlimited possibilities in the direct application to chemical processes, but possibly greater possibilities in the field of power generation from steam later bled off or exhausted for process use at moderate pressures.

An approach to the application of steam for both by-product generation and process use would be for the chemical engineers, who are familiar with the process requirements, to set the process pressure or pressures as low as possible for economical production and compile daily load curves for process-steam and power demand. In the design of plants to supply process steam and power simultaneously, this factor of relative amounts of process heat and power required is one of considerable economic influence. For each particular plant, this ratio of steam requirements to power requirements has certain determinable limits. The power engineer, by proper consideration of these limitations with other economic conditions, could assume an initial pressure and plot from the process-demand curve a curve of the by-product power available. The latter curve then could be compared with the power requirements, and if found insufficient at all hours during the day, the boiler pressure may be increased, or the temperature of the steam generated may be increased without change in pressure, or both the temperature and pressure of the turbine-thrott steam may be changed, until the electric power made available by by-product generation would balance that needed. The most effective way of all to increase the power output is to decrease the exhaust pressure.

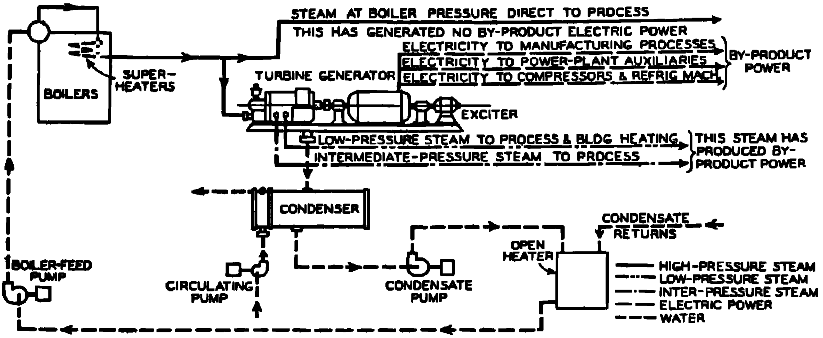

STEAM AT BOILER PRESSURE DIRECT TO PROCESS
THIS HAS GENERATED NO BY-PRODUCT ELECTRIC POWER
ELECTRICITY TO MANUFACTURING PROCESSES
ELECTRICITY TO POWER-PLANT AUXILIARIES
ELECTRICITY TO COMPRESSORS & REFRIG MACH
BY-PRODUCT POWER
TURBINE GENERATOR
SUPER-HEATERS
BOILERS
EXCITER
LOW-PRESSURE STEAM TO PROCESS & BLDG HEATING
INTERMEDIATE-PRESSURE STEAM TO PROCESS
THIS STEAM HAS PRODUCED BY-PRODUCT POWER
CONDENSER
CONDENSATE RETURNS
OPEN HEATER
BOILER-FEED PUMP
CIRCULATING PUMP
CONDENSATE PUMP
HIGH-PRESSURE STEAM
LOW-PRESSURE STEAM
INTER-PRESSURE STEAM
ELECTRIC POWER
WATER

FIG. 13-32. Schematic arrangement of "by-product" power supply.

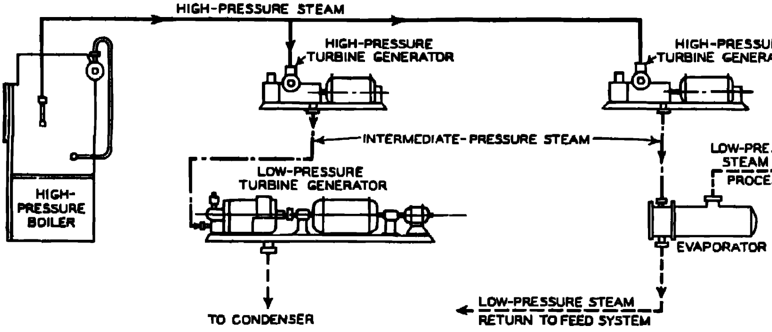

HIGH-PRESSURE STEAM
HIGH-PRESSURE TURBINE GENERATOR
HIGH-PRESSURE TURBINE GENERATOR
INTERMEDIATE-PRESSURE STEAM
LOW-PRESSURE TURBINE GENERATOR
LOW-PRESSURE STEAM TO PROCESS
HIGH-PRESSURE BOILER
EVAPORATOR
TO CONDENSER
LOW-PRESSURE STEAM RETURN TO FEED SYSTEM

FIG. 13-33. Schematic arrangement of "by-product" power supply using high-pressu boilers and turbines.

In actual practice, it may be more economical not to carry the pressure to a point where all the power required can be generated as a by-product of process steam. The propinquity of a central station and a connection with a utility system is a good thing as a stand-by and to take peak loads in excess of power generated. Unfortunately, utility companies by their demand charges under stand-by conditions tend to make such connections unprofitable.

Power Factor

Most industrial-plant power is supplied by a-c systems. Entering into practically all considerations of a-c machinery and circuits is the term power factor, the factor by which volt-amperes must be multiplied in order to find actual power in watts. The over-all power factor of industrial plants is almost invariably lagging. This is principally due to the fact that the major part of industrial equipment operates at lagging power factor. The power factor of typical a-c loads is given in Table 13-17.

In the case of induction motors, the power factor not only varies with the horsepower and synchronous speeds of the motors but also with the loads on the motors. How the power factor of a typical 20-hp 900-rpm induction motor varies with load is shown in Fig. 13-34.

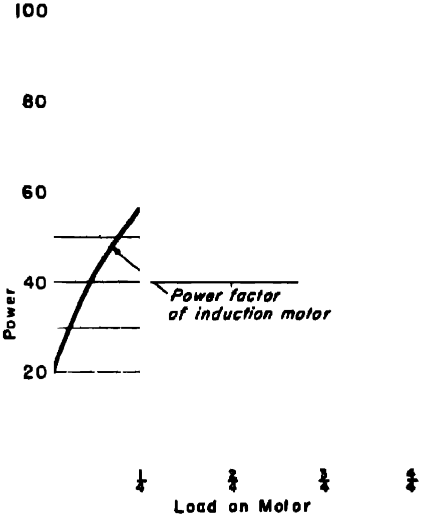


Fig. 13-34. Power factor of a typical 20-hp, 900-rpm induction motor.

Disadvantages of Low Power Factor. Low power factor is detrimental for four important reasons:

1. It prevents the full utilization of the generation and distribution system. The ratings of generators, transformers, transmission lines, feeders, and switches are limited by their current-carrying capacity, and the maximum power (kilowatts) which they may carry is proportional to the power factor. Thus capital investment per kilowatt of load increases as the power factor decreases.

2. It makes each kilowatt of power carry a higher burden of line losses. The resistance losses, being proportional to the square of the current (I^2R), vary inversely as the square of the power factor. Losses are therefore equivalent to an increase in the kilowatt load and increase the cost of delivering power to the utilization equipment.

3. It may cause an excessive voltage drop at the power-utilization equipment and may possibly affect production by reducing starting torque in motors, causing subnormal lighting, and slowing up the operation of heaters.

4. It may increase the purchased-power cost because of power-factor penalties. The detrimental effects of low power factor make it necessary for utilities to incorporate power-factor clauses in rate schedules. The power customer benefits by substantial power-bill reductions when he keeps the power factor of his plant high.

Table 13-18 shows a tabulation of the reactive factor, the ratio of reactive factor to the power factor, and the conductor requirements in per cent for various power factors. Table 13-19 shows a tabulation of the corrective kva requirements for improvement of power factor.

Methods of Improving Power Factor. High power factor is achieved by making the ratio between the kilowatts and kva of the a-c circuits as nearly 1:1 as possible. Several means are available for improving the power factor of an industrial plant. The following are the most important:

1. Minimize reactive kva component. This is a preventive method, such as avoiding oversize or low-speed induction motors and keeping induction motors as fully loaded as possible.

Table 13-17. Power Factor of Typical A-c Loads

Unity (or near unity) power factor		Lagging power factor		Leading power factor	
Load	Approximate power factor	Load	Approximate power factor	Load	Approximate power factor
Incandescent lamps (Power factor of lamp circuits operating off stepdown transformers will be somewhat below unity)	1.0	Induction motors (at rated load)		Synchronous motors (Leading power factor depending on the rated leading power factor for which they are built)	0.9 0.8, 0.7 0.6 etc
Fluorescent lamps (with built in capacitor)	0.95–0.97	Split-phase below 1 hp	0.55–0.75	Synchronous condensers	Nearly zero leading power factor (Output practically all leading reactive kva)
Resistor heating apparatus	1.0	Split phase 1 to 10 hp	0.75–0.85	Capacitors	Zero leading power factor (Output practically all leading reactive kva)
Synchronous motors (Operate at leading power factor at part loads also built for leading power factor operation	1.0	Polyphase squirrel cage			
Rotary converters	1.0	High speed 1 to 10 hp	0.75–0.90		
		High speed 10 hp and larger	0.85–0.92		
		Low speed	0.70–0.85		
		Wound rotor	0.80–0.90		
		Groups of induction motors	0.50–0.85		
		Welders motor generator type	0.50–0.60		
		Transformer type	0.50–0.70		
		Arc furnaces	0.80–0.90		
		Induction furnaces	0.60–0.70		

2. Eliminate lagging reactive kva. Many motor drives permit the use of synchronous motors which require no reactive kva from the line.

3. Offset lagging reactive kva with leading reactive kva, by using (a) leading-power-factor synchronous motors, (b) capacitors, and (c) synchronous condensers.

Figure 13-35 is a diagram showing the typical methods of applying power-factor-improvement equipment. Not all the equipment shown or the specific methods will necessarily be used in any one given plant. As shown in the figure and indicated above, the principal types of apparatus used to effect power-factor improvement are (1) induction motors properly selected to avoid oversizing, (2) synchronous motors for medium and large constant-duty drives, (3) capacitor units, and (4) synchronous condensers.

Induction Motors Properly Selected to Avoid Oversizing. The magnetizing kva of induction motors remains almost constant from full load to no load, while the kilowatts vary with the load.

Thus at light loads the motor input is largely reactive kva, and hence power factor is low.

Obviously then, any practical steps that will result in maintaining as near as possible full-load conditions will be beneficial from the power-factor standpoint. Observing these general rules will help:

1. Care in selection of motors for their jobs. For example, use of high-slip motors for punch presses will usually enable use of smaller horsepower rating than where normal-torque motors are applied.

2 Rearrangement of induction motors in the plant, where practical, to secure over-all average higher loading.

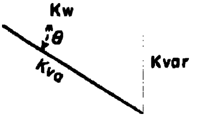
3. Use of wound-rotor induction motor where high starting torque requires oversize squirrel-cage motor.

4. Avoidance of slow-speed induction motors, which have inherently low power factor even at full load in favor of synchronous motors.

The possibility of improving power factor by the proper selection of induction motors should be given greatest consideration at the time of the original installation of these motors. After the motors are installed, it seldom pays to purchase new motors of a different type or rating or to switch motors from one machine to another, except in the larger horsepower sizes, say, over 75 hp. In the case of small motors, the cost of the changes, including changing the foundations, etc., makes the cost per reactive kva gained quite high. Less expensive means, e.g., the use of capacitors, can usually be found for obtaining the same power-factor improvement.

Table 13-18. Power Factor, Reactive Factor, and Variation of Conductor Requirements

Power factor	Reactive factor	Reactive factor, power factor	Conductor requirements, per cent
100	0	0	100
95	0.312	0.329	111
90	0.436	0.482	124
85	0.527	0.620	135
80	0.600	0.750	156
75	0.661	0.882	178
70	0.714	1.020	204
65	0.760	1.170	237
60	0.800	1.333	278
55	0.835	1.519	330
50	0.886	1.732	400



Trigometric relations
Cos θ = power factor
Sin θ = reactive factor
Tan θ = reactive factor/power factor

Table 13-19. Corrective Kva Requirements

Original power factor	Final power factor	Per cent corrective kva required
50	55	21.2
55	60	18.6
60	65	16.3
65	70	15.0
70	75	13.8
75	80	13.2
80	85	13.0
85	90	13.6
90	95	15.5
95	100	32.9

Corrective kva in the above amounts would be required to raise the power factor from the original in Col. 1 to the final in Col 2. For example, to raise the power factor of a 1,000-kw load from 75 to 90 per cent would require

1,000 × (0.132 + 0.130 + 0.136) = 398 kva.

Synchronous Motors for Medium and Large Constant-duty Drives. Synchronous motors are standard drives for the larger compressors, pumps, fans, blowers, crushers, grinders, line shafts, motor-generator sets, and for mills and rolls in cement, rubber, metal, mining, pulp and paper, flour milling, and other plants. Wherever there are suitable drives in the plant, synchronous motors may be advantageously used for they perform the double duty of driving the load and improving the power factor at a very low unit cost. There are two types of synchronous motors, unity power factor and leading power factor.

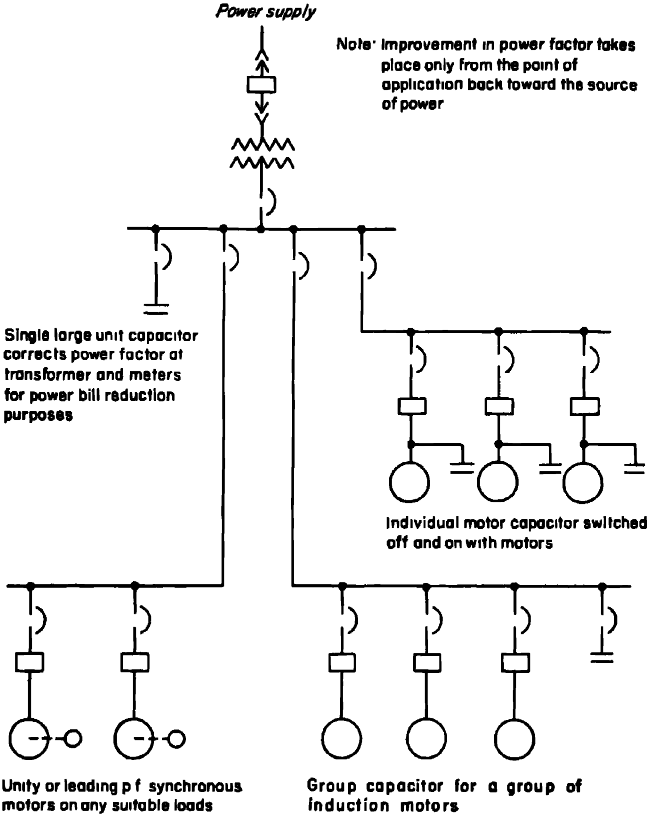

Power supply
Note: Improvement in power factor takes place only from the point of application back toward the source of power
Single large unit capacitor corrects power factor at transformer and meters for power bill reduction purposes
Individual motor capacitor switched off and on with motors
Unity or leading p f synchronous motors on any suitable loads
Group capacitor for a group of induction motors

FIG. 13-35. Principal types of apparatus used to effect power-factor improvement in industrial plants.

A unity-power-factor synchronous motor operating at full load does not actually deliver any leading kva to the system, but its substitution for a lagging-power-factor induction motor or its addition to a load whose power factor is already lagging obviously increases the ratio of kilowatts to reactive kva and thereby improves the power factor.

A leading-power-factor synchronous motor will deliver leading kva proportiona to input power, which depends on the horsepower delivered, and proportional to the ratio of input reactive kva to input kilowatt power, this ratio depending upon the power factor.

The curves of Fig. 13-36 show the approximate leading kva drawn by synchronous motors operating at partial loads with full-load excitation maintained. When operating at partial load, a synchronous motor will deliver more leading kva than when operating at full load. This is a result of the lower reactance drop in the machine at light loads. Note that unity-power-factor synchronous motors operating at partial load may actually draw a leading kva.

The principal advantage of the use of synchronous motors to improve power factor is that, over a large range of horsepowers and speeds, particularly above 100 hp, synchronous motors provide leading kva at a very low cost. For motors of several hundred horsepower, this cost reaches a value as low as $1 per kva gained by the use of a synchronous motor instead of an induction motor. There is no other available equipment that has such a low initial cost per unit leading kva furnished.

The principal disadvantage, or rather limitation, in the use of synchronous motors for power-factor improvement is that it is economical only for a more or less constant-duty constant-speed motor of moderate or large size when such is required for a plant drive. The cost per kva correction gained in the small-horsepower ratings is high—about $20 for each reactive kva gained using a 50-hp 80 per cent power factor 1800-rpm synchronous motor.

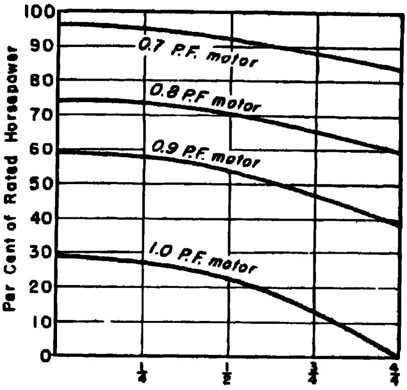


FIG. 13-36. Approximate leading kva in per cent of rated horsepower of motor that will be supplied by synchronous motors operating at various loads under full-load excitation.

Also, in general, synchronous motors are usually economical only for application to new drives. The advantage in low cost per kva gain which is obtained by using synchronous motors is generally lost if the motor is to be substituted in place of an induction motor already installed.

Where there are no suitable drives for synchronous motor applications and when a completed plant requires power-factor correction, it is usually necessary to resort to capacitors.

Capacitor Units. Capacitors are stationary devices, the sole function of which is to deliver leading reactive kva for the magnetizing requirements of induction motors and other induction equipment on the plant system. These units deliver leading kva proportional to the square of the impressed voltage and inversely proportional to their reactance.

Advantages of capacitors include the following: (1) being static devices, they require little attention for maintenance or operation; (2) they can be installed in convenient locations within the plant without disturbance to production at reasonably low installation charges; (3) installation may be made at any time the power-factor conditions warrant and in a wide variety of capacities to fit the power-factor requirements of circuits or individual motors; (4) losses are low—one-sixth to one-fourth of 1 per cent of the kva rating.

The initial cost of capacitor units per leading kva is fairly high as compared with the cost per kva of synchronous motors or synchronous condensers. The use of capacitors for power-factor improvement is most economical under certain conditions. Installed cost of 460-volt capacitors averages about $10 per kva with 230-volt units costing about twice as much. Other voltages and ratings vary from $10 to $15 per kva. The reason for the higher cost of the low-voltage capacitors is that the capacity required, as noted above, varies inversely as the square of the voltage, and at low voltage, about 460 volts, the thickness of insulation can no longer be reduced with the result that it is necessary to increase the size of the unit in order to obtain the capacity for the lower voltage. Ratings of capacitors are proportional to the frequency, and this makes the use of capacitors extremely high for installation on 25-cycle systems.

Synchronous Condensers. A synchronous condenser is similar to a synchronous motor, built primarily, however, to operate without mechanical load and to correct power factor or regulate voltage by drawing a lagging current component from or supplying a leading current component to the power system.

The cost of synchronous condensers is considerably less than that of capacitors, particularly in the larger sizes. The cost varies between $5 per kva (5,000-kva rating) to about $10 per kva (500-kva rating). They are seldom found to be economical in sizes below 500 kva and for this reason are seldom used in industrial plants.

Synchronous condensers are used particularly by power companies, where very large amounts of leading reactive kilovolt-ampere component are required and where regulation of voltage, by changing the condenser excitation, is desirable.

Amount of Investment Justified for Power-factor Improvement. Most purchased power schedules contain power factor clauses resulting in the adjustment of billing demand based on average power factor. The essential items of a typical utility's company rate schedule are as shown in Table 13-20.

Table 13-20

Demand charges		Energy charges	
First 100 kw of BD	$2.15/kw	First 5,000 kwhr	1.6¢/kwhr
Next 100 kw of BD .	1.65/kw	Next 100 hr or BD	0.9¢/kwhr
All over 200 kw of BD	1.40/kw	Next 200 hr of BD	0.7¢/kwhr
		Excess	0.6¢/kwhr

BD = billing demand = (90/pf) × metered demand

The monthly power bill under the above rate may be set in a formula as follows if the demand is over 200 kw and the kwhr use reaches into the 0.6¢ per kwhr rate.

$$\text{Power bill/month} = \$200 + \$1.9\ [90/\text{power factor} \times \text{demand}] + \$0.006 \times \text{kwhr}$$

The factor $1.9 [90/power factor × demand] is the only factor affected by power factor and is the only item necessary to consider for the following example:

Assume we have four 500-kva 4,160/480-volt substations at a plant supplying associated feeders fully loaded for the demand periods and operating with a high ratio of average load per maximum demand at a monthly power factor of 85 per cent. The metered demand would be 1,700 kw and the billing demand, 1,802 kw, at a monthly cost of $3,423.80.

The addition of 495 kva of capacitors would improve the power factor to approximately 95 per cent at an estimated total installed cost of $5,000 for 33, 15-kva 460-volt capacitor units and associated equipment. The metered demand would be the same as before (1,700 kw), but the billing demand would be reduced to 1,611 kw at a monthly cost of $3,160.90, or a saving of $262.90 per month.

Thus, the investment required for the 495 kva in capacitors would pay for itself in 19 to 20 months through savings in the power bill.

In addition, the installation of the capacitors has made available 200 kva extra capacity in transformers and main feeders plus the voltage improvement. This is the reason capacitor units can sometimes be justified on plants generating power locally. In the above example, 200 kva of generator, transformer, and feeder capacity is obtained at a cost of $5,000, or at the rate of $25 per kva.

Such economies as these are not out of the question. It is not economical to improve the power factor much above 95 per cent because of the large amount of capacity in corrective kva required for the gain. In plants that undergo rapidly changing methods and fluctuations in production, the investment for power-factor correction is expected to pay for itself in 2 years. In stable industries, large office buildings, public institutions, etc., which expect to operate in the similar manner for a number of years, an investment would be considered satisfactory if it paid for itself in 4 or 5 years.

Sometimes switching is necessary to keep light-load voltages within acceptable limits. If automatic means of switching is provided this would increase the cost of a capacitor bank $2 or $3 per kva.

The modern capacitor's reliability, relative cost level, and flexibility have resulted in one of the most economic tools that is now available to the industrial electrical engineer. However, it should be noted that most power-factor clauses in utility companies' schedules are based on average power factor. For this reason there is usually a certain point beyond which it does not pay to use capacitors to improve power factor alone. For example, a majority of industrial plants have a high kilowatt load during the main working shift and possibly a load of one-quarter or less during the remainder of the day. If capacitor units were installed to correct for a high power factor at peak load, they would be effective only part of the time (reactive kva meters are provided with a ratchet arrangement which prevents them from running backward in case of leading kva). For this type of plant, it would be necessary to plot a graph of the reactive kva load-time curve to determine the maximum kva of capacitors which would be economical to install. Any further improvement in power factor desired should be obtained by the use of a synchronous motor or possibly a condenser, since its cost per kva is much less and would probably be operated only during high load periods.

In order to obtain the most satisfactory and economical means of power-factor improvement, it is necessary to (1) make a system survey of load and power factor and compare with a log of plant operations; (2) check possibility of reducing demand peaks; (3) rearrange large motors, if practical, so they will operate at or near full load; (4) formulate a decision on the type of equipment and arrangement that will best serve the plant requirements (see Fig. 13-35).

PART 2. INDUSTRIAL-PLANT LIGHTING

Satisfactory industrial-plant lighting can be obtained only through intelligent engineering. Lighting is complicated by a diversity of many possible light sources, a variety of fixtures, and equipment arrangement. Engineering is more needed now than ever before so that advantage may be taken in selecting the best for each application. To provide satisfactory industrial-plant lighting it is necessary not only to have an intimate knowledge of the illumination problems but also of process requirements.

Eyesight conservation is concerned not only with proper lighting to aid human eyes to perform seeing tasks without undue eyestrain and fatigue but also with the care of eyes through medication and eyeglass correction if necessary. Before discussing the actual mechanisms of a satisfactory lighting installation, it is most desirable to review briefly the fundamental principles of the characteristics of light and its action on the eyes. For proper seeing, these two items are of great importance. Some understanding of the influence of lighting upon visibility, the ease of seeing, and the relationship of lighting to ease of seeing are essential factors in judging the results obtained with artificial illumination.

Light, vision, and seeing are commonplace, but the effects of seeing extend far beyond the eyes into the realm of human beings, eventually affecting their behavior and finally their efficiency and welfare. This is not an accident. It is due to the fact that lighting is a part of so much of our daily routine activities.

The importance of light, vision, and seeing cannot be overestimated. The success of most industrial occupations depends upon man's mind aided by his five senses: sight, hearing, touch, taste, and smell. Of these senses, sight is by far the most educative, educated, and useful sense. It is responsible for 75 to 90 per cent of the usefulness of most civilized adults. The industrial worker's efficiency is, therefore, greatly affected by the lighting conditions under which he works. In addition, good seeing conditions have a dual significance to good workmanship since they affect, directly, the particular job being done and, indirectly, the workman's entire attitude toward the job. This results in better health conditions, less labor turnover, fewer accidents, and decreased material spoilage.

Because adequate lighting can contribute much to efficient production, lighting is now accorded a much more important place in production than ever before in our industrial history.

After defining some of the more common illumination terms, the important items that must be considered in planning a lighting layout will be discussed.

Definition of Common Illumination Terms

As in the gradual development of any science or work, a pertinent phraseology has grown with the science of illumination. Certain of these terms will be used in subsequent discussion. Listed below are the more common terms used, together with their definitions.

Absorption. A certain amount of the light which hits an object is absorbed by that object. The amount absorbed depends upon the reflection factor of the surface.

Brightness. The brightness of an object depends upon the intensity of the illumination on it and the percentage of light it reflects, emits, or transmits. Brightness is one of the fundamental factors in seeing. When brightness values are low, seeing becomes difficult and slow. Extremely high brightness or wide variation in brightness in the visual field can cause glare and eye fatigue.

Diffusion. Many surfaces, such as plaster and opal or prismatic glass, break up the incident light from a point source and distribute it more or less over an entire surface. Diffusion tends to enlarge the image of light source and reduces its brightness by breaking up its outline. Glare is, therefore, reduced, and seeing becomes easier.

Fluorescent. Fluorescent lamps are electric-discharge lamps in which the radiant energy from an electric discharge is transferred by suitable material (phosphors) into wave lengths giving higher luminosity

Filament. Filament lamps are light sources that consist of glass bulbs containing a coiled filament electrically maintained at incandescence by heat energy created by the electric current.

Foot-candles. A foot-candle is the measure of illumination on a surface which is 1 ft distant from, and perpendicular to, a source of illumination of 1 candlepower. This is the term used to determine the illumination on a working surface.

Foot-lambert. A foot-lambert is a measure of brightness. Numerically, it is approximately equal to the product of the incident foot-candles and the reflection factor of the surface, the degree of brilliancy of any part of a surface or medium of transmitting light.

Glare. Glare may be defined as brightness out of place. The degree of brilliancy of any part of a surface or light source, when viewed from a designated direction, determines the amount of glare. Glare may be direct or reflected from any surface and is injurious to the eye.

Illumination. Illumination may be termed the effect or result of light.

Incandescent. See *Filament.*

Light. For the purpose of illumination engineering, light is radiant energy evaluated according to its capacity to produce visual sensation.

Lumen. A lumen is the unit of light flux representing a quantity of light. One lumen is the total amount of light falling upon 1 sq ft of surface, every part of which is 1 ft distant from a point source of 1 candlepower.

Luminaire. Luminaire is a term broadly used in industry to designate any type of lighting fixture, complete with globe, reflector, refractor, and housing.

A luminaire designed to distribute 90 per cent or more of its light output downward below the horizontal is classed as *direct*

A luminaire designed to distribute from 90 to 100 per cent of its light output upward above the horizontal is classed as *indirect.*

A luminaire designed to distribute from 60 to 90 per cent of its light output downward below the horizontal and from 10 to 40 per cent of its light output upward above the horizontal is classed as *semidirect.*

A luminaire designed to distribute from 40 to 60 per cent of its light output upward above the horizontal and the balance downward is classed as *direct-indirect.*

A luminaire designed to distribute from 60 to 90 per cent of its light output above the horizontal and 10 to 40 per cent of its light output below the horizontal is classed as *semi-indirect.*

Mat Finish. Mat finish is a dull finish opposite that of specular. Light hitting a mat finish has a minimum of reflected glare.

Mazda. Mazda is a trade name formerly used by General Electric and Westinghouse for most lamp bulbs made by them, both incandescent and fluorescent. The term is taken from the Persian word meaning "God of Light."

Reflection. Light is reflected from any given surface, depending on its finish. The ratio of the reflected light to the incident light is termed "reflection factor." A bright, shiny piece of metal, for example, will reflect much more light than a dark, dull piece of fabric.

Refraction. Light in passing from one medium to another passes through the second medium at an oblique angle and might be referred to as "bent" light.

Specular Surface. A specular surface is a bright, shiny surface, opposite that of mat finish. Light hitting a specular surface is reflected therefrom at an angle equal to the angle of incidence. The light reflected from a specular surface is usually very bright and has much glare.

Surface Brightness. Surface brightness is the degree of brilliancy or luminous intensity of any part of a surface when viewed from a designated direction.

Transmission. Transmission describes light going through a transparent or translucent medium. The ratio of light transmitted to the incident light is termed "transmission factor."

Watt. A watt is a unit of electrical power in which all lamp bulbs are rated.

Characteristics of Light

Both light and vision are essential for seeing. They may be considered as a partnership in which one is essential to the usefulness of the other.

Light is a form of radiant energy which is transmitted as a transverse wave motion from the source to the receiver and travels through space in the form of electromagnetic waves similar to radio, X-rays, and gamma rays. All electromagnetic radiations have a common characteristic in that they are propagated through space at the same rate. The only difference is in the wave lengths. The visible octave is only a small portion of the energy emitted by the light sources or sent out by glowing solids.

Figure 13-37 is a graph of the electromagnetic spectrum and shows the waves which manifest themselves in a variety of ways as noted above, depending upon their wavelength characteristics. Because seeing results only when radiant energy within the limits of the visible spectrum enters the eye, it is interesting to note the extremely limited range associated with vision. Some rays do not react upon the eye at all, and others are transformed into heat which may irritate the eye. Even in the visible ranges, the average observer will find that the colors in the spectrum of sunlight do not appear to be of equal brightness even though they may reach the eye with equal intensity

Figure 13-38 shows a curve indicating the degree of eye sensitivity to the various wavelengths of visible light. Those in the center portion of the visible spectrum provide normal high visual sensation Tests indicate that the yellow-green light of 5550Å has the greatest brightness If the visibility for this wavelength is designated a 100 per cent relative response, the curve represents the respective visibility with the other specular colors expressed in percentages of this wavelength. The curve is based on the average levels for daylight vision with the cones of the retina of the eye chiefly concerned in the seeing process.

At very low levels of illumination, approaching moonlight, only the rods of the eye are involved, and the eye-sensitivity curve will no longer have a peak at 5550Å. The whole curve shifts toward the blue part of the spectrum by about 480Å or to 5070Å. The wavelengths at this low level are seen as colorless only, those at the blue end of the spectrum appearing brighter than those at the red. This has proved important in installations involving very low levels, such as those required for "blackout" purposes. The eye is, therefore, capable of adapting itself under illumination values which range from a fraction of a foot-candle to several thousand. As mentioned before, the eye be 7,000 ft-c from the sun and 1,000 ft-c from the vault of the sky—a total of 8,000 ft-c. The horizontal intensity at the eye level will be about 1,000 ft-c. The light in the shade of a tree will be 300 to 500 ft-c.

2. Average daylight appears practically white or colorless. Skylight is definitely bluish because of the scattered rays of sunlight.

3. Daylight varies greatly in spectral character with the time of day and with the point of the compass. It ranges from one extreme, a combination of direct and indirect lighting on sunny days, to totally indirect lighting when the sky is completely overcast.

4. Direct sunlight has in December less than half the light value that it has in June.

5. On the average, in the United States, about one-third of the daylight hours throughout the year are cloudy. The sun shines an average of only five hours per day during the months of December and January.

6. The seeing conditions outdoors are the result of the brightness level and the distribution of brightness and color. The same seeing conditions can be produced indoors with artificial lighting, the chief difficulty being to supply the high level of illumination required to produce the necessary brightness of low-reflection-factor surfaces.

Action of the Eye

To understand the relationship of light to sight and its final result, seeing, some knowledge of the structure of the eye is needed, not necessarily knowledge of the disorders of the eye and visual nervous system but rather of the mind's ability to see as related to the character of the lighting.

The eye is one of the most delicate and miraculous organs in the human body. It is most frequently compared with a camera having compound lens (the cornea and lens of the eye combine to focus the light), a shutter (the iris), a black box (the eyeball), and a photographic plate (the retina) on which the pictures are recorded.

Paradoxically, we do not see with our eyes at all but with our brains. The image on the retina is the picture that the mind sees. Seeing is one-half physical and one-half mental. It is, therefore, more accurate to compare the eye with a television camera, for it takes light waves and transmits them to electric waves which become images. The eye is merely the device which turns light into the little electric current that the brain sorts to the ordered images we see. Without the brain or the optic nerve, the image on the retina is as useless as an undeveloped film.

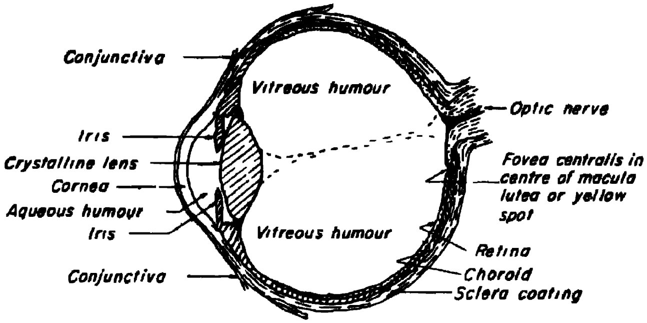
The process by which the eye is capable of receiving impressions from waves of light and passing them on through the optic nerve to the nerve cells in a definite area of the cortex of the brain remains one of the mysteries of the human body.

The eye itself is a globe composed of three layers of membrane called the sclera, the chorioid, and the retina. The eye is divided front and back into two chambers. In the front chamber are the crystalline lens, the iris (in the center of which is the pupil), the aqueous humor (a clear waterlike fluid), and the cornea. The posterior chamber is filled with a semifluid, semijelly substance, called the "vitreous humor." The inner lining of this chamber is the retina, and the back of the retina is the layer of pigment which gives the eye its color. Figure 13-39 shows a cross section of the eyeball and the position of the various elements of the eye.

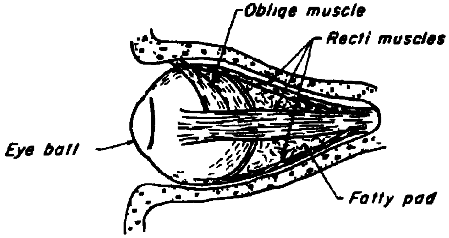
The eyeball is positioned in a bony depression of the skull which affords it much protection. The depression immediately back of the eye is filled with fat, giving the eye a cushion on which to turn. The eye can be moved in all directions by muscles attached to the eyeball and the bony socket. The exposed surface of the eye is covered by a highly sensitive membrane, the conjunctiva. The eyelids, which are quick-acting shutters ready to protect the eyes, are also lined with conjunctiva. The circular shutter over the lens, the iris, closes in over the lens in response to a strong light and to messages sent along a nerve.

All these mechanisms are a great protection to the delicate organ, but, in spite of these precautionary features, disease, deformity, and sometimes merely the strain of civilized activities hurt the eyes and reduce the efficiency of living. Inadequate food and air are quite immediately obvious in their effects, but this is not true with light and vision, since the eyes are much more patient and less complaining than the lungs or stomach. The correct principles of eye health are deeply rooted and fixed by nature, and anything that affects the general health of a person can't fail to have an effect upon his eyes. Most persons refuse to take as good care of their eyes as they do of their digestive systems. They suffer eyestrain without always knowing it and rebel at the suggestion that something else beside the lighting is the cause of the complaint.

The eye is admirably equipped to protect itself and function under widely varying light conditions. However, the prolonged use of the eyes at work, at home, or at play, under inadequate or glaring lights, leads to a strain on the physical reactions that can seriously affect the worker's health and well-being. Under proper lighting conditions the normal eye may see without strain. It is, therefore, the obligation of the lighting engineer to determine proper light sources, favorable lighting conditions, and adequate light intensities that are necessary to meet satisfactorily the requirements for the various visual tasks encountered in industry.



Cross-section of the human eye



Muscular attachments of the eyeball

FIG. 13-39. Cross section and muscular attachments of eye.

Eyestrain arises through muscular and not retinal activity. Low illumination levels or poor lighting conditions will result in poor visual acuity, and the balance between accommodation and convergence is likely to be disturbed. The efficiency of the eyes depends on their freedom from errors of refraction, defects of fixation, and the transparency of the cornea.

The first steps in reaching right seeing conditions are

1. Have the eyes of all employees examined at biennial intervals in order that refractive changes within the eye may be discovered and corrected before serious eyestrain develops.

2. Recommend that all employees who have visual defects be equipped with proper corrective lenses.

Some rules for eyesight conservation of actual preventive values, such as means of giving respite for near vision, good lighting, eye exercises, and generally better seeing conditions, will be discussed under Illumination Yardsticks.

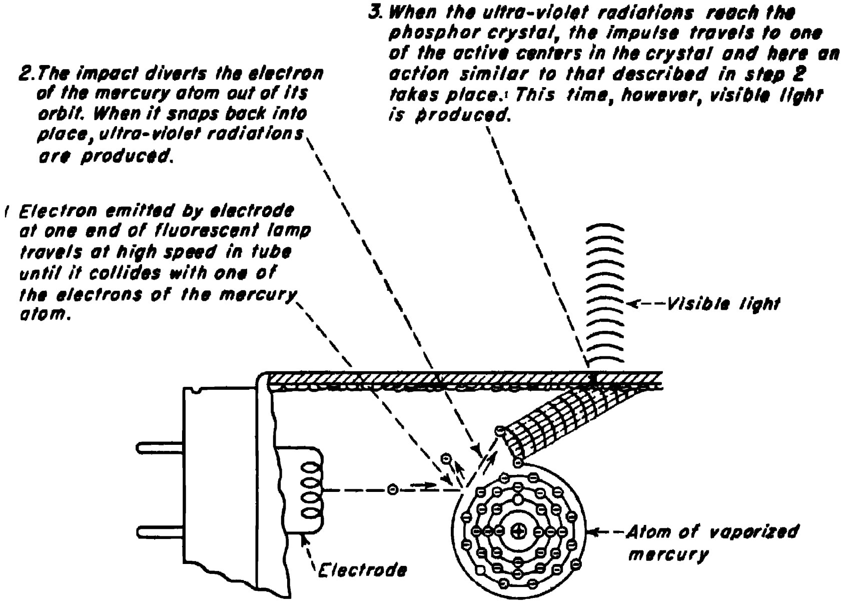
The most important facts concerning vision are enumerated below:

1. Relaxation is very closely associated with vision. It is estimated that the lack of relaxation is the cause of 80 per cent of defective sight. Tension of any kind blurs the brain and pulls the eyes out of focus.
2. About 82 per cent of all persons in this country over 50 years of age have defective vision.
3. Fifty million people in the United States either wear glasses all or part of the time or, in the opinion of eye physicians, need to wear them.
4. The eyeball is held forward in position by a fatty pad and can be operated in any direction by six muscles.
5. Four of the six muscles (known as recti muscles) are evenly spaced around the eyeball and pass directly back of the cavity.
6. The two remaining (oblique) muscles, which pass partly around the eyeball, then through a ligamentous loop before going back to the cavity, give the eye power to see at varying distances by squeezing the eye and changing the balance of pressure within the eyeball.
7. When the oblique muscles are tightened, the eyeball is elongated. When the recti muscles are tightened, the eyeball is shortened and thickened. This changing balance gives us our change of focus, the former when the eye focuses for the near point, as in reading.
8. The normal human eye forms without effort clear images of objects 18 or more feet away. Closer objects require accommodation by the recti muscles. The extrinsic muscles of the eye are means of accommodation. It is when these muscles tighten into habitual tension that eye trouble develops.
9. Eyestrain is the cause of impaired sight, not the result of it. Strain is caused frequently by improper lighting conditions. However, other circumstances result in improper use of the eyes and produce eyestrain.
10. The retina has two distinct types of vision: (*a*) central or the positive, clear vision only in the exceedingly small area in the center of eye; (*b*) collateral, or vision only to the extent of general form, general motion, and color which is progressively reduced beginning at the center of the retina outward until at the periphery it is very poor.
11. The eye sees only while it is at rest. There are no "sweeping glances" so frequently mentioned. The eyes do not move continuously, but in little jerks, jumping from point to point.
12. Our usual appreciation of the solidity of objects is due to the cooperation of both eyes. Each eye, taken separately, yields a slightly different impression, but the two impressions together combine to show what is known as "stereoscopic effect," a three-dimensional effect.

Comparison of Light Sources

Within the past few years we have witnessed a phenomenal increase in the multiplicity of light sources and control materials. Twelve years ago there were only two common light sources available: the incandescent tungsten-filament lamp and the Cooper-Hewitt mercury lamp. Newer light sources now include hot- and cold-cathode fluorescent, high-intensity mercury-vapor, and sodium-vapor lamps. All the newer sources have longer life and deliver more lumens per watt.

The choice of the light source to be used in a lighting installation deserves as much consideration as the choice of any of the other tools used in industry. The use of any light source is to convert electrical energy into light for the purpose of making the process of seeing easier and more comfortable. The possibility of carrying on work in industrial interiors depends upon lamp performance based on three factors: light, life, and uniformity. The economic factors involved in lighting will be discussed farther on in this subsection. At this point it is sufficient to state that the cost of power consumed during the life of a lamp is, in almost all cases and particularly in the case of the incandescent lamp, many times the cost of the lamp itself. Furthermore, it must be borne in mind that lamp life and light are interdependent. If the quality of light given by more than one type of lamp is considered to be suitable for a given industrial process, the final choice of type should depend on relative costs.


3. When the ultra-violet radiations reach the phosphor crystal, the impulse travels to one of the active centers in the crystal and here an action similar to that described in step 2 takes place. This time, however, visible light is produced.
2. The impact diverts the electron of the mercury atom out of its orbit. When it snaps back into place, ultra-violet radiations are produced.
1 Electron emitted by electrode at one end of fluorescent lamp travels at high speed in tube until it collides with one of the electrons of the mercury atom.
Visible light
Atom of vaporized mercury
Electrode

How a Fluorescent Lamp Produces Light

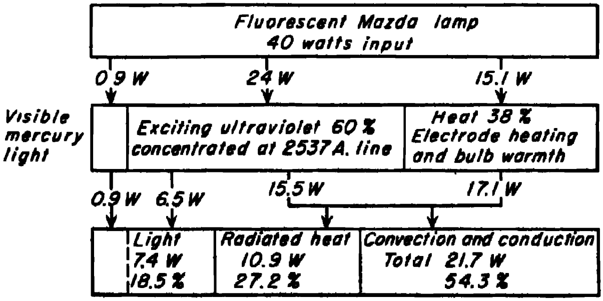


Fig. 13-40. How a fluorescent lamp produces light, and its distribution of energy.

All light sources are generators of radiant energy. As mentioned before, the generation of light has until recently been generally limited to the incandescence of hot solids. Much of the energy radiated by the tungsten filament lies outside the visible octave, and for this reason, it is not very efficient as a light source. The heat losses are on the order of 90 per cent of the total input. For a commercial tungsten 200-watt lamp, the efficiency is but 18 lumens per watt. In the relatively short period of 40 years, we have advanced from a carbon source producing 3.5 lumens per watt to the

Table 13-21. Comparison of Present-day Light Sources

Characteristic	Incandescent	Fluorescent	Mercury vapor
1. Type of source. . .	Point or concentrated	Linear	Point and linear
2. Color quality of light produced	Rich in the yellow and red portions of the spectrum varying from candle-flame yellow to light yellowish white. Also made in colors	Daylight, 4500°K white, white, warm tone, and soft white. Also made in colors: blue, green, pink, gold, and red	Almost a monochromatic light is the blue and green portions of the spectrum, which is particularly well adapted to lighting the neutral tones in metal working
3. Bulb finishes	Clear, inside frosted, white bowl, daylight, silver bowl, silver neck, color coated, and natural colored glass	Daylight, 4500°K white, white, warm tone and soft white. Also made in colors: blue, green, pink, gold, and red	Clear
4. Rated operating life. . .	750 hr up to 150 watts. 1,000 hr over 150 watts. Also special-life lamps	2,500 to 7,500 hr, varying and increasing with longer operating hours per start	2,000 to 4,000 hr, varying and increasing with increased operating hours per start
5. Lamp efficiency . .	10 to 20 lumens per watt, depending on lamp size (average of 15-in. range of 60- to 200-watt sizes)	39 to 62 lumens per watt (not including auxiliaries), with auxiliaries produces about 2 to 3 times as much light per watt as incandescent	30 to 40 lumens per watt (not including auxiliaries)
6. Lamp wattage.	6, $7\frac{1}{2}$, 10, 25, 40, 60, 100, 150, 200, 300, 500, 750, 1,000, 1,500, 2,000 in standard sizes	4, 6, 8, 14, 15, 20, 30, 40, 65, 85. Also Slimline units 18 to 96 watts	100, 250, 400, 1,000 (water-cooled), 3,000
7. Lamp size and shape . .	2- to $6\frac{1}{2}$-in. diam. in tear-drop and pear shape; also short tubular	Tubular $\frac{3}{4}$ to $2\frac{1}{8}$ in., 6 to 96 in. long, circular 25 watts, $1\frac{1}{4}$-in. tube, 12-in. diam	250- and 400-watt short tubular; 3,000-watt long tubular source
8. Relative installed costs for equal lighting intensity 40 to 50 ft-c	100 per cent	165 per cent	180 per cent
9. Relative maintenance costs			
a. Lamp replacements including labor	100	110	105
b. Cleaning and maintenance.	100	300	130
10. Relative power costs	100	40	65
11. Relative cost per foot-candle year.	100	75	85
	(See notes at end of table for basis of comparisons)		
12. Advantages	Availability and convenience. Requires no control gear, ballast, or starters. Light distribution subject to beam or spread control. Starts instantly when turned on. Will withstand repeated starts better than any other source. Can be used in all kinds of weather. Installation costs relatively low	High lumen output per watt. Less heat is obtained. Surface brightness is low. Long lamp life. Lower current consumption for a given light output. More illumination can be obtained from old wiring by a change to fluorescent	High lumen output per watt. Useful for high bay mounting. Light is of particular advantage for critical inspection work. Because of green color, operating cost relatively low
13. Disadvantages.	Color is too rich in the red and yellow. Efficiency per watt is relatively lower than fluo-	Accurate beam control from the tubular lamp is not possible. Requires auxiliary equip-	Color of light is the chief objection to widespread use. Requires auxiliary equip-

Table 13-21. Comparison of Present-day Light Sources (*Cont.*)

Characteristic	Incandescent	Fluorescent	Mercury vapor
	rescent or mercury. Much heat has to be dispersed or dissipated; characteristics of operation are greatly affected by voltage variation from normal rating	ment, ballast, starters, condensers, all of which complicate maintenance and increase weight and bulk of installation. Standard units will not give satisfactory operation in cold or high-temperature locations. If operated with a single lamp ballast, stroboscopic effect may be present. Can be operated on direct-current but at decreased lumen output, lamp life, and starting reliability	ment. Delayed start of 3 to 7 min. Cost of equipment is relatively high

NOTES. Items 8, 9, 10, and 11 are given as a basis of comparison of lighting costs for providing a grade B (40 to 50 ft-c in service) lighting intensity in an area 100 by 100 ft square under the following conditions: complete wiring and conduit installation including incoming line services, transformers, panels, etc ; 500-watt incandescent glassteel diffusers, 150-watt fluorescent RLM reflectors (3 to 40-watt lamps); 250-watt mercury enclosed RLM luminaire; 8 000 hr total operating time per year; 20 per cent annual owning cost per year (5-year amortization). All relative percentage costs are established, using the incandescent system as the base The ratio of relative costs will change for each and every type of installation, depending upon various economic factors and their relative influences. Data based on a composite of plant experience Cost of allocated facilities not included. Electric energy based on purchased cost of $0.01 per kwhr.

40-watt fluorescent lamp thirteen times as efficient. With these illuminants the electric energy is transferred to ultraviolet and visible radiations without incandescence. Figure 13-40 shows how a fluorescent lamp operates and its distribution of energy.

The ultimate efficiency that may be expected for white light is approximately 200 lumens per watt. Figure 13-41 is a graph showing the comparative efficiency of light sources. Today, with gaseous-discharge lamps, we have efficiencies of 40 to 60 lumens per watt. Within the next few years, with new illuminants or high-pressure mercury arcs, the efficiency may advance to 100 lumens per watt.

Intelligent use of light to meet industrial man-power-production requirements calls for an understanding of the characteristics of light sources. Incandescent, fluorescent, and mercury-vapor lamps are briefly appraised and reviewed in Table 13-21 to assist in an understanding of the many opportunities they offer in installing lighting to meet present and future man-power-production requirements.

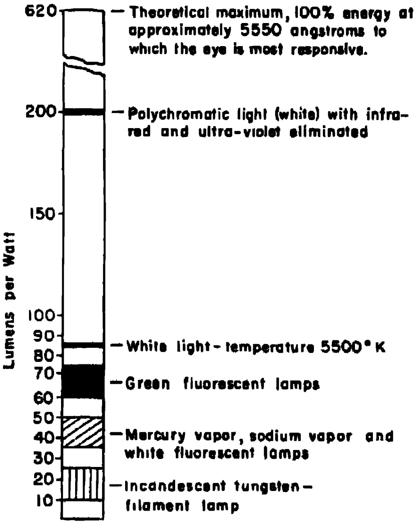


FIG. 13-41. Comparative efficiencies of light sources.

The incandescent filament lamp will always fulfill some purposes better than any other light source. Also, because of economy of installation, flexibility, ease of maintenance, and convenience of lamps and incandescent lighting equipment, it will always have a wide application.

The fluorescent lamp has a much longer life than the incandescent and will provide approximately two to two and one-half times as much light per watt as the latter type. Wartime lighting levels made possible by the use of fluorescent lamps have been carried over into civilian uses. With the return of peace after the Second World War, the range of colors, the quality, and simplification of equipment improved. It is expected that fluorescent lighting will assume the major place in general industrial-plant lighting.

The fluorescent Slimline is the newest lamp development. This lamp is appropriate for any fluorescent lighting installation. Its small diameter makes possible its use in places where space is limited. The 72- and 96-in. lengths give longer lines of light with fewer interruptions, as they are instant-start and have single-pin terminals, which make for simpler wiring. The long lengths help appearance and reduce the number of lamps, sockets, and ballast needed. The current range allows low, medium, or high lamp brightness through the control of current varying from 120 to 600 ma.

The mercury-vapor long has been the most economical light producer. The blue-green color is the chief objection to its widespread use. In many large industries, chiefly metal work, the color has its advantages. Combined with incandescent lamps, the mercury source promises to be more widely used, especially where large areas are to be lighted from high mountings.

For plant road and street lighting, the mercury color is not unlike moonlight, and its high efficiency provides more adequate illumination for the same expenditure. The sodium light will be used as a warning of danger, at heavy-traffic intersections, approaches to bridges, underpasses, and to indicate similar hazards.

Effects of Voltage Regulation on Lamp Operation

Industrial distribution systems and utilization equipment are designed on the basis of a nominal constant voltage. Lamps are designed to give the most economical combination of characteristics at rated voltage. In the case of incandescent lamps, the light output varies directly as the 3.4 power of the rated voltage, and the average life of the lamp varies inversely as about the thirteenth power of the rated voltage.

Fluorescent lamps are not affected so critically by voltage variation as are incandescent lamps. This is one of the advantages of fluorescent-lamp sources. However, undervoltage (below 105 volts for standard lamp) will decrease the light output and result in uncertain starting and operation. Overvoltage (above 125 volts) causes inferior lamp performance and danger of overheating the auxiliaries.

Mercury-vapor lamps are affected in light output very much the same as are incandescent lamps. Abrupt dips in voltage may cause objectionable light flicker in any type of lamp.

The Lighting Source for the Job

Adequate lighting may be provided by incandescent, fluorescent, or mercury-vapor lamps. It is assumed that each of these three types has its specific basic application and that there is one type of lamp best suited to any one job. As a matter of fact, this is seldom the case. More often there are two types of lighting sources each of which may serve equally as well and also some problems that will require the combination of all three sources to provide adequate lighting.

The final choice of a lighting source for any particular application may result from one of the following factors:

1. Some local physical limitation, such as machinery and structural members
2. Some economic factor peculiar to the job
3. Some operating condition, such as gas and explosion hazards, dirt, and corrosive vapors
4. The psychological reactions which different persons have for the effects of the various sources and types of lighting

It may be asked which lamp is best suited for a machine shop. Machine-shop work covers every grade of work from reading the fine calibration on a gauge to the visual task of a crane operator or a welder. Certainly, here are extremes that do not necessarily need the same type of lighting. Also, an office or salesroom may be paying 3 to 4 cents per kwhr for a small power usage, or less than 1 cent for a large usage. The same type of lamp may not be the most economical in each case.

Again, two manufacturers may be making the same type of product with about the same machinery layout, but one may be making a quality-line product in which inspection may be the most important item in the manufacturing process, while the other may be making a price-line product in which low-cost production outweighs quality. These two plants would require vastly different approaches to their lighting problems.

One process may be experimental in nature and considered more or less temporary, while for a similar process the plant may be built for long-term usage. These lighting problems will most likely be different.

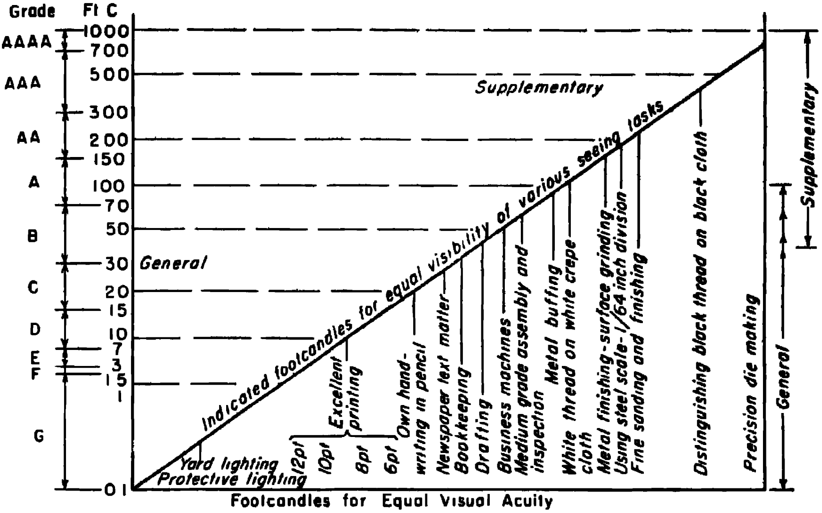


FIG. 13-42. Specifications of general lighting

In an office building, certain areas such as corridors and general conference offices may be adequately lighted with incandescent lamps, while the more detailed work will require lighting levels which can only be produced satisfactorily by fluorescent lamps. The latter are particularly effective in court offices because they help to overcome by psychological means the cramped feeling brought about by the lack of an outside view.

The one requirement which all lighting installations should have in common is that the maintained lighting level and conditions should be adequate for the various industrial seeing tasks.

In general, disregarding somewhat the economics of the problem, the three light sources may have their specific basic applications considered as follows:

1. Incandescent lighting for illumination levels up to 20 or 25 ft-c. For higher levels, wattage requirements and resultant energy costs will be excessive. Also the heat radiated may be a problem.

2. Fluorescent lighting for high illumination levels from 25 ft-c up. Fluorescent lamps illuminate with less glare and less heat and produce many more lumens per watt than do the incandescent lamps. Engineered fluorescent lighting is the most practical form of illumination for precision operations and is particularly effective in inspection departments, machine shops, drafting rooms, and other departments where seeing tasks are most precise.

3. Mercury lighting for high illumination levels where high mountings are involved.

It will prove most satisfactory in foundries, forge shops, steel mills, and large factory areas where the room proportions are excessive. It may be combined with incandescent lamps to bring back the red and orange tones into the over-all lighting spectrum and permit more natural identification of colors such as the worker's skin and production material.

Table 13-22 lists the general industrial applications for various light sources within the limitation of standard present-day light sources. This table shows the overlapping usages of light sources as mentioned above. The grades of illumination noted in Table 13-22 are outlined in Fig. 13-42.

This figure is not necessarily the recommended foot-candle illumination level for the various visual tasks shown. Visual acuity is the ability to see minute detail. Using as a basis 8-point Bodoni Book type (8-point Bodoni Book) illuminated to a level of 10 ft-c and viewed at a distance of 14 in., the other tasks will have the same visibility under the various levels indicated.

In most cases the level of illumination is not critical within 25 to 50 per cent. Actually, there is no substantial difference between 4 and 6 ft-c, 15 or 20, etc. Seeing is a very complicated process, and ease of seeing is, in some cases, more important than visual acuity. The two do not go hand in hand. The final appraisal of a lighting system rests on comfort and see-ability which will be discussed under the main titles How Much Lighting Is Enough for Optimum Seeing? and Illumination Yardsticks. For general purposes it is necessary to refer not to exact levels of illumination but to grades of illumination, as indicated in Fig. 13-42.

Table 13-22. General Industrial Applications for Various Light Sources

(Within the limits of standard present-day light sources)

Light Sources and Symbols

Incandescent filament lamps	I
Fluorescent lamps	I
Mercury lamps	H
Combination of incandescent and mercury lamps	IH
Combination of incandescent and fluorescent lamps	II
Substandard (not applicable)	X
Realm for special recommendations for each installation	*

Industrial area	Grade of illumination† G	I	F	D	C	B	A	AA and Up
Production areas								
Low bays	X	X	X	X	I I	F H	I H	*
Medium bays	X	X	X	X	I I H	I III I	I IH H	*
High bays	X	X	X	X	I H	I H IH	*	
Chemical plants	X	X	X	I I	I I H			
Supplementary lighting	X	X	X	X	X	I I	I I	I I
Receiving and shipping room	X	X	I	I				
Storage areas	X	I	I					
Aisles, passageways, and stairways	X	I	I					
Yards	I H	I H	I H					
Protective lighting	I H							
Offices								
General offices	X	X	X	X	I I	F	*	
Drafting rooms	X	X	X	X	X	I F	F	
Mail and file rooms	X	X	X	I	I I			
Service areas, halls, etc.	X	X	I	I I				

† For grade of illumination see Fig. 13-42.

Newest Developments in Lighting Sources

A new type of fluorescent lamp called Slimline is approximately ¾ in., 1 in., and 1½ in. outside diameter, 42, 64, 72, and 96 in. nominal length and rated 1,070 to 5,000 lumens. Some of the features of these new lamps are

1. Instantaneous starting; no starter needed
2. Longer life than ordinary fluorescent
3. Higher efficiency; average of 60 lumens per watt
4. Ready adaptation to certain architectural treatments because of its small diameter and longer length
5. Adaptation to flashing

Despite their efficiency, the shape of the present fluorescent lamps prevents their use in ordinary portable units in which a long tube could not easily be accommodated. The lamp manufacturers have developed a circular fluorescent lamp called the "Circline," rated at 32 watts with a 1¼-in.-diam tube and an outside-ring diam of 12 in. This lamp, in addition to being adaptable for use in portable lamps, is being used in industry for inspection lighting.

Other types of light sources may come into use in the near future. Most assuredly, drastic improvements will be made in some of our present systems.

Comparison of Lighting Fixtures and Systems

Lighting Fixtures. An efficient lighting system requires reflectors or luminaires in order to control the light distribution and to reduce the high surface brightness that is produced by the various light sources. For general purposes today, the lighting quality of an installation is frequently judged not so much in terms of lighting intensities or the quantity of light as in the degree of shadow and the amount of direct and reflected glare. The luminaire which controls these factors is the backbone of the lighting system. The type of unit, the desired distribution of light, and fixture suitability for the particular interior depend upon the type of work conducted and the room proportions.

In any given interior or for any type of machine arrangement there may be several methods of illumination design that will produce effective results. With the development of the mercury lamp in 1935 and the fluorescent lamp in 1938, the problem of selecting the types of lighting sources and lighting fixtures for a given design or condition has been greatly multiplied. To ensure that the elements of quantity and quality of lighting will be right, the system of lighting and the type of fixtures should be carefully selected. The choice deserves as much, if not more, consideration as the choice of any of the other tools used in industry.

Except for the standard RLM type of lighting unit, manufacturers have made practically no attempt to standardize fixtures. Their catalogues show thousands of various types and sizes of units, varying from the open-lamp unit to decorative trollers. However, an analysis of the many designs shows that many makes are used to serve a definitely limited number of lighting situations and that practically all the units may be classified and put into a few groups to meet specific installation conditions and seeing tasks.

The different types of reflector and luminaire equipment used currently in the various areas of industrial plants are listed in Table 13-23. This table is an outline of the several classes of fixtures used with the three principal light sources: incandescent, fluorescent, and mercury. After classification, the choice of the particular make may be made, with consideration given to quality, appearance, and ease of installation and maintenance as prime requisites. It must be remembered that the best type and size of lighting unit is the one which meets the need for lighting, contributes the most to aid the worker accomplish his duties, and provides the decorative effect desired. The engineer should not make the choice of lighting fixture or installation on a cost analysis until it is first reduced to a comfort and see-ability basis of comparison. These factors will be discussed later in this subsection.

Ever since the technique of lighting has been a problem for serious consideration, there has been a tendency to use larger surfaces of lower brightness for the distribution of light. Any brightness in close proximity and in the line of vision greater than 1,000 ft-lamberts will tend to be uncomfortable. The best system of lighting confines brightnesses in the usual field of vision to 300 ft-lamberts or less. The general order of brightness of the various light sources and some of the more common types of fixtures are given in Table 13-24. These figures show the necessity for shielding practically all

Table 13-23. Present Types of Lighting Equipment and Systems

Group	Description	Wattage
	Incandescent Industrial	
A	RLM metal-dome reflectors—three sizes. 12 to 20 in. with porcelain enamel reflecting surfaces that provide well-distributed light for simple-factor seeing tasks	Six wattages—150- to 1,500-watt inside-frosted lamps. three wattages—150- to 300-watt white-bowl lamps
B	Glass- or metal-bowl reflectors providing concentration or reflection for more exacting tasks	Six wattages—200- to 1,500-watt; clear or inside-frosted lamps
C	Glassteel diffusers—three sizes combining 10-, 12-, or 16-in. diffusing globes with RLM-type reflector to provide high-quality diffused lighting for more exacting tasks	200- to 1,000-watt clear lamps
D	Deep-bowl high-bay units with specular reflectors to provide lighting in difficult areas where fixtures must be mounted 20 ft or more above floor	Five wattages—300 to 1,500 watts
E	Special dome or trough reflectors with silvered-bowl lamps for diffused lessened-glare lighting conditions	Four wattages—150- to 500-watt silvered-bowl lamps
F	Supplementary units of many sizes, designs, and uses. Available in many types of adjustable mounting. Used only to supplement general lighting	
	Fluorescent Industrial	
G	RLM and open-end units with porcelain-enamel steel reflectors for general lighting in ordinary industrial interiors	Two- or three-lamp 40-watt units. Two-lamp 85 watt units and two- and four-lamp Slimline—120 to 600 ma
H	Continuous troffer for flush mounting. Form of RLM type unit designed for end-to-end mounting; used to provide high level of general illumination in large factory areas or bench lighting. Flush mounting used for installation in acoustical tile or plastic ceilings, particularly in areas where conditions are controlled	Two- or three-lamp 40-watt units. Two-lamp 85-watt units and Slimline
J	Supplementary units of many sizes with diffused or concentrating reflectors. Used in lighting small areas for high lighting levels above 100 ft-c without heat discomfort	
	Mercury Industrial*	
K	RLM standard dome or glassteel units with porcelain-enamel steel reflectors. Used for general lighting in industrial interiors where color and stroboscopic effect are not important	400-watt AH-1 lamp 250-watt CH-5 lamp 100-watt AH-4 lamp
L	Deep-bowl high-bay units with metal, mirrored, or prismatic glass reflectors. Used to provide economical lighting in areas where fixtures must be mounted 20 ft or more above floor	400-watt AH-1 lamp
M	Trough reflector units for high-bay mounting—40 ft or more above floor	3 000-watt H-9 lamp
	Incandescent Commercial	
N	Direct-lighting units with controlled distribution and equipped with louvers or lens plates to reduce the unit brightness. Recessed units give uniform illumination in low ceiling areas and may be used in combination with other units to add variety and achieve unusual lighting effects	Four wattages—100 to 500 watts

Table 13-23. Present Types of Lighting Equipment and Systems (*Cont.*)

Group	Description	Wattage
P	Semidirect or semi-indirect units including a variety of luminous-bowl or prismatic-glass fixtures. Used to provide illumination fairly well distributed on both horizontal and vertical work surfaces. In using this unit it is necessary to guard against brightness in the field of view whenever prolonged visual tasks are encountered	Five wattages—200 to 1,000 watts
Q	Indirect lighting units with metal, glass, or plastic bowls which direct most or all the light to the ceiling for reflection to the working plane. This type of unit is inherently inefficient but has the advantages of low brightness, inconspicuous appearance, and almost shadowless, uniform distribution of light. Very uneconomical for lighting levels above 30 ft-c	Four wattages—300- to 1,000-watt inside-frosted lamps
	Fluorescent Commercial*	
R	Suspension-type semidirect or semi-indirect units with glass and plastic diffusing panels or metal louvers to reduce brightness and shield the lamps from the eye. May be used singly or continuously for high ceiling areas. Distribution of different units varies, but the majority of fixtures light both up and down	Two to four lamps. 40 and 85 watts, and two- or four-lamp Slimline—120 to 600 ma
S	Surface-mounted units similar to group R except that they are designed for mounting directly on the ceiling. For general lighting in all rooms with low ceilings (approximately 10 ft or less) or in any room where the width is equal to or greater than the ceiling height	Two to four lamps. 40 and 85 watts and Slimline
T	Recessed troffer units for installation flush with the ceiling surface and equipped with louvers or glass diffusing or lens panels; designed for use with acoustic-tile ceilings or simple installation in lines or rectangular forms in any suspended ceiling. Fixture appears to be part of ceiling and is only type of direct lighting unit satisfactory for office use	One and two lamps. 40 and 85 watts and Slimline
U	Indirect lighting units with metal or translucent glass or plastic bowls which direct most of the light to the ceiling for reflection to the working plane	Two and more lamps. 40 and 85 watts and Slimline
V	Suspension-type direct unit with metal or low-transmission-factor glass. Similar to group R and has characteristics of group Q except that this unit may be used to produce lighting levels up to 100 ft-c economically	Two and three lamps. 40 and 85 watts and Slimline

* There are many combinations of incandescent and mercury industrial systems. The most popular is an alternate spacing of incandescent and mercury luminaires. Other units are combination reflectors, either metal or glass, with one mercury lamp and one or more incandescent lamps.

lamps in an industrial system designed for maximum comfort and see-ability except when the fixtures are mounted high and outside the normal viewing angle. Table 13-24 also shows that, with reference to brightness, frosted incandescent bulbs are superior to clear ones for general lighting. They reduce the shadows cast by small objects because the entire surface of the frosted bulb acts as a light source. The larger the light source is in relation to the obstruction, the smaller and less pronounced will be the shadows.

Figure 13-43 is a chart which indicates the approximate wattage load per square foot required to produce different light-intensity levels with the various types of lighting fixtures listed in Table 13-23. On the chart, follow the horizontal line beginning at the maintained foot-candle value desired, until it intersects the curve corresponding to the fixture type. The watts per square foot may be read directly below the intersection point. For large rooms (50 by 100 ft) higher horizontal intensity levels may be expected; for small rooms (12 by 15 ft) lower values will result. This chart should be used for estimating loads to determine the order of magnitude only. The quantity and quality of lighting that will be obtained in a working area under normal working conditions will depend upon such important items as brightness, color, contrast, room proportions, and the size and shape of the objects in the room.

Table 13-24. General Order of Brightness of Various Light Sources and Luminaire Equipment

Light source or equipment	*Exposed brightness approx ft-lamberts*
1. Incandescent lamps	
a. Open dome and bowl reflectors	
Clear bulb—filament	3,000,000
Inside-frosted bulb—1 000-watt lamp	120 000
Inside-frosted bulb—500-watt lamp	95,000
Inside-frosted bulb—200-watt lamp	65 000
Inside-frosted bulb—100-watt lamp	50,000
Inside-frosted bulb—60-watt lamp	16,000
b. RLM domes and white	
Bowl lamps—300-watt lamp	10,400
Bowl lamps white—200-watt lamp	6,300
Bowl lamps white—150-watt lamp	9,500
c. Glassteel diffusers	
16-in. globe—1 000-watt lamp	6,300
16-in globe—750-watt lamp	4,800
12-in globe—500-watt lamp	5,400
12-in. globe—300-watt lamp	3,100
10-in globe—200-watt lamp	2,850
d Open dome and trough reflectors (mat finished with silvered-bowl lamps)	
26-in diam side of reflector—500-watt lamp	1,820
19-in diam side of reflector—300-watt lamp	2,400
19-in diam side of reflector—200-watt lamp	1 720
2 Type-H mercury lamps	
a. Open dome and bowl	
400-watt type-H-1 lamp	407 000
250-watt type-H-9 lamp	390 000
b Glassteel diffusers—16-in globe	
400-watt H-1 plus three 150-watt incandescent	7 200
400-watt H-1 lamp	5 000
3. Fluorescent lamps	
a. Open enameled-trough reflectors	
85-watt 3500° white 60-in T-bulb	2 180
40-watt 3500° white 48-in T-bulb	1 750
40-watt 4500° white 60-in T bulb	950
b. Lenses and louvers when properly applied will reduce the surface brightness to 300 ft-lamberts and less	
4. Natural brightness	
a. Sun at zenith.	450,000 000
b. Cloudless sky	2,500–5 000
c. Moon (full)	900–1,300

Lighting calculations and tables giving light-intensity levels for continuous-row 40-watt fluorescent lighting units may be used for the 425-ma Slimline lighting units as their lumen-output value per foot of lamp is approximately equal.

The most important facts concerning the selection of lighting fixtures are listed below:

1. The luminaire is the backbone of any lighting system, for upon it combined with the light source depends the quantity and quality of the illumination.

2. The lighting installation in both commercial and industrial plant areas should be made as inconspicuous as possible.

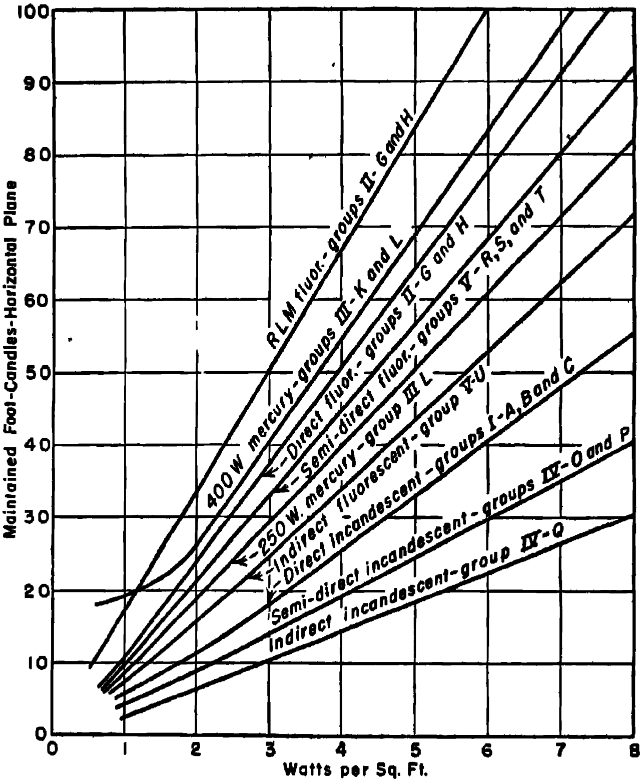
3. Lighting fixtures should be carefully designed with proper shielding and made of materials to provide high efficiency and ease of maintenance

4. In planning a lighting system for any project, the lighting result should first be selected. The subject of "lighting result" is very broad in scope and covers

a. The aesthetic effect, architectural treatment, and decorative scheme
b. The type, size, and color of the light source
c. The type of luminaires, reflectors, or lighting equipment
d. The wiring method to be used

5. The incandescent lamp is a highly concentrated light source, and when used for general lighting, even in indirect equipment under the most favorable lighting conditions, the contrasts in brightness are usually so great that lighting levels in excess of 30 or 35 ft-c are seldom comfortable.

6. The fluorescent lamp is a diffuse light source and has a low surface brightness. In the standard lamps the order of brightness is about four candlepower per square inch, and while this is too great for comfort when used unshielded, it is quite simple to reduce the brightness to safe limits by simple luminaire construction.



This chart is intended as a guide in estimating load and wiring capacities for foot-candle levels with various types of lighting fixtures. When planning layouts for selected fixture, the illumination should be calculated using photometric distribution curves for that particular figure.

Fig. 13-43. Approximate load required to produce different light-intensity levels with various types of lighting fixtures.[8]

7. By properly combining several fluorescent lamps within a reflector or behind diffusing panels, it is easy to create an extended low brightness source. For this reason, direct-type fluorescent-lighting units can be made to produce just as effective lighting characteristics as an indirect incandescent-lighting installation.

8. When heating from the artificial lighting source is a problem, the fluorescent lamp actually directs less radiant heat toward the worker than does an incandescent lamp of the same wattage.

9. Loss of human life, as well as property by explosions and fires, warrants careful study where the application is to be made for hazardous chemicals, processes, and

industries. The increased expense incident to vaportight, dust-tight, and explosion-resisting equipment installations is justified for hazardous locations.

Lighting Systems. In general, there are three types of lighting systems: direct, semidirect, and indirect. Listed below are the advantages and disadvantages of each system.

The general advantages of the *direct* type of illumination are

1. First cost is lowest.
2. Efficiency of power utilization is best.
3. Maintenance upkeep is lowest.

The disadvantages are that it is

1. Not so good as other systems as far as direct and reflected glare is concerned.
2. Worse for the production of shadows but, with proper equipment and proper installation, may be made satisfactory.
3. Not so favorable in appearance as other types. The direct type of illumination is applicable for most industrial installations, but for eye comfort such installations must have louvers, reflectors with proper light cutoff, or diffusing glass plates and panels.

The advantages of *semidirect* are

1. Favorable appearance
2. Good for elimination of shadows and glare
3. First cost intermediate between the other two types
4. Second in efficiency as far as power and maintenance are concerned

The disadvantages are that it is

1. Not so good as indirect lighting so far as direct reflected glare is concerned
2. Not so efficient for use with dark ceilings and walls

The advantages of *indirect* are that it is

1. Best for the elimination of glare
2. Best for the elimination of shadows

The disadvantages are

1. Poor efficiency as far as power and maintenance are concerned
2. Highest first cost
3. Cannot be economically used with dark ceilings and walls

In making a selection of lighting fixtures, various factors must be considered with respect to the particular problem at hand. For instance, in an office the major factors rank as follows:

1. Freedom from direct and reflected glare
2. Adequate illumination levels on the horizontal working plane
3. Pleasing aesthetic or decorative results
4. Utilization factors, such as economics of installation and maintenance

In general industrial-process areas the order of importance might be

1. Adequate illumination levels on both horizontal and vertical working planes
2. Freedom from detrimental direct and reflected glare
3. Utilization factors
4. Aesthetic or decorative results

Many needs for lighting around a plant, such as stairs, storage areas, loading platforms, and like areas, are well satisfied by the simplest and cheapest expedient. Comfort and see-ability, as far as a lighting quality is concerned, will be discussed under Illumination Yardsticks. Installation hazards, such as explosion-resisting, excessive corrosive conditions and dust, call for different forms of fixture construction and location, all of which reduce the lighting output which must be charged against safety and/or ease of maintenance. Even clear-glass covers reduce the light output of fixtures 10 to 20 per cent.

Chemical-process plants in almost all cases have a particularly difficult problem of lighting. The dangerous nature of many of the chemicals requires dust-tight and explosion-resisting lighting fixtures. Also, there is a need for special lamps and equipment for applications: (1) where high ambient temperatures prevail, (2) where there is excessive vibration, and (3) where the service is too rough for the standard lamps.

In general, there are three main methods of lighting for explosive or highly flammable atmospheres, and the choice of the method depends upon the seriousness of the risk involved.

1. Light sources placed external to the building and arranged to throw their light through a glass roof or window direct on the working plane or by reflection from light-covered surfaces

2. Light sources placed outside the danger zone but inside the building or in the walls so that relamping can be done in a nonhazardous location

3. Flameproof fittings or lighting units under a small inert-gas pressure employed for those hazardous gas areas where it is impractical to light the working area from sources listed in (1) or (2).

In most explosive-dust locations it is sufficient to use a well-gasketed dust-type of fitting which effectively encloses the lamp, has a smooth contour which prevents an excessive accumulation of dust, and which is designed to hold the temperature of the unit within safe limits.

In addition to installations in chemical plants which require treatment for explosive materials, fumes, and vapor, there are other areas which require vaporproof units for installation in areas where excessive moisture, alkali fumes and vapors, and acid fumes and vapor are prevalent. Fittings or hoods for use with vaporproof units are made of black or galvanized cast iron, silicon aluminum alloy, or bronze. In general, in locations where explosive materials, acid fumes, or excessive moisture are present, either the cast iron with an asphaltum-base-paint finish or the silicon-aluminum-alloy fitting will give good service. In places where alkali vapors are present, galvanized cast-iron fittings are preferable; aluminum fittings should not be used. In some extreme acid conditions, lead-alloy or plastic fittings should be considered.

Some of the significant facts concerning the selection of a lighting system for use in various plant areas are as follows:

1 The best type of natural lighting is that which is reflected from a cloudless northern sky. It is uniform and diffused and produces no highlights, glare, or shadows.

2. The closest approach to natural light obtainable with artificial lighting is by means of a large-area low-brightness light source. Properly applied indirect lighting approaches the natural condition noted in 1 if the lighting level is adequate for the work involved. However, the broad expanse of the illuminated ceiling is by far the brightest area in the room. Glare may not be eliminated to the point where it is not objectionable.

3. All local lights for a machine should be built in and made just as much a part of the machine as any of its working parts. Very few local lights will prove satisfactory when they are under the unguided control of the operator.

4. With incandescent lamps, indirect-type lighting is usually considered essential for satisfactory office and drafting-room lighting. The lower brightness of the fluorescent lamp permits the use of semi-indirect equipment to give similar lighting quality, with the result that an increase of lighting efficiency of about one-third is gained because of the differences in the utilization factors of the semi-indirect equipment over those of the direct type.

5. The fluorescent lamp has little competition in any industrial application so far as brightness, which may cause uncomfortable direct or reflected glare, is concerned. It is easier to achieve all the objectives of adequate lighting with a fluorescent-lighting system than by any other means.

6. The larger the light source, in relation to an obstruction, the better. Shadows will be smaller and much less pronounced.

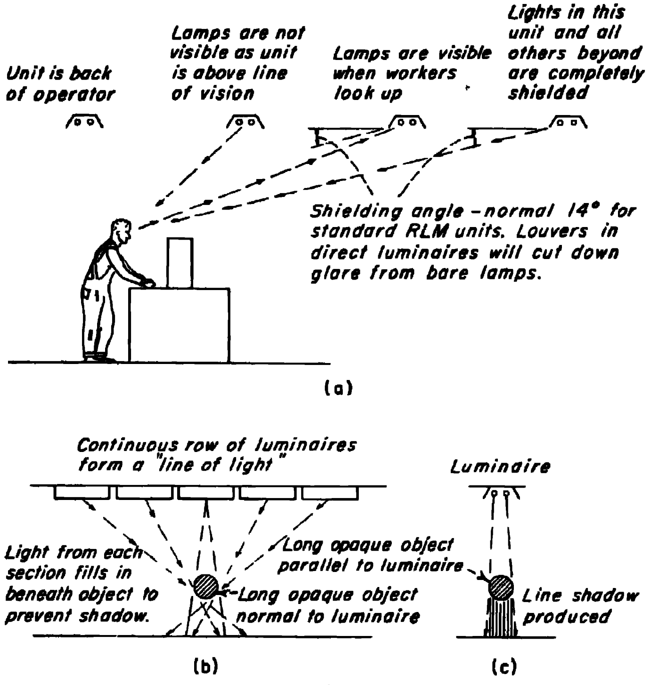
7. Often the fluorescent lamp has been referred to as a large-area low-brightness source. This claim is not entirely true, since the lamp itself is a slender linear source whose width is very small in comparison with its length. However, with reference to brightness and glare, fluorescent equipment can be designed, fabricated, and engineered for installations which make it an excellent low-brightness source.

Exterior Plant Lighting. Exterior plant lighting is used in industrial plants for (1) protection; (2) passageway, roadway, and yard illumination; (3) outdoor work; (4) advertising, stack lighting, etc.

Protective-lighting installations must be completely reliable and resistant to weather, damage, and sabotage. A failure of a part of the system should not leave a vulnerable area unprotected.

Most exterior plant lighting is done by some form of street-lighting unit or floodlighting equipment. The following classes of equipment are available:

1. Street-lighting units with distribution controlled by reflectors and lenses mounted on poles for lighting fence lines, roadways, wide passageways between buildings, etc. A Fresnel-lens unit developed specifically for fence lighting provides a uniform level of illumination with good utilization in a vertical plane—a most important feature in protective lighting.



Unit is back of operator
Lamps are not visible as unit is above line of vision
Lamps are visible when workers look up
Lights in this unit and all others beyond are completely shielded
Shielding angle - normal 14° for standard RLM units. Louvers in direct luminaires will cut down glare from bare lamps.
(a)
Continuous row of luminaires form a "line of light"
Light from each section fills in beneath object to prevent shadow.
Long opaque object parallel to luminaire
Long opaque object normal to luminaire
Luminaire
Line shadow produced
(b)
(c)

FIG. 13-44. Fluorescent light techniques (a) Luminaires placed normal to the line of sight provide maximum shielding of lamps. (b) Luminaires run perpendicular to long opaque objects tend to eliminate shadows. (c) Luminaires run over and parallel to objects tend to create sharp line shadows.

2. Floodlight projectors, either open or closed units, for general illumination of large areas, for high levels around gatehouses, for particularly vulnerable positions, and for high-mounting long-distance projection.

3. Searchlights, high candlepower, narrow beam, with pilot-house controls mounted on sentry boxes and guard towers.

4. Symmetrical or elliptical angle reflectors or reflector lamps with flood or spot distribution for sign lighting, gates, doors, control stations, or other small areas.

The design of the exterior plant lighting arrangement is dependent upon the character of the installation to be illuminated. For close-up illumination, it is immaterial whether or not broad-beam floodlights or street lights are used. However, experience indicates that street lighting is usually more economical and efficient for general outdoor illumination than are the other types of installation. A row of modern 6,000-lumen enclosed street lights, spaced at approximately 125 ft and mounted at a height of about 23 to 25 ft will provide an illumination averaging 0.2 to

0.3 ft-c. in a strip between 60 and 100 ft wide. Reasonable estimating costs of street-lighting installations are $100 per 100 ft where poles have to be supplied or $60 per 100 ft where the units and aerial wiring may be mounted on existing poles.

Future Developments in Lighting Installations

Industrial plants present an infinite variety of processes, machinery layout, and materials to be handled, but these all combine with some modifications the same application of illumination to machines, bench work, inspection, office areas, etc. Each individual problem must be studied to determine the proper type of lighting system to be installed. To prove satisfactory, the system must be engineered on the basis of certain fundamentals, such as the purpose for which the lighting is needed, the physical characteristics of the area to be lighted, the illumination intensity necessary, and the expected results to be attained.

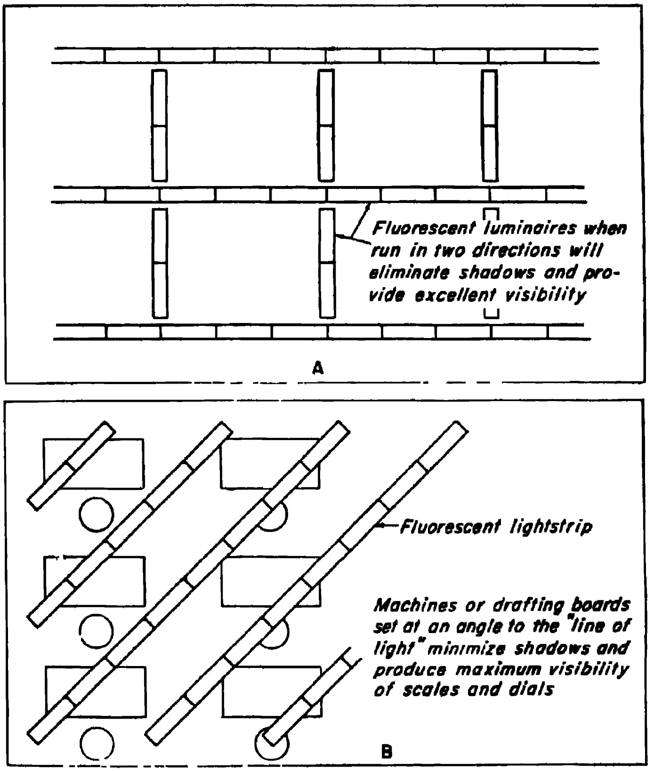

Fluorescent luminaires when run in two directions will eliminate shadows and provide excellent visibility
A
Fluorescent lightstrip
Machines or drafting boards set at an angle to the "line of light" minimize shadows and produce maximum visibility of scales and dials
B

Fig. 13-45. Two methods of installing continuous-row luminaires to produce maximum visibility results.

The days are past when any symmetrical pattern of outlet spacing would do and fixtures were chosen after construction was under way. The present trend is toward built-in lighting and functional lighting treatment from large-area low-brightness light sources by properly combining several lamps within a reflector or behind diffusing glass or other shielding material. Line of light, using fluorescent continuous-strip luminaires, will be most useful in developing lighting systems. The advantages of this system are illustrated in Figs. 13-44 and 13-45.

Figure 13-44*A* shows that, when fluorescent lighting strips are run at right angles to the normal line of sight, maximum shielding will be produced

Figures 13-44*B* and 44*C* show the necessity of locating the lines of light with reference to the machines and work in order to prevent harsh shadows.

Figure 13-45 shows two methods of installing continuous-strip luminaires to produce maximum visibility results.

Figure 13-45*A* shows a system with the luminaires strip running in two directions to form a grid system of lighting. This type of system, when properly located with reference to the machines and work, will almost entirely eliminate shadows.

When direct-type lighting is used in offices or drafting rooms, the fixtures should be equipped with louvers or lens plates to improve shielding. Figure 13-44*B* shows a scheme where the machines or drafting boards are set at an angle to the lines of light. This system will minimize shadows and produce maximum visibility of scales and dials.

There will be new lighting developments in the future. There is every indication that the present light sources and luminaire designs offer great opportunities for sound and constructive planning in the field of illumination. There will be little danger of obsolescence provided the intensity level is adequate for the fundamentals discussed under Illumination Yardsticks and provided these fundamentals are not ignored in planning the installation.

Combined light and power systems, using higher voltage lighting circuits, supplied from load-center distribution systems will provide a less expensive over-all electrical system for some types of industrial plants.

For incandescent lamps, 120-volt systems have been used almost universally. Higher voltage incandescent lamps are fragile and are generally not satisfactory. All fluorescent lamps have ballast in series with them. By combining a slight transformer action with the ballast, the fluorescent lamp can be operated on circuits from 120 to 265 volts. Standardized ballast ratings are 118, 208, 230, and 265 volts. For general lighting, the introduction of higher voltage fluorescent-lamp ballast has opened new avenues to higher voltage lighting circuits, thereby reducing the cost of the system which supplies the lighting load. The 460/265-volt three-phase four-wire grounded-neutral system can supply not only the 460-volt power but also the 265-volt power for general area fluorescent-lighting circuits. Fluorescent-lamp and starter voltages are essentially the same regardless of the voltage rating of the ballast. It is therefore believed that no sacrifice in safety is made with the higher voltage circuit operation, since in changing ballasts the circuit should be deenergized regardless of circuit voltage. Voltages as high as 260 should be used only in fixtures for general factory lighting where the mounting height is more than 8 ft. The National Electrical Code requires that all lighting for offices, individual local units for benches, machines, etc., shall be operated at a maximum nominal rating of 120 volts.

Amount of Light for Optimum Seeing

The main problem of illumination engineering has to do with long hours of close attention to more or less severe seeing tasks in the plant, shops, offices, and drafting rooms. Lighting intensity is only one factor to consider in the matter of good lighting practice. However, in a large majority of cases, improved seeing demands higher levels of illuminations for the visual tasks involved. Speed of vision is directly proportional to the ability to see detail. Light acts as a magnifier of detail. The eyes are thus greatly handicapped when they have an inadequate amount of light for ease of seeing, but the level required to produce maximum possible visual acuity—the ability to discriminate fine detail—is not usually required. This is especially so when the job to be done does not demand great visual effort. It may probably be quite unrelated to the actual seeing of detail. On the other hand, the fact that lighting of a given intensity is sufficient to enable a given task to be performed does not necessarily mean it c[illegible] be done easily. It is not correct to limit the considerations to mere momentary adequacy. When a job can be seen clearly, easily, and comfortably over the entire working period, then, and then only, can it really be done well.

It is not so much what your eyesight is, but what it can do, that counts. A century or so ago, when most work was done outdoors, a "normal" eyesight meant good vision at a distance. Today, much of the working world is under our noses, and normal eyes do not exist. Some work requires clear vision 8 in. from the face and other jobs demand extreme depth perception, color sensitivity, or lightning rapidity in far-to-near accommodation.

When someone asks "How much lighting is enough?" a flood of questions demand answers:

1. What are the ages, eye characteristics, and previous light education of those who work by the illumination?
2. What kind of work is involved? What is its size, color, background, etc?
3. Is the visual task a prolonged one?
4. What kind of performance is demanded? Are speed and accuracy vital factors?
5. Is the work to be viewed in a horizontal, vertical, or sloping plane?
6. What is the clarity of the atmosphere?
7. Are industrial hazards present, and what is their nature?
8. What humanitarian aspects of seeing, such as eyesight conservation and fatigue, shall be considered?
9. Are aesthetic or artistic effects, having to do with decorative functions, to be a governing consideration in the selection of the system?

Many installations supplying 50 to 100 ft-c are now in use with supplementary illumination of from 150 to 250 ft-c. However, a superabundance of light does not answer the purpose. It often adds to the difficulty of concentration. Lighting has suffered from the foot-candle concept or method of measuring the illumination level. Light output is measured in lumens, and the value of the illumination is merely the arrival of some of these lumens at some surface. The greatest value of light output for the least wattage is certain to result in objectionable features, such as excessive shadows and reflected glare. Many factors determine the adequacy of the installation.

Vision involves four fundamental elements:

1. The *task* of what one sees (not controllable)
2. The *eyes* with which one sees (indirectly controllable)
3. The *light* which makes it possible for the eyes to function (controllable)
4. The *surroundings* or the background against which the task or object is viewed (controllable to a degree)

Each of these is influenced by many other items. Providing light for seeing becomes a problem of controlling and coordinating the various factors outlined below:

1. Visibility requirements as determined by the character of visual task and the operation involved are
    - *a.* Size of the object to be seen
    - *b.* Time allowed for seeing
    - *c.* Contrasts in reflection factors
    - *d.* Average reflector factor
    - *e.* Color differences
2. Ability to see is influenced by the following external conditions:
    - *a.* Average brightness of surroundings
    - *b.* Size, brightness, and location of bright areas within the field of view
    - *c.* Brightness contrasts within the central field of view
    - *d.* Brightness within the surrounding field
    - *e.* Preexposure of the eye to brighter or darker surroundings

These modifying factors, which will be discussed under Illumination Yardsticks, are much more important in influencing the over-all appraisal of a lighting system than the absolute value of the illumination level. This is the reason why grades of illumination are referred to in Table 13-22 and Fig. 13-42. As noted before, there is substantially no difference in most cases between 4 and 6 ft-c, 10 and 15 ft-c, 20 and 30 ft-c, or 50 and 70 ft-c, etc. Brightness relationships, rather than high lighting intensities, determine good seeing conditions. For this reason it is necessary to study the various visual tasks and their background relationship for proper lighting results. In most industrial plants, the assortment of machines varies; the operations, designs, or processes change; and differences exist in tolerances and degree of inspection. Therefore, the lighting factors involved, even for industries in like process groups, do not always remain the same. Furthermore, a complete lighting system that will prove entirely satisfactory cannot be supplied on a blueprint unless the designer has an intimate knowledge of the processes and their operational requirements.

As a basis for selecting the lighting intensity, the most trustworthy information about minimum required recommendations is contained in lists approved by the American Standards Association. These lists give values of minimum foot-candles in service measured on a horizontal plane 30 in. above the floor.

Thus all that seems necessary is to spot a sufficient number of outlets on the plans and specify light sources large enough to produce the intensity recommended. The problem isn't so easy. There are many items encountered in most industrial processes which will lead to unsatisfactory lighting results unless they are given proper consideration. The most important are

1. A large percentage of the work or surfaces to be seen in industrial plants, such as gauges, verniers on machines, etc., are vertical or inclined, rather than horizontal.
2. In a room full of equipment and material, the light from a general lighting system is very likely to be blocked by interference and sponged up by the "surround," expiring without reaching the critical work surfaces.
3. Lighting requirements for industrial plants are different for each type of plant even though the same natural laws and engineering principles are effective.

A ray of light falling straight downward on a vertical surface produces no light on that surface. Increasing levels are obtained as the slope of the ray increases up to 35 deg from the vertical. As the ray slopes still more, the amount of vertical illumination drops because of the shrinking effect of increasing distance. With most artificial lighting systems, the lighting level on a vertical plane is seldom greater than one-half that produced on a horizontal plane at the same location. In the majority of cases it is much less. As the manufacturing art advances, machinery becomes more complex, and workmen must react more quickly to every change that occurs in their work or their environment. Progressively better lighting is required with every advance in the technique or speed of production.

When a person enters or when a machine is brought into a room, the illumination falls immediately. The degree of this decrease depends on the size, shape, and color of the objects in the room. In one recent machine-shop installation the illumination measured in an empty room was 45 ft-c, but when the machines were installed the illumination dropped to 33. When illumination on vertical surfaces is important, special consideration should be given to the fixture locations, and luminaires should be chosen that will give distinctly higher candlepower in directions nearer the horizontal than the vertical axis.

A symmetrical system of general lighting for industrial areas is very likely to be full of obstacles and to produce unsatisfactory lighting results. In a large empty room or one where the machines are low in relation to the work positions, many lighting units can contribute to the work. When machines that block the light are installed, only a few luminaires produce light that reaches the work. A recent check of a drafting-room installation which was advertised as a 50-ft-c job showed that the useful illumination on the vertical-type drafting boards, inclined to a very steep fixed angle, was less than 15 ft-c. The "in-service" lighting level will be much lower than the calculated value unless it is especially designed to suit the work position, with proper consideration given to fixture location and luminaire selection, so that the light liable to interference is controlled.

The worker's eyes do not see foot-candles unless they are looking directly at the light source. Reflected light, and not incident light, is what reaches the eyes and causes objects to be visible. Reflected light is measured in terms of brightness, the unit for which is the foot-lambert.

The illumination necessary to reveal an object is subject to more or less accurate determination, provided the following conditions can be correctly stated:

1. Size of detail
2. Distance from eye
3. Difference in reflection factor between the detail and its immediate surroundings

Figure 13-46 is a nomograph of the illumination required at the viewing angle for a seeing task when the above conditions are known. For example, a detail 8/1,000 in. in size, when viewed at a distance of 15 in., will subtend an angle of 1.83 ft (log: visual angle = 0.26). With a difference in reflection factors of 0.4, the illumination required will be approximately 21 ft-c.

Again, most problems cannot be solved so easily. Size of the detail to be seen and the difference of reflection factors may present quite a problem. For example, in the simple process of threading a needle, one does not only have to see the eye of the needle and the thread, but one must see when the thread is nearly in the center of the eye. This problem, like many other cases in industry, may present a more difficult task than the original one.

Up to the present the most serious handicap that the technical man has been under is the fact that there is no universally agreed-upon method of assessing the difficulty of a seeing task or of estimating how tired the performance of a task has made the person performing it.

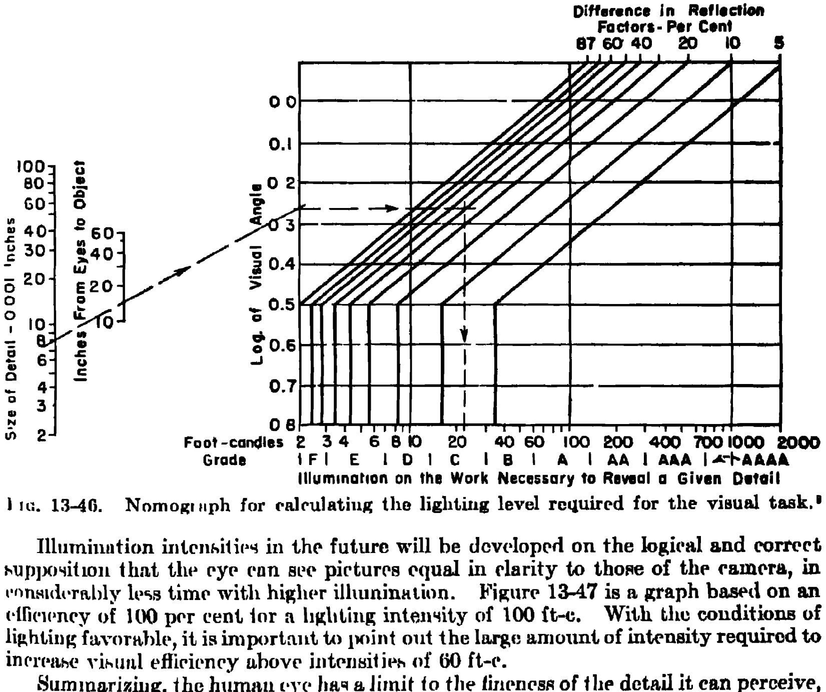


FIG. 13-46. Nomograph for calculating the lighting level required for the visual task.[9]

Illumination intensities in the future will be developed on the logical and correct supposition that the eye can see pictures equal in clarity to those of the camera, in considerably less time with higher illumination. Figure 13-47 is a graph based on an efficiency of 100 per cent for a lighting intensity of 100 ft-c. With the conditions of lighting favorable, it is important to point out the large amount of intensity required to increase visual efficiency above intensities of 60 ft-c.

Summarizing, the human eye has a limit to the fineness of the detail it can perceive, and it is impossible to go below that limit without optical aids to vision. To approach the limit requires an illumination of 50 to 60 ft-c under ideal seeing conditions, and many tasks can be made substantially easier by intensities up to 1,000 ft-c.

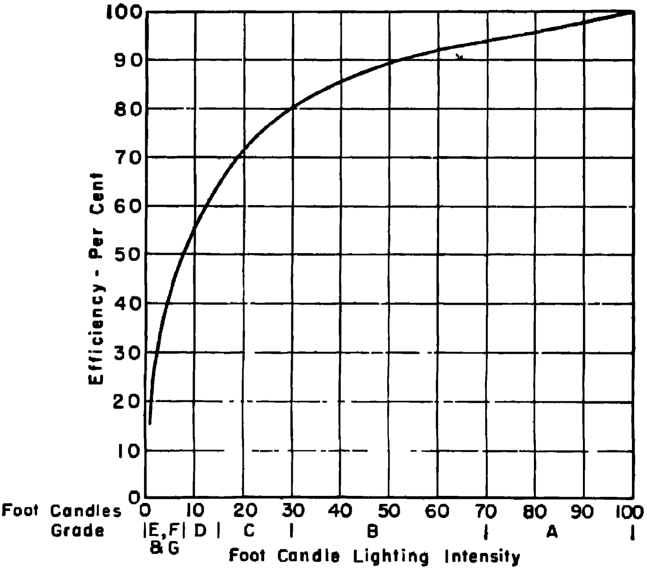
The most important facts concerning the level of lighting intensities are listed below:

1. From a study of the color-response characteristics of the human retina it might be concluded that the yellow color, such as produced by the sodium-vapor lamp which has all of its luminous output concentrated at the 5890 Å wavelength, would be the best for general-purpose lighting. A study of this and other monochromatic lamps shows that they do not meet the broad criteria of satisfactory interior illumination.

Man's seeing judgment depends to a large extent upon the color of the object being viewed. Monochromatic light greatly distorts colors, and any effect in favor of such light may be completely counterbalanced by the deleterious psychological effects upon individuals especially the effects of those colors which change the appearance of the skin.

2. The eye automatically and unconsciously adapts the size of the pupil to the brightness of the field of view; the aperture increases with a decrease in field brightness and decreases with an increase in field brightness.

3. A significant factor in most efficient lighting is the time available for seeing. The workmen's eyes, like a camera, can take "snapshots" when the illumination is high, while with insufficient light it must take "time exposures." Thus, where workmen must react quickly to changes that occur in their work or environment, progressively better lighting is required for every advance in the technique or speed of production.



This chart shows the relative efficiency of an average human eye based on its ability to perceive under various levels of illumination under the best type of lighting. (Uniform and diffused with no shadows, no high lights, no glare and no reflection such as that reflected from a cloudless northern sky).

Fig. 13-47. Lighting intensity vs. visual efficiency.

4. Both speed of vision and visual acuity increase sharply (Fig. 13-47) up to 50 t 60 ft-c, while both increase at a lower rate of light intensities above 60 ft-c. Some gain is realized in higher levels of lighting, probably up to 1,250 ft-c, provided the surroundings are adequately illuminated. The gain for better lighting is greater for people who have eye defects. A decline of visual efficiency below 20/40 approache serious proportions.

5. High intensities of artificial illumination, particularly incandescent lighting, have so much invisible heat energy associated with them that cooling systems becom necessary.

6. Varying brightness contrasts are a much more effective factor in seeing than in increasing the intensity of the illumination proportionally upward.

7. Unless the worker is looking directly at the light source, his eyes do not see foot-candles (incident light) but foot-lamberts (reflected light). Objects are seen by light reflected from them, and in some cases, a very high average foot-candle level

actually reduces the ability to see his work. Because of reflection-factor differences, the brightness of the surroundings may be greater than the brightness of the work.

8. The average flux method commonly used in calculating depreciated illumination is

$$\text{Ft-c} = \frac{\text{total lumens} \times \text{utilization factor} \times \text{depreciation factor}}{\text{floor area}},$$

It cannot be successfully used in a large percentage of industrial applications. It may be used as a basis for calculating the average lighting level on a horizontal plane throughout the entire area, but it tells nothing of the difference in lighting levels between various locations or how much light is actually delivered to the work position, particularly where it is inclined, vertical, or underneath a projection of the machine or workpiece. It is necessary to go much further than providing mere illumination; study of what is being illuminated and the method to be employed must be coordinated with the straightforward engineering problem of illumination.

9. The only real test of a lighting installation is by working at the job for which it was designed. Quality of lighting is often equal in importance to quantity of lighting.

10. One of the most salient facts concerning the relationship of light to sight is the change the human eye undergoes during its life. The pupil of the human eye contracts with age, reducing the amount of light that reaches the retina. This is not a disease but is a condition incident to age. There are general physical changes throughout the whole body between the ages of 40 and 50. The result is a lowering of the muscular tone with the eyes showing a slight blurring of near vision. No special treatment or use of glasses for close work is necessary. However, in order to counterbalance this condition and to maintain the same brightness of the retinal image, the level of illumination should be increased about 25 to 50 per cent.

11. Ordinary clear crown glass used in eyeglasses and safety goggles absorbs at least 10 per cent of the reflected light. Their use will, accordingly, require corresponding increases in lighting intensities.

12. The Illumination Engineering Society Lighting Handbook, published in 1947, recommends 5 ft-c minimum for very easy seeing tasks, 10 ft-c for casual seeing tasks, 30 for ordinary and 50 for difficult seeing tasks. Very difficult tasks may require up to several hundred foot-candles.

Illumination Yardstick—Complaints and Factors That Affect Lighting Results

The purpose of the following material is to discuss artificial-lighting conditions in connection with seeing that will help the eyes as much as possible in the performance of their allotted tasks. Any specification of the most desirable conditions must be related to ideal requirements. The fact that some of the conditions so specified may be impracticable or economically impossible for certain types of installations does not alter the fact that they are desirable. Almost always the ideal that is talked about today is made technically possible only tomorrow.

There is no perfect lighting system: every type, even natural daylight, possesses some fault.

With respect to artificial lighting, the most important factors in determining the lighting results are *comfort* and *see-ability*. Installations that become obsolete quickly are principally those in which the lighting is uncomfortable and/or inadequate for the job. Uncomfortable lighting causes much greater eyestrain and fatigue than an insufficient light-intensity level. The factors that tend to make a lighting installation comfortable are not engineering and physics alone but a branch of applied psychology. It is comparatively easy to design a lighting installation for efficiency, somewhat more difficult to design an installation for aesthetic effect which will please a given group of workers, and very difficult to design any type of installation which a given group will at once consider *comfortable*.

The reason for this is that comfort depends upon personal preferences which are influenced, to a large degree, by habit and previous experiences. Lighting is for people to see by. All humans do not react alike under fixed conditions, nor do they have the same likes and dislikes. Comfort is, therefore, a subjective affair and is not predictable; however, for the purposes of this discussion, it will be defined as freedom from any objectionable sensations and painful, disagreeable, or irritating conditions.

What are these conditions that are uncomfortable and objectionable in the over-all results obtained in a lighting system? The principal causes of inadequate lighting and their relation to physical reactions are given in Fig. 13-48. As a reminder that seeing is an activity of the entire human being, it has been estimated that one-quarter of our bodily energy is used in seeing. When adequate lighting levels are available for the seeing task, other factors upon which visibility and ease of seeing depend are too commonly neglected to make the most of the light that is available. Where the illumination level is adequate, the distribution of light becomes the most important feature for comfortable and efficient visual conditions.

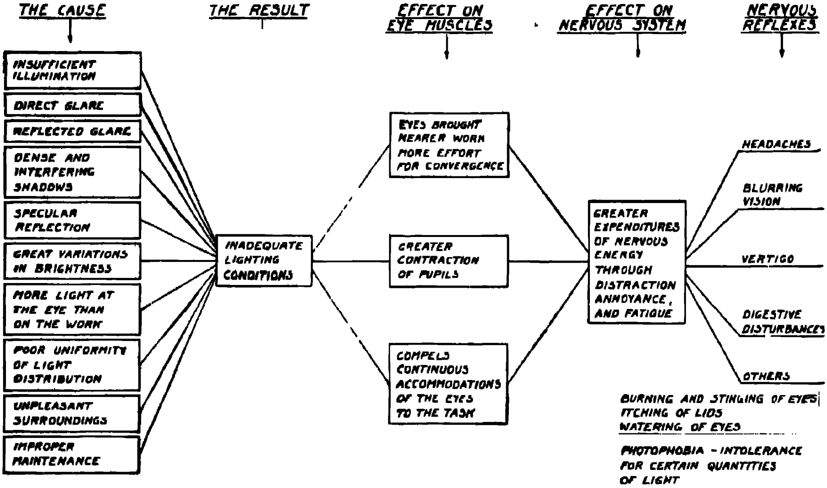

THE CAUSE
THE RESULT
EFFECT ON EYE MUSCLES
EFFECT ON NERVOUS SYSTEM
NERVOUS REFLEXES
INSUFFICIENT ILLUMINATION
DIRECT GLARE
REFLECTED GLARE
DENSE AND INTERFERING SHADOWS
SPECULAR REFLECTION
GREAT VARIATIONS IN BRIGHTNESS
MORE LIGHT AT THE EYE THAN ON THE WORK
POOR UNIFORMITY OF LIGHT DISTRIBUTION
UNPLEASANT SURROUNDINGS
IMPROPER MAINTENANCE
INADEQUATE LIGHTING CONDITIONS
EYES BROUGHT NEARER WORK MORE EFFORT FOR CONVERGENCE
GREATER CONTRACTION OF PUPILS
COMPELS CONTINUOUS ACCOMMODATIONS OF THE EYES TO THE TASK
GREATER EXPENDITURES OF NERVOUS ENERGY THROUGH DISTRACTION ANNOYANCE, AND FATIGUE
HEADACHES
BLURRING VISION
VERTIGO
DIGESTIVE DISTURBANCES
OTHERS
BURNING AND STINGING OF EYES
ITCHING OF LIDS
WATERING OF EYES
PHOTOPHOBIA - INTOLERANCE FOR CERTAIN QUANTITIES OF LIGHT

Fig. 13-48. Common causes of inadequate lighting and their relation to physical reactions.

Table 13-25 lists the common causes of unsatisfactory lighting and suggests remedies for correcting these conditions. In suggesting remedies or specifying adequate lighting conditions, there are two main factors which must be considered, viz.:

1. The immediate field of vision for the task at hand
2. The change in the effects of the surroundings with the passage of time

With respect to the immediate field of vision, research has indicated that visual acuity, contrast sensitivity, and perception of motion are at their best when the critical detail is in the center of the field of vision and surrounded by a uniform field of about the same order of brightness as the center field.

With respect to effects of surroundings influencing seeing conditions over a period of time, it is important that variety of the right kind be present if desirable results are to be obtained. In industry there is plenty of variety, but unfortunately it is usually the wrong sort where differences of contrasts and brightnesses cause distraction. Color has such an influence on seeing conditions, it is necessary to consider both decoration and lighting together as an interdependent pair. The important part that color can play in creating a successful seeing system will be discussed under Relation of Light and Color.

In examining the causes and remedies for uncomfortable and unsatisfactory lighting conditions as given in Table 13-25, it is essential to note that control of light is of primary importance not only where it leaves lighting equipment but also where it is reflected from surfaces.

Table 13-25. Factors Affecting Lighting Conditions

Cause of uncomfortable and unsatisfactory light	*Remedy*
. Insufficient illumination on the work for the seeing task involved. Remember that brightness relationships are more important to good seeing conditions than the highest foot-candles for the least wattage.	1. Sometimes a greater illumination level may be the only agency to make one see enough. It takes about four times as long to perceive small detail under 2 ft-c than it does under 100 ft-c. For a 1:10 ft-c ratio the time is about twice as long. See How Much Lighting Is Enough for Optimum Seeing.
. Direct glare caused by bright light in the range of vision. Glare results at any and all intensities where unsupported by proper levels of total field brightness to stabilize adaptation. The eye receives clear impressions of detail only in the absence of glaring light. Glare is caused by too much light in the worker's eyes compared to that on the work. Discomfort from a light source of fixed brightness increases with its visible area and, vice versa, the comfort increases with a reduction of source brightness.	2. *a.* Shield lamps properly. Avoid bare lamps in continuously occupied areas. *b.* Mount lighting fixtures as high as practical. The higher the mounting, the less the glare effect. If the mounting height is doubled, three or four times the brightness of the source may be tolerated. *c.* Increasing lighting intensities helps, but the foot-candles must be increased 10 times to stand doubling the brightness of the light sources. *d.* Best solution is to have a low-brightness broad-distribution light source. Comfort from a light source of fixed brightness increases as its visible area becomes larger. *e.* Make the lighting installation as inconspicuous as possible. Glare effect is multiplied by the number of lighting units visible from any work position. What may be suitable for a small office may be very uncomfortable in a large room.
Direct daylight glare caused by improper placement of work with respect to sunlight.	3. *a.* Properly shade windows. *b.* Shift working surfaces in such a way that the natural light falls on the working plane and away from the worker's eye.
Reflected glare caused by reflections from polished metals, glossy paper, or any material possessing a sheen.	4. *a.* Position light sources so as to keep reflections away from the eyes. *b.* Again, the best solution is a large-area low-brightness source.
. Reflected daylight glare caused by natural light that bounces from the work plane to the eyes.	5. *a.* Shade the source. *b.* Provide a high enough artificial-lighting-intensity level to offset reflected daylight.
. Specular reflection from machine surfaces which creates direct or diffused images of the light source on the work, polished furniture, and other shining surroundings	6. *a.* Reflections may be reduced by increasing the general level of illumination with low-brightness light source. *b* May be corrected sometimes by changing the mounting height of the offending units. *c.* Avoid the use of bare lamps.
Dense shadows caused by a workman on piece of equipment cutting off the light	7. Large, closely spaced units or continuous rows shine around objects and erase shadows.
Range of brightness within the visual field is too great. While the eye focuses on objects directly before the eye, it does a partial job of seeing masses well to the side of the direct line of vision.	8. *a.* Avoid competition between the nerves in the eye by keeping the brightness range between the work and the surroundings within reasonable limits—5:1 if possible. *b.* Splash enough light on the side walls to hold a ratio of at least 10:1 or better. May look a bit inefficient by calculation but worth much more psychologically.
. More light at the eyes than on work caused by improper placements of light source.	9. *a.* Angle lighting may cure bright surface jobs. *b.* Vertical or horizontal local lighting may solve some inspection problems. *c.* Install local lighting where needed.
Unpleasant surroundings, dingy, gloomy atmosphere. You get fooled by uncomfortable illumination even if the lighting intensity is adequate. Lighting cannot be evaluated by simple seeing.	10. *a.* Where the ceilings serve as a secondary light source, such as in semi-indirect or indirect lighting, the ceiling should be finished a flat white. Avoid the use of glossy paints because the shiny surfaces cause specular reflections of light sources. *b.* Walls should be colored because they are usually the background against which the task is viewed. Finishes will be discussed under Relation of Light and Color. *c.* Floors should be light—the lighter the finish of the floor covering, the higher the

Table 13-25. Factors Affecting Lighting Conditions (*Cont.*)

Cause of uncomfortable and unsatisfactory light	*Remedy*
	efficiency of the lighting system. The reflected light back to the ceiling helps relieve brightness contrasts between the luminaires and the ceiling. *d.* Desk tops should be dull finish and light in color. *e.* Plate glass should never be permitted to be placed on any desk top.
11. Improper maintenance. Lamps age and blacken and give out less light. Dirt and dust reduce reflecting and transmitting qualities of the fixtures and reflecting surfaces.	11. Five musts for proper maintenance. *a.* Clean lamps and fixtures frequently. *b.* Repaint walls and ceiling regularly. *c.* Replace burned-out lamps promptly. *d.* Maintain proper circuit voltage. *e.* Establish a definite maintenance program. Very few plants spend enough on lighting maintenance. Keeping the lighting system in best condition more than pays for itself. This will be discussed further under Economic Factors.

Summarizing the information given in Table 13-25, the basic considerations according to which lighting should be designed and criticized are as follows:

1. There should be enough light, and it should be suitably distributed on the surface to be seen.
2. There should be no glare which causes discomfort to the eye and affects its ability to see.
3. Any shadows should be agreeable, revealing rather than hiding. We depend upon shades and shadows to show form of solids. Any shadow that does not reduce the illumination more than 20 per cent will hardly be noticeable. One that causes more than a 50 per cent reduction exceeds good practice.
4. The surrounding field should be uniformly bright and equal to or less but never more than the central field brightness. Always have more light on the work than at the eye.
5. The central and surrounding visual field should be so colored as to give a pleasing sense of variety, while at the same time holding the brightness levels within reasonable limits—10:1.
6. The color of the light should be suited to the object illuminated by it. Be critical about colors of interiors. When color judgment is an important consideration, do not permit personal preferences to dictate selection.
7. The lighting installation should be as inconspicuous as possible or designed to contribute materially to the attractiveness of the environment.
8. The lighting installation should be as comfortable, efficient, economical, easy to maintain and as flexible as conditions permit.
9. Comfortable lighting is always important. Disturbing reflecting brightnesses from machines and processes are often encountered even though the product itself may be dull finish.

One of the salient features which must be recognized in industrial lighting is the relationship between the lighting of the work and the lighting of the surrounding areas. The lighting of the surrounding areas may often prove irritating or annoying even when one is not looking directly at them. The nerves of the eye compete with each other. While the eye focuses on, and sees well, objects directly before the eye, it also does a partial job of seeing masses well to the side of the direct line of vision.

Nerves in side walls of the retina warn you of the presence or approach of objects from the side. One may have to turn his eyes to recognize the object, but without turning, one is conscious that something is there. If the light in the side areas is strong, the side nerves of the eye will be calling for a small pupil aperture to restrict the admission of light. If, at the same time, the direct object of vision is not well lighted, the main object nerve will be calling for a large aperture. These contradicting demands tire the eye and make you restless to have the triangle solved.

How much energy is actually wasted when one attempts to work or play in anything but comfortable and adequate lighting is graphically illustrated by the following example.

Suppose you drive an automobile on a bright sunny day over a 50-mile stretch of straight road without heavy traffic. At the end of the ride you notice no particular exhaustion. But take the same automobile, the same road, and make the same drive at night in a fog. After 50 miles of this you know you have done some work. You have gripped the wheel, tensed your muscles, strained your whole body, and used up a tremendous amount of energy trying to see. The only actual difference between the two cases has been the amount of light available for seeing.

Seeing conditions affect employee efficiency and morale. **The benefits claimed for good seeing and lighting conditions are**

1. They enable one to see faster and do more accurate work.
2. They make seeing easier and minimize eyestrain and fatigue.
3. They improve labor conditions, encourage the use of floor space, and promote cleanliness and safety.

The wide range of effects associated with seeing are not an arithmetical summation of a series of different effects. Some of the advantages claimed for good lighting are supported by scientific evidence as discussed above, while others can be justified by logical inference and practical experience.

Some of the things the individual can do toward making the seeing tasks easier are listed below:

Avoid Glossy Finishes. *Many people needlessly abuse their eyes by insisting on plate-glass tops for their desks.* In practically every office where the occupants complain of inadequate or uncomfortable lighting there will be found high local brightnesses that are inevitable with the use of highly polished or glossy surfaces. A beautiful mahogany desk with a glass top is as bad as could be found; yet the occupants rebel at the recommendation that the glass be removed and the furniture be finished a light flat color to help the visual task. In the plant, also, light finishes on the machinery, the walls, the ceiling, and the floor can play an important part in making the visual task much easier. (See Relation of Light and Color for further discussion.)

Avoid Abuse of the Sun on the Eyes. Sunlight is very beneficial to the eyes and health, but it is necessary to know how to use sunshine to get the most out of it. Abuse of the sun on the eyes may cause great damage. No matter how suddenly you are plunged into darkness, there is no pain, but time must be given for the pupil to enlarge sufficiently for the low-intensity level. The converse happens when you go out into strong sunlight from an area where the lighting level is low. There is a feeling of eyestrain because the decreasing of the pupil (the drawing together of the shutter in its attempt to protect the eye against sudden brilliance) requires considerable time. The sudden contraction of the iris muscle is painful.

If on going out into bright light the eyes are directed downward for two or three minutes, while the size of the pupil is changing, the eyelids and eyelashes will protect the eye from the excessive light until accommodation is complete. Following this simple procedure, one may step from comparative darkness to bright light without the slighest discomfort.

The eye can be strengthened in its light tolerance by judicious exposure to light.

Avoid Wearing Sunglasses Except When Necessary. The promiscuous wearing of dark glasses has become a fad in recent years. As useful as sunglasses are, they should be worn only when necessary. They are needed when eyes are exposed to bright direct sunlight or reflected light from highways, pavements, water, or sand. They are not necessary indoors, in the shade, outdoors on a cloudy day, or for driving a car at night.

It is also important to note that dark glasses protect the eye only from visible and ultraviolet rays. The ultraviolet rays can be harmful to the eye and cause a painful inflammation of the outermost layers of the eyeball (conjunctiva or cornea). Sunlight also abounds in infrared or heat rays. The eye itself acts as a strong convex lens with the result that the infrared rays are focused on one spot in the eye and can cause "eclipse blindness" if the direct rays of the sun are viewed for some time. These rays are not held back by dark glasses to the same extent as visible and ultraviolet rays.

The preferred type of sunglasses for beach wear are ones having green or bluish green lenses. These colors absorb both the short ultraviolet and some of the infrared rays.

Smoke- or flesh-colored lenses are the best type for people who are exceptionally sensitive to the dazzle of reflected sunlight. These diminish the quantity of light more or less uniformly over the entire range of the spectrum.

When sunglasses are not necessary, take them off. Wear them only when the eyes are exposed to bright light or reflected glare.

Avoid Too Many Motion Pictures in Improperly Lighted Theaters. The bright motion-picture screen amid the very dark surroundings in most theaters presents very poor and unsatisfactory seeing conditions. Looking at a motion picture is a trying task even for normal eyes because of abominably high brightness ratios of 1:1,000 and higher on a relatively small bright area against a large black background. If viewing the screen were really a critical seeing task, it would be unbearable owing to the very unpleasant and trying contrasts.

It is possible to supply some light to the immediate surrounding without affecting the desirable contrasts in the picture itself. Seeing would then become easier and more effective. It is time the motion-picture industry realized the very poor seeing conditions usually encountered in most theaters and changed to lower brightness contrasts.

Establish Good Reading Habits. Despite the fact that today we have large better print, blacker ink, better paper, and finer illumination, the number of people who experience discomfort when they read is greatly increasing. The cause for this is not always inadequate or uncomfortable lighting conditions. Most people form bad reading habits such as

1. Reading with the body in lying or a strained uncomfortable position. Sit properly and hold the book toward the eyes. The proper distance for reading is 14 to 16 in.

2. Reading when sick or very tired and when one should be sleeping. While reading is a relaxation compared to most activities, it is no substitute for sleep and rest.

3. Reading when one is tense from hurry, fear, or worry. Lack of relaxation tenses the muscles, pulls the eye out of perfect focusing, and interferes with the normal functioning of the brain.

4. Reading to excess when it is not possible to have the print reasonably stationary, as on a train or in an automobile. The eyes under these conditions are subjected to continual fixation and a tremendous reduction in clarity of sight.

5. Reading from highly glazed paper, colored paper that is printed with contrasting ink, or other blurred or poorly printed material.

6. Skimming over a page will cause trouble because the eye is used when only partly focused, and if skimming is followed to excess, the ability to produce a complete focus will be lost, and dim or blurred vision will result.

7. Do not stare. If you have formed a habit of looking too fixedly at objects, begin to blink naturally. It does not interfere with vision and actually increases the amount of time you may actively see.

All functions of the body and mind are performed through the use of nervous energy. When they are done easily and normally, the amount of energy consumed is small. To read to best advantage avoid the bad reading habits listed above, and arrange the light so that it is adequate and comfortable.

Arrange the Office to Make the Best Use of the Lighting. Arrange the desks and working surfaces wherever possible so that (1) the fixtures are positioned with the lowest brightness area of the lighting equipment exposed to vision, (2) the reflection of light from surfaces in the work area is directed away from the eyes, and (3) worker does not face a window directly. When the eyes are adjusted to indoor intensities outdoor glare is extremely uncomfortable.

Do Not Blame All Ocular Discomfort on the Lighting. It is erroneous to conclude that all ocular discomfort and headaches are caused entirely by poor lighting conditions. Any person who is healthy, has been trained to see properly, is normal in all physical and psychological aspects, and who has normal or properly corrected vision is not likely to experience prolonged discomfort unless the lighting level is very inadequate and/or the over-all lighting conditions are very poor. There are over 20 well-known conditions in the body that may cause headaches. Ocular discomfort may result from any one or combination of the following:

1. General fatigue from any cause, physical or psychological, such as improper diet, hunger, muscular tiredness, physical defects, debilitating effect of noises, poor posture, etc.

2. Physical disorders, sinus and kidney trouble, venereal diseases, allergy to certain food, etc.

3. Emotional disturbances, such as excitement due to overwork, mental depression, nervous breakdown, etc.

The above causes of ocular discomfort are just as common under natural lighting as artificial lighting. Poor lighting conditions can and do contribute toward the difficulties by reducing the resistance and thereby increasing vulnerability.

The important thing to remember is that many conditions may cause periods where the eyes tire easily and feel strained without having very much wrong with the lighting system or anything permanently wrong with the eyes. At times, lighting and the eyes do cause headaches but not nearly so often as is generally supposed. See your physician before the oculist.

Some Things That the Architect and Engineer Can Do Toward Making the Visual Task Easier. The architect and engineer can make the visual task much easier by avoiding the use of

1. Long narrow rooms
2. Glossy paint on ceilings and side walls of room
3. "Hammered"-glass partitions and skylights
4. Glass-block walls
5. Decorations that are dark in color
6. Machine finishes of high-gloss black enamel or low-contrast colors

Facts Concerning Factors That Affect Lighting and Seeing. Some of the most important facts concerning factors that affect lighting and seeing are listed below.

1. Nature in her kinder moods provides ideal lighting conditions. The best light is uniform and diffused, such as that reflected from a cloudless northern sky. With this light there are no shadows, no high lights, no glare, and no reflection.

2. Eyestrain increases as the distance between the eye and the object decreases.

3. It has been stated that a decrease of illumination from 100 to 1 ft-c triples the fatigue.

4. Direct rays less than 20 to 30 deg from the line of vision are especially likely to create glare. The discomfort effect of a glare source is determined chiefly by the relation of its brightness to that of the background.

5. Glare causes a decrease of visibility. The effect of constant glare is to waste light—approximately 85 per cent when the glare source is 5 deg from the line of vision, 75 per cent for 10 deg, and 60 per cent for 20 deg.

6. When the vertical plane is of chief interest to the workers, it is impossible to produce a high illumination relative to that produced on the horizontal plane at the same point by means of general lighting *only* without undue glare.

7. Factors that affect glare are as follows:
   - *a. Area.* Area increases in direct proportion to the area of constant brightness source.
   - *b. Brightness.* From a comfort standpoint, doubling the brightness of the source is about the same as quadrupling its area.
   - *c. Distance.* For a constant size of glare source, the resultant glare will decrease inversely as the square of the distance. It also appears as though the effect of reflected light varies inversely as the square of the mounting height.
   - *d. Angle.* Veiling effect varies inversely as the square of the angle between source and line of vision.
   - *e. Surroundings.* Glare varies inversely with the relative brightness of surroundings raised to the 0.6 power.

To summarize, the discomfort glare varies according to the following formula:

$$\text{Glare factor} = \frac{A \times B^2}{D^2 \times L^2 \times S^{0.6}} = \frac{A \times B^2}{D^2 \times L^2 \times F} = \frac{(CP)^2}{SA \times D^2 \times L^2 \times F}$$

where A = apparent area of source, sq in.
B = brightness of source in foot-lamberts ÷ 1,000

D = distance in feet from source to eye + 10
L = angle in degrees above line of vision ÷ 10
S = surrounding brightness in foot-lamberts ÷ 10
F = foot-lambert factor
CP = candlepower in direction of the eye

8. Correct provision of artificial seeing conditions can be provided only by coordinating the various diverse effects so that the job stands out well and has a pleasing appearance.

Relation of Light and Color

Industrial-plant lighting should be designed, installed, and maintained to meet the following objectives:

1. To make the work on the eyes easier.
2. To provide more and better light to industry for the least expenditure of labor and materials.
3. To improve morale among workers by enabling them to work with speed and precision without eyestrain and with less nervous tension This is especially important for those tasks which must be performed continuously over prolonged periods of time.

It is being recognized more and more that seeing is not entirely benefited by greater and greater light intensities. The lighting level is important, but seeing also depends upon other conditions for perception and visibility. The general lighting system is therefore not limited to the installation of light sources alone. Color is important to everyone who sees, and the surfaces on which the light falls are very significant. Everything seen is subjugated to two fundamentals—brightness and color. When light strikes an object, some wavelengths may be absorbed, some may be transmitted, and some are reflected. The rays that are reflected constitute the color we see.

Light, sight, and paint are the primary tools for seeing. Paint used to cover walls, ceiling, floors, and machinery has an influence on seeing conditions and is so important that planning a set of conditions for seeing should include both lighting and painting taken together as an interdependent pair. Coupling scientifically applied colors with adequate lighting can solve many vision problems that neither one method nor the other could successfully solve by itself. It may at first sound presumptuous for the electrical engineer to be involved in the specifications for the painting of the space, equipment, and machines he has been asked to light. But the use of color has been found to be of such an immense value in giving clearer vision and preventing eye fatigue, it is the only logical procedure if the seeing conditions are to be made as good as possible.

It is not suggested that adequate seeing conditions will make the work easy and eliminate the need for effort; they will, however, so influence the workers in industrial plants that they will be stimulated in the performances of their tasks and will maintain better their rate and quality of work over the entire working period.

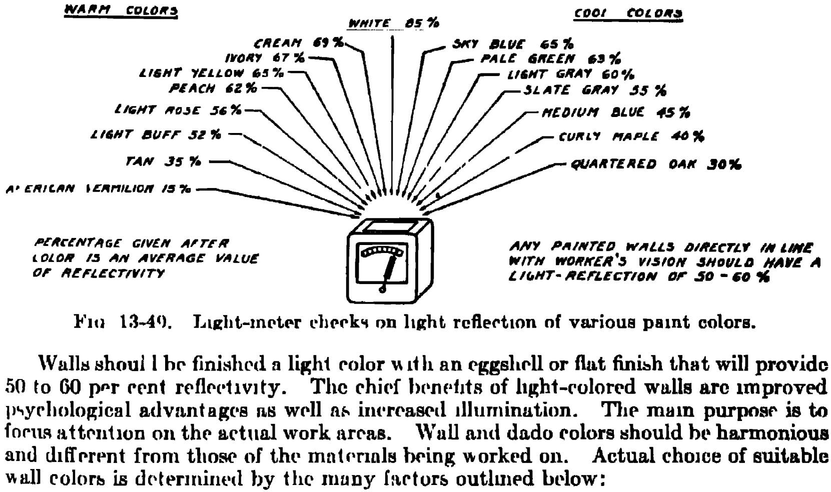
When considering the subject of industrial-plant painting, the aim is more than upkeep maintenance. The basic reason for painting any plant interior should be to realize the most out of both natural and artificial light. Painting in industrial plants is not interior decoration. The use of color is to establish seeing conditions that automatically make the seeing task easier and avoid distraction. Other advantages gained by proper selection of colors are

1. It improves the appearance of the plant.
2. It aids sanitation and "good housekeeping."
3. It reduces fatigue and absenteeism.
4. It cuts accident toll and reduces spoilage and rejects.
5. It fosters better employee-management relationship.

Three guiding factors in coordinating lighting and painting are reflectivity, finish, and color. The reflectivity of various paint colors is given in Fig. 13-49. It has been shown that coverage is an important factor in that thin coats of paint have much lower reflectivity than heavy coats. Recommended coverage is usually not more than 460 sq ft per gal.

Generally flat or eggshell finishes are desirable over glossy surfaces. The former will result in better distribution of light and brightness, while the latter cause specular reflections from the light sources and annoying glare when in line of vision. All the advantages of interior industrial painting generally stem from the fundamental fact that properly painted ceilings and walls greatly increase the lighting efficiency, comfort, and see-ability.

White paint reflects more light than any other color, and for this reason most commercial and factory interiors should have the ceiling and upper walls painted with white to provide a high reflection factor of 80 per cent or more. (The exception would be those rare cases where the location of the operator's work forces him to look directly upward.) Light ceilings are essential where indirect or semi-indirect lighting installations are made because the ceiling is a secondary reflector. In all types of lighting installations, the high reflection factor of white ceilings will increase the illumination, provide light diffusion, reduce shadows, and, in general, improve seeing comfort.



PERCENTAGE GIVEN AFTER COLOR IS AN AVERAGE VALUE OF REFLECTIVITY

ANY PAINTED WALLS DIRECTLY IN LINE WITH WORKER'S VISION SHOULD HAVE A LIGHT-REFLECTION OF 50 - 60 %

FIG 13-49. Light-meter checks on light reflection of various paint colors.

Walls should be finished a light color with an eggshell or flat finish that will provide 50 to 60 per cent reflectivity. The chief benefits of light-colored walls are improved psychological advantages as well as increased illumination. The main purpose is to focus attention on the actual work areas. Wall and dado colors should be harmonious and different from those of the materials being worked on. Actual choice of suitable wall colors is determined by the many factors outlined below:

1. *Physical Dimensions (Size and Height) of the Area.* In small areas, walls assume greater importance than in larger areas. They should be painted in such a manner as to provide variety yet be inconspicuous. The contrast should be color and not brightness. In a large area, the equipment and machinery will provide the variety, and the walls may be a relatively unimportant background. Certain colors which may look attractive on a color card may prove too bright or distracting when actually applied on a large area. The larger the area, the softer the color should be.

2. *Color of Natural Lighting.* Daylight varies greatly in spectral character with the time of day and with the point of compass. Even northern light, which has the st diversity in intensity, varies from a bluish white when the sky is clear to practically white when the sky is overcast. For typical characteristics of natural light see Table 13-26. When daylight is predominant in an area, complementary wall colors will usually give the best results.

3. *Color of Artificial Light.* The color quality of various light sources was given in Table 13-21. These qualities, together with a listing of the complementary colors, are given in Table 13-26. The color of light from any fluorescent installation is different from the accustomed yellowish-white color of incandescent-filament lighting. The color of interiors is important. For the effect of the type of lighting on wall colors as classified by a large group of observers, see Table 13-27. The reflection factor of a

given surface may vary for different light sources. The variation depends upon the reflection characteristics of the surface and the color content of the lamp output. The effect of the type of lighting on the human skin is given in Table 13-28. The psychological reactions which different persons have toward the various light sources and colors present a problem which cannot be ignored if the result is to be a comfortable and satisfactory interior and lighting scheme.

The color of lighting used in grocery stores, kitchens, cafeterias, and restaurants must be given due consideration. The daylight 4500 and 3500°K white fluorescent lamps in general have proved unsuitable to foodstuffs. The "soft-white" fluorescent lamp which has an approximate color temperature of 2800°K is kind to the complexion, raw and cooked food, and to many displays of wearing apparel. The effect of types of lighting on meats, fruits, and vegetables is given in Table 13-29.

Always choose paint colors under the same type of lighting with which they are to be used.

4. *Atmospheric Conditions.* No matter what type of light source is employed, brightness, as well as color, plays a significant part in the final result obtained. Low-intensity fluorescent lighting will almost certainly be cool, while if the illumination were increased to 50 ft-c, the result would be far more stimulating.

Table 13-26. Characteristics of Natural and Artificial Light

	Natural light	
Principal direction of light	Predominant color of light	Complementary color (contrasting hues)
Eastern	Neutral in color	Warm gray, sunlight
Southern	Yellowish tinge	Shades of blue
Western	Reddish tinge	Cool green
Northern	Cool blue	Light buff, peach
	Artificial light	
Type of light source	Predominant color of light source	Complementary colors*
Incandescent filament	Yellowish white	Light blue, light green, bluish green
Soft white fluorescent	Warmest in hue of all types of white fluorescent lamps (rich in red tones)	Neutral gray, grayed green, grayed blue
3500°K white fluorescent	Yellowish green cast	Warm gray, beige, peach
4500°K white fluorescent	Greenish blue cast	Buff, coral
Daylight fluorescent	Cold bluish	Tan, coral
Combination incandescent and 3500 or 4500°K fluorescent	About the same as soft-white fluorescent	Same as soft white fluorescent
Mercury	Bluish green	Light purple, reddish gray

* Complementary colors are known as contrast colors. If a disk is divided into differently colored sections and is rotated rapidly, different color effects will be experienced according to the colors used. If, for example, only red and yellow sectors are used, the rotating card will appear to be orange. The two colors will bend to produce an intermediate color. If the two sectors are red and green, both colors will disappear and the rotating disk will look gray. Colors that cancel each other out in this manner instead of blending are called "complementary" colors. Each and every color has a complementary color, i.e., yellow and violet—the sum of red and blue; red and green—the sum of yellow and blue; blue and orange—the sum of red and yellow, and so forth for all the secondary colors.

Among the many odd attributes of color is the fact that some hues give the feeling of warmth and some of coolness. In general, colors in the blue end of spectrum tend to create a cool atmosphere and will have an effect equivalent to an appreciable fall in temperature. On the other hand, colors in the red end of the spectrum can compensate for the effects of a chilly atmosphere and can create a warmer, more friendly atmosphere. Ivories and creams, frequently used, are classed as warm tones but have been found to become tiresome. Red and orange colors irritate, hence are believed

Table 13-27. Effect of Type of Lighting on Wall Colors

(Illumination level, 50 ft-c)

Color-conditioning paint*	Daylight fluorescent	3500°K white fluorescent	Soft-white fluorescent	Incandescent filament
Ivory	Faded gray, unattractive	Slightly greenish gray	Slightly yellow	Cream
Rating	Poor	Good	Preferred	Good
Sunlight	Bluish yellow	Slightly bluish	Vivid cream	Rich yellow
Rating	Fair	Preferred	Good	Good
Peach	Slightly pink	Normal, slightly cold	Normal, slightly pinkish	Normal, same as white light
Rating	Good	Fair	Preferred	Good
Light rose	Purplish pink	Yellowish	Normal, vivid pink	Pinkish orange
Rating	Fair	Good	Preferred	Good
Equal parts of light green and white	Grayish green, cold	Whitish gray green	Pale green	Pale green
Rating	Fair	Good	Preferred	Fair
Light green	Fresh bluish green	Greenish gray	Slightly grayish	Yellowish pale green
Rating	Preferred	Good	Good	Good
Equal parts of light blue and white	Grayish blue, cold	Bluish green	Pale blue	Whitish blue
Rating	Poor	Good	Good	Fair
Light blue	Bluish green	Grayish green	Slightly grayish	Grayish faded blue
Rating	Good	Preferred	Good	Good
Equal parts light gray and white	Bluish white, cold	Cold gray	Warmer, pale gray	Off shade of white
Rating	Poor	Poor	Fair	Good
Light gray	Cold gray	Faded gray	Warm gray	Warmer gray
Rating	Poor	Poor	Fair	Good

Color-conditioning Trim and Dado Colors*

Wall colors	*Trim and dado colors*
Ivory, sunlight, or peach	Tan
Light rose	Deep rose
Green and light green	Medium green and deep green
Blue and light blue	Medium blue and deep blue
Gray and light gray	Medium gray and deep gray

* The vividness of sunlight, peach, and light rose may be toned down the same as green and blue, *i.e.*, equal parts of these colors and white.

Table 13-28. Effect of Type of Lighting on Human Skin

(Illumination level, 50 ft-c)

Description of skin	Daylight Mazda F	3500°C white Mazda F	Soft-white Mazda F	Incandescent filament
Untanned skin	Pale, slightly gray	Sallow, slightly greenish for older skin	Enhanced	Yellowish, appearance slightly tan
Rating	Fair	Acceptable	Preferred	Good
Tanned skin (medium)	Grayish tint	Intensified greenish tint, very sallow	Healthy glow	Intensified reddish tint
Rating	Fair	Fair	Preferred	Fair
Deep pigmentation	Intensified grayish tint	Intensified greenish tints very sallow	Healthy glow	Deep rich brown
Rating	Poor	Fair	Good	Preferred
Sunburned skin (red)	Slightly reddish, deep-purple tint	Tanned appearance	Slightly reddish	Fiery red
Rating	Fair	Preferred	Good	Poor
Untanned skin, light make-up	Overemphasized make-up in contrast to skin	Make-up grayed, sallow appearance	Blended skin and make-up improved appearance	Intensified make-up, slightly sallow
Rating	Poor	Fair	Preferred	Good
Untanned skin, heavy make-up	Further overemphasized make-up, very pale skin	Sallow appearance of unpainted skin intensified	Enhanced skin make-up	Slightly sallow
Rating	Poor	Fair	Preferred	Good

to be a frequent cause of anger. The colors in the center of the spectrum—light blue, green or yellow green—seem to be the most effective, both for the psychological and lighting factors.

The effect of warmth due to hard physical work can be compensated in exactly the same way as that of a high atmospheric temperature.

Table 13-29. Effect of Types of Lighting on Meats, Fruits, and Vegetables
(Illuminating level, 50 ft-c)

Test article	Daylight Mazda F	3500° white Mazda F	Soft-white Mazda F	Filament (100 watts)
Red meats	Poor	Fair	Good	Preferred
Dressed chicken	Poor	Fair	Good	Good
Butter	Poor	Good	Good	Preferred
Chocolate	Poor	Fair	Fair	Good
Bread (brown crust)	Fair	Good	Good	Good
Oysters (opened)	Poor	Poor	Fair	Preferred
Parsley	Good	Good	Fair	Poor
Carrots	Good	Good	Good	Good
Tomatoes	Fair	Fair	Fair	Preferred
Red apples	Fair	Fair	Fair	Preferred
Onions	Poor	Fair	Fair	Fair
Green apples	Good	Good	Fair	Poor
Squash	Fair	Good	Preferred	Good
Bananas	Fair	Good	Good	Good
Red and green peppers	Fair	Good	Good	Good
Green beans	Good	Good	Fair	Fair
Cauliflower	Poor	Poor	Good	Good
Green cabbage	Fair	Poor	Fair	Fair
Red cabbage	Fair	Fair	Good	Preferred
Corn	Good	Preferred	Good	Good
Lemons	Poor	Poor	Fair	Good
Peaches	Good	Good	Good	Good
Oranges	Fair	Fair	Good	Preferred
Plums (reddish purple)	Fair	Good	Preferred	Good

NOTE: With reference to painting, light peach has proved the most "appetizing" wall color for restaurants and cafeterias under all types of lighting. The soft white is the best type of fluorescent lighting for kitchens, restaurants, cafeterias, etc.

In such nonwork areas as corridors, washroom, rest rooms, etc., color can provide a welcome "change of pace." The color preferences of the two sexes determine the color recommendations for the respective facilities. Color of nonworker-area industrial interiors must be chosen for their "livability," and it is wise to bring the employees involved into the final decision. Men usually express a preference for cool colors while women like warm colors.

5. *Noise.* Noise has a great effect on seeing and causes a continual irritation of the central nervous system. Owing to the deleterious effects of noise, all practical measures to reduce the noise level should be taken. Even though acoustical materials cause a reduction in the reflectivity of the ceilings and walls, the loss of illumination will be more than compensated for by the increase in comfort, except where the intensity level is low for the visual tasks.

Color has played an important part in recent years through the practice of painting machinery to improve visibility. To judge from current advertising, a little paint will make any lighting system, however inadequate, a satisfactory installation. (One paint manufacturer published colored pictures of brilliantly painted machines and plant interiors with the area lighted with tin cone reflectors of ancient vintage.) Brilliant colors have no place in creating successful seeing conditions in industry, since from their very nature they introduce uncomfortable contrasts.

Bearing the above comments in mind, the following general recommendations for industrial interiors are made:

Ceilings. White or near-white paint should be used on ceilings with the lighting installation arranged to blend with the ceiling so as to present the least possible distraction. Where direct-type lighting is used, the finish of the ceiling becomes less important. When factory ceilings are excessively high, a gloss white may be satisfactory. Where such factors as dirt resistance and washability must be considered, a glossy finish will give the best performance; however, never use a glossy finish where light may be reflected to the eyes.

Structural. A medium tint of bluish green or pea green on structural beams will give a feeling of support. If the number of exposed beams is large, aluminum paint may be satisfactory, while white is preferred.

Upper Walls and Columns. Grayish tints having a reflection factor of 50 to 60 per cent, will be most satisfactory on wall areas. A dado may be used to advantage especially in the smaller shops. Walls should be finished in hues to complement either the exposure or the type of artificial light, whichever produces the most predominant lighting in the area.

Lower Walls and Columns. Colors blending with upper walls should be used on lower walls and columns. Avoid colors that go "flat" under the lighting system. Use color for recognition for such items as markers or traffic aids, fire exits, safety precautions, etc.

Floors. A great deal has been said lately about painting factory floors in light colors "to aid in light reflection." Except where it is desirable to reflect light upward under large horizontal surfaces such as aircraft wings, a not too bright color of medium depth is recommended with "traffic stripes" separating the working areas from those used for the movement of materials. These traffic stripes can be as bright as necessary to serve as "keep-clear" warnings. Some existing floors cannot be successfully painted because they are oil soaked or otherwise in such condition that painting would present an almost impossible maintenance problem.

Machines. The machines should be painted in shades to delineate machine parts and danger areas and promote visibility and safety. Avoid using different colors too profusely, or they will loose their effectiveness as contrast mediums. The color scheme of the walls, floors, and machines should present a harmonious pattern limited as to similarity. On no account should a uniformly colored light finish be used over machines, walls, and even floors, as they will give an extremely monotonous appearance.

Three-dimensional seeing, a science of color and light for better vision in industry, offers a solution to the problem of improving industrial seeing conditions.

Desks. Desk tops need a higher finish than the 5 to 7 per cent prevalent today. Thirty-five per cent is the minimum recommended with the desk sides higher in reflectance because the vertical illumination is, in almost all cases, much less than the horizontal.

The evaluation of reflection factors in finishing ceilings, walls, and equipment will result in better light utilization and more desirable ratios. This will, of course, depend on a proper analysis of practicability. There are many industrial operations which are not sufficiently clean to justify light-colored finishing, such as, for example, some chemical-plant areas, foundries, etc., where a coat of light paint may last little longer than the time to put it on. Here the lighting system must be designed to give the illumination required from direct-type lighting units which properly shield the lamp source. However, in the majority of industrial plants, the most satisfactory system proves to be a combination of adequate lighting intensity and a simple color scheme having a proper reflectance value. The result will be a well-planned seeing condition.

While each plant presents certain individual problems, the general plans developed through the systems of "color conditioning" and three-dimensional seeing serve as guides in applying acceptable color principles.

Some of the important facts concerning the relation of light and color are listed below:

1. Lighting for seeing also involves paint for seeing. Color plays an important

part in every industrial-plant installation. Applications vary widely from a simple matter of personal preferences for private offices to the more complicated industrial inspection tasks where small color differences must be detected.

2. Color appearance of objects depends not only on the reflectance characteristics of the objects themselves but also on the color content of the light with which they are illuminated. The appearance of materials having complex spectral reflectance characteristics, when illuminated by fluorescent sources which have complicated energy distributions, is often impossible to predict.

3. Light greatly outstrips paint in ability to produce variations in color effects. Tables 13-26, 13-27, 13-28, and 13-29, indicate some of the changes that take place under filament incandescent and three types of "white" fluorescent lamps.

4. The choice of industrial finishes depends on many psychological and practical viewpoints. The four most important aspects of color in industrial lighting operations are

a. The finishes of ceilings, walls, and machines to provide a pleasing background of brightness and color for optimum visual efficiency and comfort.

b. The finishes of work and danger areas for safety and visual efficiency.

c. The provision of appropriate color-quantity lighting for inspection tasks involving the discrimination and evaluation of color.

d. The color scheme should be simple. The large number of people using the area and the many jobs carried out make durability an important aspect, and only a simple scheme can be durable.

5. Yellow, red, and blue are the primary colors. From these all the hues of the spectrum can be produced, but none of the primary colors can be obtained by any mixture.

6. Office equipment is sadly in need of reform. The majority of typewriters, accounting machines, etc., are finished in a high-gloss black enamel. There is no reason why such machines should not be finished in a light color and have a finish to prevent undesirable reflections. The use of colored paper always reduces visibility. Glossy finishes on paper are also undesirable. Blotting pads and desk tops should be light in color and free from undesirable reflections.

7. One of the principal factors of importance to good seeing conditions is the general "atmosphere" of the surroundings—brightness, color, variety.

8. In inspection work where the products must be held to narrow color-tolerance limits, the best lighting system is one which is rich in the spectral region where there is maximum absorption (minimum reflectance). This will accentuate the differences in color. The source should produce light throughout the visible spectrum and be so located that specular reflections are avoided. For detecting small color differences Table 13-30 gives the best type of light.

Table 13-30

Colors	*Type of lighting*
Blue, purple, violet .	Incandescent filament
Red, pink	Daylight fluorescent
Green, yellow	3500°K white fluorescent

Observations on the Relations of Lighting Effects and Psychological Reactions

In 1845, the first lighting survey was conducted at the President's Mansion (White House) in Washington, with recommendations submitted to Mrs. James K. Polk. As a result, some 18 kerosene lamps replaced 150 tallow candles in the East Room, thereby approximately doubling the lighting intensity and reducing daily lighting expense from $8.75 to $2.11. A total of 416 candles lighted the White House at the time. When the remarkable "coal oil burners" were installed, there were many who marveled, but those were the days of *fractional* foot-candles--not eye comfort.

Lighting standards have advanced greatly since then. Thirty years ago, the first industrial lighting code of the Illuminating Engineering Society recommended, with considerable temerity, a lighting intensity of 1.25 to 1.35 ft-c. The reasons for the low foot-candle level were not based on the needs or habits of the human eye but rather on the prohibitive expense, excessive heat from the light source, detrimental glare of high candle-power sources, and the inadequacy of the wiring.

Today, the lighting problem in industrial plants is dynamic and changing in importance and scope. Dominant human factors, including inherent differences in the optical and neural apparatus for seeing, defects from injury or disease, age of the eyes, previous light education, etc., all produce a considerable range in solving lighting problems for various conditions and individuals.

In giving attention to the needs which different persons seem to feel for illumination, it has been observed that there is to some degree a correlation existing between the nature of a person, the type of work he enjoys or is qualified to do, and the quantity and quality of lighting he finds satisfactory, regardless of whether or not it is a "correct" lighting job under the modern practice type of installation.

We are fortunate in that persons engaged in the same type of work or activity usually have enough in common to make satisfactory the use of some general type of lighting for the entire group. To provide special lighting effects for each individual would be very impractical and prohibitive in costs; however, the individual should be studied as a means of learning the characteristics of the group.

The characteristics of individual groups and the needs they seem to require regarding illumination are outlined in Table 13-31. The psychological reactions which different types of people give to each of the more common types of lighting effects are outlined in table.

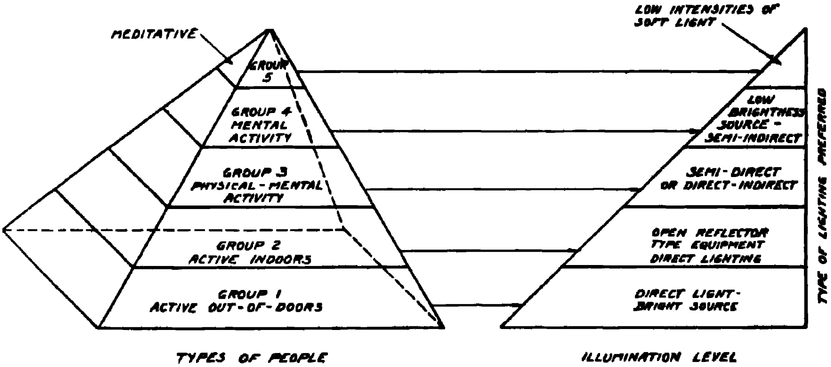

MEDITATIVE
GROUP 5
GROUP 4 MENTAL ACTIVITY
GROUP 3 PHYSICAL - MENTAL ACTIVITY
GROUP 2 ACTIVE INDOORS
GROUP 1 ACTIVE OUT-OF-DOORS
LOW INTENSITIES OF SOFT LIGHT
LOW BRIGHTNESS SOURCE - SEMI-INDIRECT
SEMI-DIRECT OR DIRECT-INDIRECT
OPEN REFLECTOR TYPE EQUIPMENT DIRECT LIGHTING
DIRECT LIGHT - BRIGHT SOURCE
TYPE OF LIGHTING PREFERRED
TYPES OF PEOPLE
ILLUMINATION LEVEL

FIG. 13-50. Types of people and their choice of lighting effects.

Figure 13-50 shows a pyramid divided into five sections of equal height with each section representing a type of person. Group 1 is classified as active out-of-doors. This type of person is inclined toward muscular expression and is fond of bright sunlight. Emphasis is inclined to be placed on physical activity with mental effort being at a minimum. In supplying lighting for this group, it has been found they demand that the lamp "shine" and that it be a bright source. In many applications even bare incandescent lamps may be satisfactory. The unit should be mounted well above the line of vision as a means of reducing glare.

Group 2 is classified as active indoors. This type of people operate machines, doing work of a routine nature which requires a minimum of mental activity. Since the tasks are usually of a concentrated order, local lighting focusing attention upon a central point finds favor with this group. Open-type lighting reflectors may be used to advantage in the general lighting system in this type of occupancy.

Group 3 is the physical-mental activity type, whose thinking deals with people and material things, such as physicians, merchants, industrialists, etc. While they enjoy daylight extensively, people who perform this type of work usually prefer a soft type of uniform illumination. Their choice of artificial lighting, oddly enough, is generally an enclosed visible light source.

Group 4 is the mental activity type. Their thoughts deal with ideas and ideals. Work involving physical activity is usually foreign to their nature. The choice of artificial lighting is a low brightness source of the indirect incandescent type or the semidirect fluorescent type.

Table 13-31. The Key to Harmonized Lighting

Characteristics of the more common lighting effects	Types of people				
	Active out of doors	Active indoors	Physical-mental activity	Mental activity	Meditative
1. *Bare lamps* High brightness of light source. Radiant light, poor distribution of light producing sharp shadows and glare. Natural example: bright sunshine.	↑ Exciting active lively ←	Physical activity	Annoying	Irritating	Extremely irritating
2. *Open reflectors (rim type)* Light directed downward from fixture, high intensity of light on working plane, sharp shadows, reduced glare. Natural example: moderate sunshine.	Physical activity	↑ Concentrated physical activity ←	Physical activity	Depressingly annoying	Irritating
3. *Enclosed or shielded fixtures* Lower brightness of light source, diffused light, soft shadows, generally uniform illumination. Natural example: overcast sky or shady location.	Subtle	Mentally active	↑ Physically and mentally active ←	Physical activity	Annoying
4. *Direct-indirect fluorescent, semi-indirect incandescent* Low brightness of apparent ceiling, well-diffused light, soft shadows, high uniformity of light distribution. Natural example: late afternoon.	Dead	Subtle	Restful	↑ Mentally active ←	Physically and mentally active
5. *Indirect lighting supplemented by bright local sources* Low intensity of indirect lighting supplemented by local lighting having small but bright light sources. Natural example: twilight.	Dead	Dead	Dull	Restful	↑ Meditative ←

Group 5 is the meditative type. These people are very sensitive and usually find expression in passive thought or meditation: their interests are not akin to the commercial or industrial world. This group includes inspirational writers, musicians, etc. This type prefers a lighting effect which simulates twilight with local high-lighting on the work.

It is important to note that the distribution of people may be considered as approximately represented by the volume of the five sections of the pyramid and, even though most people are intermediate or combination types, their natural expression can be used as an index or key to harmonized lighting, which will please the various classifications or groupings of people, regardless of whether or not the electrical engineer considers the type of lighting adequate in quality and quantity for the visual tasks performed. To provide satisfactory lighting, it is necessary to impose the human factors upon calculations.

Some of the important facts concerning the relationship of lighting effects and psychological reactions are noted below:

1. The process of seeing is very complicated and much affected by the flexibility of the individual nervous system.

2. The production and control of light are physical sciences; however, lighting is also concerned with vision, seeing, and physiological and psychological effects.

3. The psychological effects of lighting are concerned with such items as attention, impression, distraction, annoyance, alertness and mental fatigue.

4. Lighting calculations and practice were, and still are, largely empirical. Our present knowledge concerning the relation of light to sight is neither comprehensive nor well understood. Only recently has the science of lighting begun to be a part of our present meager knowledge in the science of seeing.

5. Design of good lighting involves not only the physics, mathematics, and economics of light generation and control, but also:

a. The physiology of visual sensation and its related processes.

b. The psychology of the mind, which correlates, integrates, and evaluates the sensation.

6. The most important item in considering any lighting system is the ***human factor***, or the people for whom the lighting is being installed.

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SECTION 14

MECHANICAL POWER TRANSMISSION

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POWER BELTING, FASTENERS, PULLEYS, AND SHAFTING

BY WILLIAM STANIAR, M.E.

POWER BELTING

Belting is one medium of delivering a given amount of power at a low cost per unit of time which should result, provided the correct type and size are employed, in the transmission of power at a low cost per horsepower per year over a long period of useful service. The function of power belting is to transmit power economically. Therefore the user should be familiar with the characteristics and properties of the various types of belting available. Mechanical and atmospheric operating conditions vary; hence knowledge of such conditions results in low maintenance, efficient power delivery, and lengthened life of the belting and other transmission appurtenances.

Belting Types

1. Leather
2. Rubber
3. Stitched cotton duck, impregnated
4. Solid woven cotton, impregnated
5. Camel's hair, impregnated
6. Balata
7. Teon

These types vary in characteristics and atmospheric resistance based on the substance employed and the various methods of manufacture.

Leather Belting

The best leather for power belting is derived from the hides of three- or four-year-old steers. Such hides are of two classes, "packer" and "country." Packer hides are those taken from the animal in large packing houses by experts and hence are relatively free from flesh cuts. Country hides are those taken off by butchers. Belting leather is made from that portion of the hide approximately 48 in. from the root of the tail toward the head and 24 in. from each side of the backbone. This portion is known as the "butt bend." The quality of the leather is generally determined by the relation of the belting strip to the backbone center line.

Preparing the green hide for leather belting consists of (1) tanning by either the oak or mineral process, (2) cutting out the butt bend, (3) stuffing or currying the tanned butt bend, (4) cutting the butt bend into centers and sides, (5) stretching sides and centers, and (6) glazing or jacking the sides and centers.

Constructing the leather belt consists of a series of operations: (1) cutting the centers and sides into the required belting widths, (2) matching the strips for thickness, (3) scarfing and tapering the ends of the strips, (4) squaring the scarfed ends, (5) assembling and cementing by either a nonwaterproof glue or a waterproof cement, (6) pressing, (7) trimming and stamping, and (8) measuring and finishing.

The required strips are cut from both sides of the butt bend and are known as "rights" and "lefts." These strips are cemented end to end and must alternate right and left throughout the length of the belt, with all edges toward the backbone on one side and all edges toward the belly on the other. The method is to join tail end to tail end and shoulder end to shoulder end. The construction is shown by Fig. 14-1.

In cementing together two single-ply leather belts to form a double-ply belt, the ends are so arranged that each part of one lies in a reversed position to the other. The lap of a single-ply belt is made as shown in Fig. 14-2 and that of a double-ply belt as shown in Fig. 14-3.

First-quality leather belting depends upon the leather having fibers of firm texture and being free from sponginess and looseness. Leather cut within 15 in. of each side of the backbone of the hide is generally considered first quality, but modern practice tends to select belting strips from the butt bend with regard to firmness of fiber rather than definite measured limits.

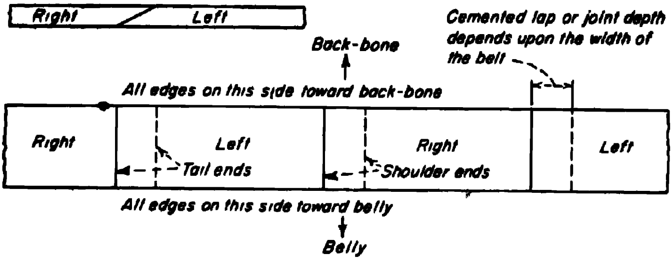
Right
Left
Back-bone
Cemented lap or joint depth depends upon the width of the belt
All edges on this side toward back-bone
Right
Left
Tail ends
Right
Shoulder ends
Left
All edges on this side toward belly
Belly

Fig. 14-1. Single-ply leather-belt construction.

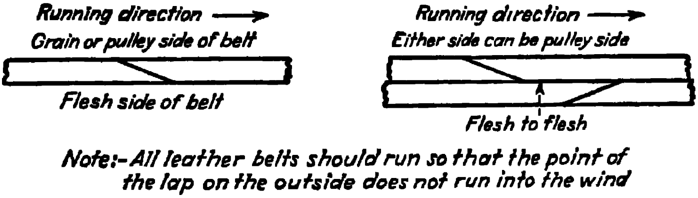
Running direction
Grain or pulley side of belt
Flesh side of belt
Running direction
Either side can be pulley side
Flesh to flesh
Note:-All leather belts should run so that the point of the lap on the outside does not run into the wind

Fig. 14-2. Single-ply leather-belt lap.

Fig. 14-3. Double-ply leather-belt lap.

Second-quality leather belting is known as "low-cut leather" since it is cut from the butt bend beyond the 15 in. limit. It is cheaper than first-quality belting and can be used for light-power, normal driving.

Several grades and brands of leather belting are manufactured, but essentially there are four distinct types: (1) nonwaterproof oak tanned, (2) waterproof oak tanned, (3) waterproof mineral retanned, (4) waterproof combination of oak tanned and mineral retanned.

Table 14-1. Minimum Pulley Diameters in Inches for Oak-tanned Leather Belting at Various Belt Speeds

Belt thickness, in.	Belt speed, fpm					
	1,000	2 000	3,000	4,000	5,000	6,000
Single ply						
8/64	1½	2	2¼	2½	3	3½
10/64	2	2½	3	3½	4	4½
12/64	2¾	3½	4	4½	5	5½
14/64	3½	4	4½	5½	6	7
Double ply·						
15/64	5	6	7	8	9	10
18/64	6	8	9	10	12	13
20/64	8	9	11	12	14	16
23/64	9	11	13	15	17	19
26/64	11	14	16	18	20	22
Triple ply:						
28/64	18	21	24	28	31	35
32/64	20	24	28	32	36	40
35/64	22	27	32	36	40	45

Oak-tanned Leather Belting (Nonwaterproof). ***Characteristics***

1. Fabricated with hot glue.
2. Made in one ply, two ply, and three ply. Known as singles, doubles, and triples. Four ply is seldom used.
3. Widths of singles, ⅛ to 10 in.; doubles, 1 to 72 in.; triples, 12 to 72 in.
4. Not flexible when new, therefore not adaptable to small-diameter high-speed pulleys.
5. Possesses low stretch.
6. Coefficient of friction: new, 0.27 to 0.45; well worn, 0.35 to 0.60.
7. Can be made endless by manufacturer or at point of application.
8. Will not resist moisture, steam, acid, or high temperatures.

Oak-tanned Leather Belting (Waterproof). This type is similar to regular oak-tanned belting with one exception. It is fabricated with a moistureproof pyroxylin cement, and the leather is specially dressed to resist the action of moisture.

Horsepower Formulas and Tables for Oak-leather Belting (Nonwaterproof and Waterproof)

SINGLES:
Light: ⅛ in. thick, 40 lb allowable effective tension per inch width
Medium: $\frac{5}{32}$ in. thick, 45 lb allowable effective tension per inch width
Heavy: $\frac{3}{16}$ in. thick, 50 lb allowable effective tension per inch width
FORMULA:

$$\mathrm{Hp} = \frac{T \times W \times S}{33{,}000}$$

$$W = \frac{\mathrm{hp} \times 33{,}000}{S \times T}$$

where T = allowable effective tension, lb per in. width
W = width of belt, in.
S = speed of belt, fpm

Table 14-2. Horsepower Rating for Medium Single-oak Belting $\frac{5}{32}$ In. Thick, Based on 45 Lb Effective Tension

Width of belt, in.	Belt speed, fpm											
	750	1,000	1,250	1,500	1,800	2,000	2,500	3,000	3,500	4,000	4,500	5,000
1	1	1.36	1.71	2.05	2.43	2.7	3.4	4.1	4.8	5.4	6.1	6.9
2	2	2.72	3.42	4.06	4.86	5.4	6.8	8.2	9.6	10.8	12.2	13.8
3	3	4.1	5.1	6.1	7.3	8.2	10.2	12.3	14.4	16.2	18.3	20.7
4	4	5.4	6.8	8.12	9.72	10.8	13.6	16.4	19.2	21.6	24.4	27.6
5	5	6.8	8.5	10.1	12.1	13.5	17.0	20.5	24.0	27.0	30.5	34.5
6	6	8.1	10.2	12.18	14.5	16.2	20.4	24.6	28.8	32.4	36.6	41.4
7	7	9.5	12.0	14.21	16.91	18.9	23.8	28.7	33.6	37.8	42.7	48.3
8	8	10.8	13.6	16.2	19.4	21.6	27.2	32.8	38.4	43.2	48.8	55.2

These ratings are based on 165 deg arc of contact and are not corrected for the action of centrifugal force.

For light singles ⅛ in. thick, figure 90 per cent of these ratings.
For heavy singles $\frac{3}{16}$ in. thick, figure 112 per cent of these ratings.

DOUBLES:
Light: ¼ in. thick, 60 lb allowable effective tension per inch width
Medium: $\frac{5}{16}$ in. thick, 65 lb allowable effective tension per inch width
Heavy: ⅜ in. thick, 75 lb allowable effective tension per inch width

Table 14-3. Horsepower Rating for Medium Double-oak Belting $\frac{5}{16}$ In. Thick, Based on 65 Lb Effective Tension

Width of belt, in.	Belt speed, fpm												
	500	850	1,000	1,250	1,500	1,800	2,000	2,500	3,000	3,500	4,000	4,500	5,000
3	3	5	6	7.4	9	10.6	12	15	18	20	24	27	30
3½	3.42	5.8	6.9	8.6	10.3	12.3	14	17	20.8	24	28	31	35
4	4	6.7	8	10	12	14	16	20	24	27.5	31.5	35.5	39.5
5	5	8.5	10	12.3	14.7	17.5	20	24	30	35	40	44	49
6	6	10	12	15	18	21	24	30	35	41	47	53	59
8	8	13.5	15.5	20	23.5	28.5	31.5	40	47	55	63	71	79
10	10	17	20	25	30	35	40	49	59	69	79	89	99
12	12	20	23.5	30	35	42.5	47.5	60	71	83	95	107	118
14	14	23.5	28	34.5	41	50	55	69	83	97	110	124	138
16	16	26.5	31.5	39	47	57	63	79	94	110	125	142	157
18	18	30	35	44	53	64	71	90	107	123	142	160	177
20	20	33	40	49	59	71	78	98	118	139	158	177	200
24	24	40	47	59	71	85	95	118	142	166	190	212	236
30	30	50	59	74	89	106	118	147	177	206	236	266	300

These ratings are based on 165 deg arc of contact and are not corrected for the action of centrifugal force. For light doubles ¼ in. thick, figure 92 per cent of these ratings. For heavy doubles ⅜ in. thick, figure 114 per cent of these ratings.

TRIPLES:

Medium: ½ in. thick, 100 lb allowable effective tension per inch width

Heavy: $\frac{9}{16}$ in. thick, 110 lb allowable effective tension per inch width

Table 14-4. Horsepower Rating for Medium Triple-oak Belting ½ In. Thick, Based on 100 Lb Effective Tension

Width of belt, in	Belt speed, fpm										
	200	400	600	800	1,000	1,500	2,000	2,500	3,000	3,500	4,000
12	7	14	21	28	35	55	70	91	110	127	140
14	8.5	17	25.5	34	42.5	64	85	106	128	148	170
16	9.5	19	28.5	38	47.5	73	95	121	146	169	190
20	12	24	36	48	60	91	120	151	182	212	240
24	14.5	29	43.5	58	72.5	110	145	182	220	254	290
28	17	34	51	68	85	127	170	212	254	300	340
30	18	36	54	72	90	136	180	228	272	320	360
36	22	44	66	88	110	164	220	274	328	380	440
44	27	54	81	108	135	200	270	334	400	465	530
48	29	58	87	116	145	218	290	362	436	510	580
60	36	72	108	144	180	272	360	450	544	635	720
72	44	88	132	176	220	326	440	550	652	820	880

These ratings are based on 165 deg arc of contact and are not corrected for the action of centrifugal force. For heavy triples, $\frac{9}{16}$ in. thick, figure 111 per cent of these ratings.

Mineral-retanned Leather Belting (Waterproof). This type is oak leather retanned by a mineral process. The mineral process of retanning produces a leather entirely different from that created by the regular oak tannage, and it therefore has different characteristics.

Characteristics

1. Made in similar plies, thicknesses, and widths as regular oak but possesses greater power-transmitting ability ply for ply and width for width.
2. Fabricated with a cement insoluble in water, moisture, or steam.
3. The leather is treated to resist water, moisture, steam, and acids.

4. Extremely flexible in all thicknesses, therefore adaptable to high-speed small-diameter pulleys.
5. Coefficient of friction, 0.60 to 0.87.
6. Can be made endless by manufacturer or at point of application.
7. Will resist water, moisture, steam, temperatures to 130°F, mineral oil, cutting compounds, and also the action of the mists and vapors of the corrosive acids for a considerable period of time.
8. Cannot be dressed with the compounds used for oak-tanned leather.

Horsepower Tables for Mineral-retanned Leather Belting

SINGLES:

Light: $\frac{1}{8}$ in. thick, 45 lb allowable effective tension per inch width
Medium: $\frac{5}{32}$ in. thick, 50 lb allowable effective tension per inch width
Heavy: $\frac{3}{16}$ in. thick, 55 lb allowable effective tension per inch width

Table 14-5. Horsepower Rating of Medium Single Mineral-retanned Belting $\frac{5}{32}$ In. Thick, Based on 50 Lb Effective Tension

Width of belt, in	Belt speed, fpm										
	650	850	1,000	1,250	1,500	1,800	2,000	2,500	3,000	3,500	4,000
1	1.0	1.29	1.52	1.9	2.3	2.7	3.0	3.8	4.5	5.3	6.1
1¼	1.23	1.6	1.9	2.4	2.8	3.4	3.8	4.8	5.7	6.6	7.5
1½	1.48	1.92	2.3	2.8	3.4	4.1	4.5	5.6	6.8	8.0	9.1
2	2.0	2.6	3.0	3.8	4.5	5.4	6.0	7.5	9.1	10.6	12.0
2½	2.46	3.2	3.8	4.8	5.7	6.8	7.6	9.4	11.4	13.3	15.2
3	3.0	3.9	4.5	5.7	6.8	8.2	9.1	11.4	13.7	16.0	18.2
4	4.0	5.1	6.0	7.6	9.1	11.0	12.0	15.2	18.2	21.2	24.1

These ratings are based on 165 deg arc of contact and are not corrected for the action of centrifugal force. For light singles $\frac{1}{8}$ in. thick figure 90 per cent of these ratings. For heavy singles $\frac{3}{16}$ in. thick figure 112 per cent of these ratings.

DOUBLES:

Light: in. thick, 65 lb allowable effective tension per inch width
Medium: $\frac{5}{16}$ in. thick, 70 lb allowable effective tension per inch width
Heavy: $\frac{3}{8}$ in. thick, 80 lb allowable effective tension per inch width

Table 14-6. Horsepower Rating of Medium Double Mineral-retanned Belting $\frac{5}{16}$ In. Thick, Based on 70 Lb Effective Tension

Belt width, in	Belt speed, fpm												
	500	850	1,000	1,250	1,500	1,800	2,000	2,500	3,000	3,500	4,000	4,500	5,000
2	2.1	3.6	4.2	5.3	6.4	7.6	8.5	10.6	12.6	15.0	17.0	19.0	21.0
2½	2.7	4.5	5.3	6.6	8.0	9.5	10.6	13.2	16.0	18.5	21.0	24.0	26.5
[illegible]	3.2	5.4	6.4	8.0	9.6	11.5	12.8	16.0	19.0	22.0	25.0	29.0	32.0
3½	3.7	6.3	7.5	9.3	11.1	13.4	15.0	18.5	22.0	26.0	30.0	33.5	37.0
4	4.2	7.2	8.5	10.6	12.8	15.3	17.0	21.0	25.0	30.0	34.0	38.0	42.5
5	5.3	9.0	10.6	13.2	16.0	19.0	21.0	26.5	32.0	37.0	42.0	48.0	53.0
6	6.4	10.8	12.8	16.0	19.0	23.0	25.0	32.0	38.0	45.0	51.0	57.0	64.0
8	8.5	14.5	17.0	21.0	25.0	30.0	34.0	42.0	51.0	60.0	68.0	76.0	85.0
10	10.6	18.0	21.0	26.5	32.0	38.0	42.0	53.0	64.0	74.0	85.0	96.0	106.0
12	12.7	21.6	25.0	32.0	38.0	46.0	51.0	64.0	77.0	89.0	102.0	115.0	127.0
16	17.0	29.0	34.0	43.5	51.0	61.0	68.0	85.0	102.0	118.0	136.0	152.0	170.0
18	19.2	32.5	38.0	48.0	57.0	69.0	76.0	95.0	115.0	134.0	153.0	172.0	190.0
20	21.0	36.0	42.0	53.0	64.0	76.0	85.0	107.0	128.0	150.0	170.0	190.0	210.0
24	25.5	43.0	51.0	64.0	76.5	92.0	102.0	127.0	152.0	178.0	204.0	230.0	256.0

ratings
force. For light doubles ¼ in. thick, fig
thick, figure 114 per cent of these ratings.

TRIPLES:
Medium: ½ in. thick, 110 lb allowable effective tension per inch width
Heavy: 9/16 in. thick, 120 lb allowable effective tension per inch width

Table 14-7. Horsepower Rating of Medium Triple Mineral-retanned Belting ½ In. Thick, Based on 110 Lb Effective Tension

Width of belt, in.	Belt speed, fpm									
	500	1,000	1,500	2,000	2,500	3,000	3,500	4,000	4,500	5,000
12	20	40	60	80	100	120	140	160	180	200
14	23	46	70	92	115	140	161	184	207	230
18	30	60	90	120	150	180	210	240	270	300
24	40	80	120	160	200	240	280	320	360	400
30	50	100	150	200	250	300	350	400	450	500
36	60	120	180	240	300	360	420	480	540	600
42	70	140	210	280	350	420	490	560	630	700
50	83	166	250	332	415	500	581	664	747	830
60	100	200	300	400	500	600	700	800	900	1,000
72	120	240	360	480	600	720	840	960	1,080	1,200

These ratings are based on 165 deg arc of contact and are not corrected for the action of centrifugal force. For heavy triple, 9/16 in. thick, figure 108 per cent of these ratings.

Combination of Oak-tanned and Mineral-retanned Leather Belting. This type is made of a combination of oak-tanned and mineral-retanned leathers. It is fabricated flesh side to flesh side and is available in doubles only. By this construction the high frictional grip of the mineral-retanned leather and the tough rigidity of the oak-tanned leather are combined into one belt. The combination of these features is desirable in heavy-powered slow-speed drives and particularly where the use of shifting belts for such driving is required.

It is specially manufactured for shifting drives and for stepped-cone and flanged-pulley operations, since the oak ply resists the transverse crumpling action caused by shifter forks, the edges of stepped cones, and pulley flanges.

Characteristics

1. Made in thicknesses and widths similar to medium and heavy double-oak leather.
2. Fabricated with a cement insoluble in water, moisture, or steam.
3. More flexible than double-oak leather, based on the mineral-retanned ply.
4. Coefficient of friction, 0.60 to 0.87.
5. Can be made endless by manufacturer or at point of application.
6. Will resist water, moisture, steam, temperatures to 130°F, mineral oils, and cutting compounds.
7. Nonresistant to the corrosive acids or their mists and vapors.
8. Should not be dressed with the compounds used for regular double-oak leather.

Horsepower Tables for Combination Oak-tanned and Mineral-retanned Leather Belting

Medium: 5/16 in. thick, 70 lb allowable effective tension per inch width
Heavy: 3/8 in. thick, 80 lb allowable effective tension per inch width

NOTE: Use Table 14-6 for horsepower ratings of medium-weight, 5/16-in.-thick combination oak-tanned and mineral-retanned leather belting.

For heavy weight 3/8 in. thick, figure 114 per cent of the ratings given in Table 14-6.

Atmospheric Effects on Leather Belting. *Oak-tanned Nonwaterproof.* An atmosphere charged with steam, moisture, or at a temperature above 115°F causes separation of the laps and plies. An atmosphere charged with the mists or vapors of the corrosive acids attacks the leather.

Oak-tanned Waterproof. An atmosphere charged with steam or moisture or at a temperature of 115°F has no effect on the laps and plies. An atmosphere charged with the mists or vapors of the corrosive acids attacks regardless of the special dressing of the leather and the use of pyroxylin cement.

Mineral-retanned Waterproof. An atmosphere charged with steam, moisture, or a temperature to 140°F has no effect on the laps and plies of the leather. An atmosphere charged with the mists or vapors of the corrosive acids attacks this type of leather, but its action is slow.

Combination of Oak-tanned and Mineral-retanned Waterproof. This type is steam- and moistureproof and will resist a temperature of 140°F owing to a special dressing of the oak ply, the use of pyroxylin cement, and the mineral-retanned ply, but the oak ply deteriorates when exposed to the mists or vapors of the corrosive acids.

Leather-belt Dressing or Lubrication. The term "dressing" as applied to leather belting signifies the lubrication of the fibers of the leather. It does not mean creating a sticky surface. Leather belting of all types requires periodical dressing or lubrication, since the heat generated by frictional contact with the pulleys tends to dry out the currying ingredients. This results in the leather becoming dry and hard. The leather should be dressed with a compound containing the currying ingredients so that penetration and a relubrication of the fibrous structure result.

Leather-belting manufacturers supply dressing compounds which contain the currying materials employed in the manufacture of oak-tanned nonwaterproof and waterproof leather belting. Mineral-retanned leather requires a compound different from that required by oak-tanned leather. Such compounds can also be obtained from the respective manufacturers.

Under normal atmospheric and operating conditions leather belting should be dressed every two or three weeks. The most effective results are obtained by applying the lubricant to the outer surface or ply. This permits the material to work its way through the leather and therefore prevents a slippery driving surface on the belt and an accumulation or gumming of the lubricant on the pulley face. If the pulley-contact surface of the belt is very dry, a small amount of the lubricant can be applied to this surface with safety

Cheap and low-grade stick and semiliquid belt dressings which generally contain the ingredients shown in Table 14-8 should not be used on leather belting of any type:

Table 14-8

Stick type	Per cent	Semiliquid type	Per cent
Graphite	22	Tar	36
Rosin	27	Rape seed or cotton oil	54
Vegetable oil	44	Rosin	10
Paraffin	7		

Correct Dressings for Leather Belting. *Oak-tanned Nonwaterproof* Preferably, compounds supplied by leather-belting manufacturers. Commercial castor oil and neat's-foot oil (3 parts castor to 1 part neat's-foot).

Oak-tanned Waterproof Same as for oak-tanned nonwaterproof.

Mineral-retanned Waterproof. Compounds supplied by leather belting manufacturers only.

Combination Oak Tanned and Mineral Retanned. Compounds supplied by leather belting manufacturers only.

Standard Prices of Leather Belting. The price of leather belting is set at a standard of 24 cents per foot of single ply, 1 in. wide; double or two ply, 48 cents per foot, 1 in. wide. These list prices are subject to current trade discounts.

Rubber Belting

Rubber belting, generally known as "friction-surface rubber," is composed of two or more thicknesses, termed plies, of various weights of cotton duck, impregnated and frictioned with a tough slow-aging rubber compound.

The cotton duck is the element of tensile strength. Therefore the belt is graded to the weight and quality of the duck. For the average rubber belting the duck weight varies from 28 to 32 oz which, according to the United States trade standard, signifies that the actual scale weight of a piece of duck 36 by 42 in. when dry is 28 to 32 oz. The actual weight of the duck is not solely the strength factor because quality, weave, and length of the fiber have a decided effect. The average ultimate tensile strength of 32-oz duck 1 in wide is 440 lb. In calculating the strength of rubber belting, a factor of safety of 20 is employed. The tensile strength of the entire belt is approximately one-third less than the sum of the plies when taken separately.

Table 14-9. Minimum and Maximum Plies of Rubber Belting

Width of belt, in.	Minimum number of plies	Maximum number of plies	Width of belt, in.	Minimum number of plies	Maximum number of plies
2	3	4	16	5	8
3	3	4	18	5	8
4	3	5	20	5	8
5	4	5	22	5	8
6	4	6	24	5	8
8	4	6	26	6	8
10	4	6	30	6	10
12	4	6	36	6	10
14	5	8	42	6	10

Table 14-10. Proper Number of Plies for Regular 32-oz Duck Construction, Based on Pulley Diameter and Speed

Number of plies	Pulley diameters, in		
	Under 2,000 fpm	2,000 to 4,000 fpm	4,000 to 4,500 fpm
3	2	3	5
4	4	6	8
5	10	14	18
6	16	20	24
7	22	26	30
8	28	32	38
9	34	38	44
10	40	44	50

The friction or the grade of the rubber compound employed governs the degree of adhesive tendency between the duck plies and also the coefficient of friction of the belt. This grading is measured in pounds and is based on the quality of the rubber compound. Frictions used are 20, 22, 24, 26, and 28 lb This pound friction is determined by the resistance of the rubber compound to the weight required. If a 28-lb friction is desired, a 28-lb weight or pull is applied to a peeled-back ply of the belt 1 in. wide It must not separate longitudinally more than 1 in. per min. The duck and the friction are the grading characteristics of rubber belting as a 28-oz-28-lb friction belt.

The manufacture of the belt is a series of operations: (1) The duck is thoroughly inspected and dried; (2) it is impregnated and frictioned with the rubber compound—the rubber must be compounded so that the resultant material will be highly elastic tough, and capable of retaining these qualities throughout the transmission life of the belt—(3) the cutting or sizing of the impregnated and frictioned duck to the belt width required; (4) folding and placing the duck strips based on the thickness or number of plies required if of folded construction—if of raw-edge construction, folding is not employed—(5) vulcanizing; (6) inspecting, testing, measuring, talcing, and rolling

Organic accelerators, antioxidants, flex-resisting ingredients, and many other new compounding materials have been developed by the rubber industry. All these improvements have been applied to the bonding material used in rubber-fabric belts. As a result, the quality of the bond has been steadily improved.

Characteristics. *Regular Folded Type and Raw-edge Type*

1. Made in 2, 3, 4, 5, 6, 7, 8, 9, 10, and 12 plies.
2. Made in widths from 1 to 84 in.
3. Ultimate tensile strength of good quality 32-oz duck per in. width equals 440 lb; of 28-oz duck per in. width, 360 lb.
4. Moisture-, water-, and steamproof.
5. Heatproof to 130°F.
6. Will not resist corrosive acids or their mists and vapors.
7. Cannot be penetrated by any belt dressing.
8. Deteriorates when exposed to excessive mineral oil.
9. Average coefficient of friction, 0.35 to 0.45.
10. Can be made endless, preferably by manufacturer.

Horsepower Formula for Rubber Belting (Standard-weight Duck Type). The general expression for horsepower transmitted by a belt is as given for the various types of leather belting; but for belting made of plies of fabric, the effective tension or $(T_1 - T_2)$ is expressed in pounds per inch width per ply of thickness or "pounds per ply inch." Therefore, the formula for duck-ply belting is expressed as follows:

$$\text{Hp} = WNS(T_1 - T_2)/33{,}000$$

where W = width of belt, in.
N = number of plies
S = belt speed, fpm
T_1 = tight-side tension, lb per ply inch
T_2 = slack-side tension, lb per ply inch
$(T_1 - T_2)$ = effective tension, lb per ply inch

Rubber-belt manufacturers have revised and simplified the formula for horsepower capacity in terms of maximum tension T_1. The simplified formula is

$$\text{Hp} = [WNS(T_1 - T_c)/33{,}000]\,[1 - (1/R)]$$

where Hp = hp capacity of belt
W = width, in.
N = number of plies
S = belt speed, fpm
T_1 = tight-side tension, lb per in. width
T_c = centrifugal tension per in. width
$[1 - (1/R)]$ = correction factor for arc of contact (see Table 14-11)

Thirty-two-ounce duck is generally used for good-quality rubber belting. Based on an average ultimate tensile strength for this weight of 440 lb, the allowable effective tension is 25 lb per in. width of duck or a factor of safety of 17½. $(T_1 - T_c)$ can therefore be calculated as 25 lb for a 32-oz duck belt and 20 lb for a 28-oz duck belt.

Initial Stretch of Ply-construction Rubber Belting. Friction-surface ply-constructed rubber belting will stretch initially; therefore under normal operating conditions it should be adjusted occasionally during the early periods. When applying this type of belting, it should be cut short of the actual length necessary as follows:

Three-, four-, and five-ply thickness, cut short $\frac{5}{32}$ in. per foot;
Six-, seven-, and eight-ply thickness, cut short $\frac{1}{8}$ in. per foot;
Nine-, ten-, and twelve-ply thickness, cut short $\frac{3}{32}$ in. per foot.

Table 14-11

Arc of contact, deg	$1 - (1/R)$	Arc of contact, deg	$1 - (1/R)$
90	0 29	190	0 51
100	0 32	200	0 53
110	0 35	210	0 55
120	0 37	220	0 57
130	0 39	230	0 58
140	0 41	240	0 60
150	0 44	250	0 61
160	0 46	260	0 63
170	0 48	270	0 64
180	0 50		

Atmospheric Effects on Rubber Belting. An atmosphere charged with moisture, water, steam, or a temperature of 120°F has no effect.

An atmosphere charged with mists or vapors of sulfuric and nitric acids deteriorates and finally destroys the outer or surface cotton-duck plies.

At temperatures above 120°F the rubber compound tends to dry out and crack.

When exposed to nitric acid the rubber compound becomes slimy, and the cotton disintegrates.

Continuous exposure to mineral oil causes deterioration of the rubber compound

Dressing the Rubber Belt. A rubber belt of any type cannot be penetrated by dressing compounds; therefore it is not possible to lubricate the internal fibers. All compounds applied remain on the surface of the belt; hence caution must be used in the compounds employed. Ply separation and loss of power-transmitting ability are frequently caused by the use of sticky, resinous dressings owing to the belt sticking to the pulley and the accumulation of lumps on the pulley face.

Rubber belting of all types should be dressed periodically with vegetable, castor, or linseed oils. These will remove the rubber "bloom" from new belting and soften the surface of used belting. These oils furnish a nondangerous tackiness or adhesiveness between the belt and the pulleys.

Stitched Cotton-duck Impregnated Belting

Cotton-duck impregnated belting is commonly known as "stitched canvas." It is composed of plies of impregnated cotton duck stitched or sewed together The quality of the belt depends upon the grade and weight of duck, the method of stitching, and the impregnating compound.

FIG. 14-4. Stitched cotton-duck belt—round-edge construction.

Construction. Generally, 36- to 37½-oz duck is employed for stitched cotton duck impregnated belting, having an ultimate tensile strength of 550 to 600 lb per in. width. The duck is cut and folded to the required belting size and stitched firmly together. The structure is impregnated with compounds to furnish durability, lubrication, and frictional value. After impregnation the excess compounds are removed, and the belting lengths are tensioned.

Types. There are two general types, the "round-edge" and the "folded edge" The round-edge construction, shown in section for four-ply and six-ply belting by Fig. 14-4, employs the "interlocked" stitch, particularly above four ply, in addition to the straight-through stitch; while the folded-edge construction, shown in section by Fig. 14-5 for four ply and six ply, employs the straight-through stitch for all thicknesses. The round-edge construction, in view of the interlocked stitch, results in a securely bonded belt and therefore furnishes resistance to ply separation. The round edge resists to a considerable degree the action of shifter forks and also tends to prevent premature cracking of the belt edge.

Table 14-12. Relation of Number of Plies to Pulley Diameters and Speeds

Belt speed, fpm	Pulley diameter, in				
	4 ply	5 ply	6 ply	8 ply	10 ply
Up to 1,000	3	4	6	12	24
1,000–2,000	4	5	8	15	30
2,000–3,000	5	6	10	18	36
3,000–4,000	6	7	12	21	42
Over 4,000	7	8	14	24	48

Characteristics

1. Made in 3, 4, 5, 6, 8, 10, and 12 plies.
2. Made in widths from 2 to 72 in.
3. Coefficient of friction, 0.25 to 0.60.
4. Possesses flexibility, elasticity, and resistance to ply separation.
5. Will hold any type of metallic fastener and can be rawhide or wire laced.
6. Should not be made endless.

Fig. 14-5. Stitched cotton-duck belt—folded-edge construction.

7. Resistance to moisture, water, steam, heat, and slight mineral oil.
8. Should not be used in the presence of the corrosive acids or their mists and vapors.
9. Actual acid contact quickly attacks the duck. Caustic stiffens the entire structure.
10. Temperature up to 140°F has no effect on the better grades of this type of belt. Above 140°F a drying out of the impregnating compound occurs.

Table 14-13. Horsepower Ratings of Good-grade, Stitched-cotton Duck, Impregnated Belting

Belt speed, fpm	Horsepower transmitted by a belt 1 in. wide				
	4 ply	5 ply	6 ply	8 ply	10 ply
200	0.36	0.45	0.54	0.63	0.72
400	0.72	0.90	1.08	1.26	1.44
600	1.08	1.35	1.62	1.89	2.16
800	1.44	1.80	2.16	2.52	2.88
1,000	1.80	2.25	2.70	3.15	3.60
1,200	2.16	2.70	3.24	3.78	4.32
1,400	2.52	3.15	3.78	4.41	5.04
1,600	2.88	3.60	4.32	5.04	5.76
1,800	3.24	4.05	4.86	5.67	6.48
2,000	3.60	4.50	5.40	6.30	7.20
2,200	3.94	4.93	5.92	6.89	7.88
2,400	4.26	5.32	6.38	7.46	8.52
2,600	4.56	5.70	6.84	7.98	9.12
2,800	4.86	6.08	7.29	8.50	9.72
3,000	5.15	6.44	7.73	9.02	10.30
3,200	5.42	6.78	8.12	9.50	10.84
3,400	5.67	7.08	8.50	9.92	11.34
3,600	5.90	7.38	8.86	10.33	11.80
3,800	6.11	7.64	9.16	10.68	12.22
4,000	6.29	7.86	9.43	11.00	12.58
4,200	6.46	8.07	9.69	11.30	12.92
4,400	6.60	8.26	9.91	11.55	13.20
4,600	6.73	8.42	10.10	11.77	13.40
4,800	6.83	8.54	10.25	11.95	13.66
5,000	6.92	8.65	10.38	12.11	13.84
5,200	7.00	8.75	10.50	12.24	14.00
5,400	7.06	8.83	10.59	12.35	14.12
5,600	7.11	8.89	10.66	12.44	14.22
5,800	7.14	8.92	10.71	12.49	14.28
6,000	7.15	8.94	10.73	12.51	14.30

This table, based on an arc of contact of 165 deg is corrected for centrifugal force.

Dressing the Stitched-cotton Duck Belt. The impregnating compounds of good-quality stitched-cotton duck belting serve as a lubricant to the cotton fibers during the life of the belt. When the belt is used in the proper atmospheric conditions, a tacky frictional surface is evident. Other forms of dressings are not required. The pulley surface of the belt should be periodically brushed, particularly when abrasive substances are present.

Camel's-hair Belting

Camel's-hair belting is composed of hair and cotton woven tightly together. It is known as "camel's-hair" belting because the principal power-transmitting medium is the long hair of Bactrian camels. The camel's hair and cotton yarns are woven together under heavy tension, resulting in a solidly woven one-piece fabric. After weaving the fabric is thoroughly impregnated with suitable compounds for the purpose of lubrication and protection.

Characteristics

1. Made in four thicknesses: lightweight, $^3/_{16}$ in. thick; single weight, $^1/_4$ in. thick; double weight, $^9/_{32}$ to $^3/_8$ in. thick, extra-heavy weight, $^9/_{16}$ in. thick.
2. Made in widths from 1 to 41 in.
3. Possesses high elastic qualities and flexibility
4. Remains soft and pliable during entire life.
5. Does not shrink.
6. Ability to absorb shocks.
7. Not affected by mineral oil, unless excessive
8. Moisture-, water-, and steamproof
9. Will resist temperatures up to 300°F
10. Will resist the action of the corrosive acid mists and vapors for long periods but stiffens under the action of caustic

Table 14-14. Relation of Thickness to Pulley Diameters and Speeds

Weight of hair belting	Thickness, in	Minimum pulley diameters, in	
		Up to 2 000 fpm	2 000 to 4 500 fpm
Light	$^3/_{16}$	3	5
Single	$^1/_4$	5	7
Double	$^9/_{32}$ to $^3/_8$	10	14
Extra heavy	$^9/_{16}$	16	24

Table 14-15. Allowable Effective Tensions of Camel's-hair Belting, Pounds per Inch Width

Weight	Thickness, in	Pounds
Light	$^3/_{16}$	40
Single	$^1/_4$	50
Double	$^9/_{32}$ to $^3/_8$	75
Extra heavy	$^9/_{16}$	100

Horsepower Formula and Table. Camel's-hair belting is solid woven, hence the horsepower capacities are based on definite allowable effective tensions per inch of width. The formula is similar to that used for leather belting.

Elastic Qualities. Because of the hair construction, camel's-hair belting possesses considerable elasticity. It should therefore be applied to the pulleys extremely tightly. Because of its high elastic qualities, initial stretch is immediately evident. This comes out rapidly. After one or two take-ups the belt will not stretch beyond its elastic limit.

Dressing the Camel's-hair Belt. The impregnating compounds used preserve and lubricate the structure. The heat of transmission and excessive oil gradually lessen the impregnating compound, therefore this lubricant and preserver should be replaced. Camel's-hair belting should be dressed periodically by special oils or dressings furnished by the manufacturers. No other dressings should be used.

Table 14-16. Table of Horsepower for Camel's-hair Double Belting

Speed, fpm	Width of belt, in.											
	3	4	5	6	8	10	12	14	16	18	20	24
500	3	4	5	6	8	10	12	14	16	18	21	25
700	4	5	7	8	11	14	17	20	23	26	29	35
800	4	6	8	9	13	16	20	23	26	30	33	40
900	5	7	9	10	14	18	22	26	30	34	37	45
1,000	6	8	10	12	16	21	25	29	33	37	42	50
1,200	7	9	12	14	19	25	30	35	40	45	50	60
1,500	9	12	15	18	24	31	37	44	50	56	63	75
1,800	10	14	18	21	29	37	45	53	60	68	75	90
2,000	12	16	20	24	32	42	50	58	67	75	84	100
2,500	15	20	25	30	41	52	63	73	84	94	105	126
3,000	18	24	30	36	49	63	75	88	100	113	126	151
3,500	20	27	34	41	57	73	87	102	117	132	146	175
4,000	23	31	39	46	65	83	100	116	133	150	166	200
4,500	26	35	44	52	73	93	112	131	150	169	187	225

For light or single, multiply ratings given by 0.55, for light-double, multiply ratings given by 0.75; for extra-heavy, multiply ratings given by 1.33 This table is conservative for average conditions; it is based on an arc of contact of 165 deg and is not corrected for centrifugal force

Solid-woven Impregnated Cotton Belting

Solid-woven impregnated cotton belting is made from long-fiber fine-quality domestic cotton woven solidly together to form various thicknesses of belting fabric. To protect the cotton from mechanical and atmospheric effects it is impregnated thoroughly with special compounds.

Table 14-17. Relation of Thickness to Pulley Diameters and Speeds

Thickness	Minimum pulley diameters, in.	
	Up to 2,000 fpm	2,000 to 5,000 fpm
Single	4	6
Heavy single	6	8
Double	8	10
Triple	15	18

Characteristics

1. Made in single, heavy single, double, and triple thicknesses.
2. Made in widths from 1 to 36 in.
3. Alternate wet and dry conditions cause stretch and shrinkage.
4. Ability to absorb shocks.
5. Not affected by mineral oil, unless excessive.
6. Moisture-, water-, and steamproof.

Horsepower Formula and Table. This type of cotton belting is a solid-woven product; therefore the horsepower capacities are based on definite allowable effective tensions per inch of width. The formula is similar to that used for leather belting.

Dressing the Solid-woven Cotton Belt. A solid-woven cotton belt is dressed by the impregnating compounds. They protect and lubricate the cotton fibers. Contact with mineral oil and the heat of transmitting power gradually reduce the compound. It should be replaced by special dressings furnished by the manufacturer. No other dressings should be used.

Table 14-18. Allowable Effective Tensions, Pounds per Inch Width

Weight	*Pounds*
Single	40
Double, up to 8 in	80
Double, 8 in and over	100
Triple	120

Balata Belting

Balata belting is constructed from 36- to 38-oz cotton duck, thoroughly impregnated with a gum called "balata." The gum resembles gutta-percha. The proper-

Table 14-19. Horsepower Table for High-grade Solid-woven Cotton Belting, Based on an Allowable Effective Tension of 100 Lb

Speed, fpm	Width, in.																			
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
500	1½	3	4½	6	7½	9	10½	12	13½	15	16½	18	19½	21	23	24	26	27	29	30
1,000	3	6	9	12	15	18	21	24	27	30	33	36	39	42	45	48	52	55	58	61
1,500	4½	9	13½	18	23	27	32	36	41	45	50	55	59	64	68	73	77	82	86	91
2,000	6	12	18	24	30	36	42	48	55	61	67	73	79	85	91	97	103	109	115	121
2,500	7½	15	23	30	38	45	53	61	68	76	83	91	98	106	114	121	129	136	144	152
3,000	9	18	27	36	45	55	64	73	82	91	100	109	118	127	136	145	155	164	173	182
3,500	10½	21	32	42	53	64	74	85	95	106	117	127	138	148	159	170	180	191	202	212
4,000	12	24	36	48	61	73	85	97	109	121	133	145	158	170	182	194	206	218	230	242
4,500	13½	27	41	55	68	82	95	109	123	136	150	164	177	191	205	218	232	245	259	273
5,000	15	30	45	61	76	91	106	121	136	152	167	182	197	212	227	242	258	273	288	303
5,500	16½	33	50	67	83	100	117	133	150	167	183	200	217	233	250	267	283	300	317	333
6,000	18	36	55	73	91	109	127	145	164	182	200	218	236	255	273	291	309	327	345	364
6,500	19½	39	59	79	98	118	138	158	177	197	217	236	256	276	295	315	335	355	374	394
7,000	21	42	64	85	106	127	148	170	191	212	233	255	276	297	318	339	361	382	403	424
7,500	23	45	68	91	114	136	159	182	205	227	250	273	295	318	341	364	386	409	432	455
8,000	24	48	73	97	121	145	170	194	218	242	267	291	315	339	364	388	412	436	461	485
8,500	26	52	77	103	129	155	180	206	232	258	283	309	335	361	386	412	438	464	489	515
9,000	27	55	82	109	136	164	191	218	245	273	300	327	355	382	409	436	464	491	518	545
9,500	29	58	86	115	144	173	202	230	259	288	317	345	374	403	432	461	489	518	547	576
10,000	30	61	91	121	152	182	212	242	273	303	333	364	394	424	455	485	515	545	576	606

Based on an arc of contact of 165 deg and not corrected for centrifugal force Horsepower capacities for the various allowable effective tensions can be proportioned to this tabulation.

ties that distinguish it from other gums and make it especially valuable as a substance for power belting are its toughness and waterproof qualities. Balata gum will not bond with any foreign material. It will bond with balata only.

Table 14-20. Relation of Number of Plies to Pulley Diameters and Speeds

Number of plies	Minimum pulley diameters, in.		
	Up to 2 000 fpm	2,000 to 4,000 fpm	4,000 to 5,000 fpm
3	3	4	5
4	5	6	8
5	8	10	12
6	12	14	16
8	18	20	24
10	30	34	38

Characteristics of Balata Belting

1. Made in 3, 4, 5, 6, 7, 8, 9, and 10 plies.
2. Made in widths from 1 to 72 in.
3. Possesses low stretch.
4. The duck is impregnated with the gum, not frictioned.
5 Will resist water, moisture, and steam
6 Is dry-heat-proof to 105°F.
7. Balata gum melts at 120°F.
8. Low resistance to the action of mineral oil.
9. Can be made endless by manufacturer.

Dressing the Balata Belt. This type cannot be penetrated by any dressing compound. If the pulling surface becomes dry and hard, vegetable castor oil should be applied sparingly. This oil has a softening effect and produces a "tacky" nondangerous surface on the belt.

Teon Belting[1]

Teon belting is of cotton-duck ply construction. The plies are held together by a special cement reinforced by metallic brads arranged in longitudinal rows at close pitch. This metallic stitching aids in preventing ply separation.

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[1] English manufacture.

Table 14-21. Horsepower Ratings for Balata Belting 1 In. Wide

Speed of belt, fpm	3 ply	4 ply	5 ply	6 ply	7 ply	8 ply	9 ply	10 ply
500	0 60	0 90	1.21	1 51	1 81	2 12	2 42	2.71
750	0 90	1 36	1 81	2.27	2.72	3 18	3.63	4 08
1,000	1.21	1.81	2.42	3 03	3.63	4 24	4.84	5.44
1,250	1.51	2.27	3 03	3.79	4 55	5 30	6.06	6.82
1,500	1.81	2.72	3 63	4.55	5.45	6.36	7.27	8.17
1,750	2.12	3.18	4.24	5.30	6.36	7.42	8.48	9.54
2,000	2.42	3.63	4.85	6.06	7.27	8.48	9.70	10.90
2,250	2.72	4.09	5.45	6.82	8.18	9.54	10.90	12.27
2,500	3.03	4.54	6.06	7.58	9.10	10.60	12.12	13.64
2,750	3.33	4.99	6.66	8.34	10.00	11.66	13.32	14.99
3,000	3.63	5.44	7.26	9.10	10.90	12.72	14.52	16.34
3,250	3.93	5.90	7 87	9.85	11.81	13.78	15.74	17.71
3,500	4.24	6.36	8 48	10.60	12.72	14 84	17.96	19 08
3,750	4.54	6.81	9.09	11 36	13.63	15.90	18.18	20.44
4,000	4.84	7.27	9 70	12.12	14.54	16.96	19.40	21.81
4,250	5 15	7 72	10 30	12 88	15 45	18 02	20 60	23 17
4,500	5 45	8.18	10 90	13 64	16 36	19 08	21 80	24 54
4,750	5 75	8 63	11 51	14 40	17 28	20 14	23 02	25 91
5,000	6 06	9 08	12 12	15 16	18 20	21 20	24 24	27 28

Based on an arc of contact of 165 deg and not corrected for centrifugal force. Ratings based on 38-oz duck. For 36-oz duck, figure 92 per cent of ratings given.

Characteristics of Teon Belting

1. Made in light or three ply, medium, or four ply, heavy or six ply, and extra heavy or eight ply.
2. Widths: light, 1 to 4 in.; medium, 1 to 20 in.; heavy, 3 to 26 in.
3. Resistance to ply separation.
4. Can be made endless by manufacturer.
5. Will resist water, moisture, and steam.
6. Will resist heat to 300°F.
7. Will resist the action of the mists and vapors of the corrosive acids.
8. Will not resist caustic.
9. Will not resist bromic acid.
10. Possesses resistance to mineral oil.

Dressing the Teon Belt. This type cannot be penetrated by any dressing compounds. If the pulley surface becomes dry or glazed, vegetable castor oil or an oil supplied by the manufacturer should be applied sparingly.

Table 14-22. Relation of Number of Plies to Pulley Diameters and Speeds

Weight and number of plies	Minimum pulley diameters, in.		
	Up to 2,000 fpm	2,000 to 4,000 fpm	4,000 to 5,000 fpm
Light, three ply	3	4	5
Medium, four ply..	4	5	6
Heavy, six ply .	9	12	14
Extra heavy, eight ply	16	20	24

Table 14-23. Horsepower Ratings of Teon Belting per Inch Width

Speed of belt, fpm	Lightweight	Medium weight	Heavy weight	Extra-heavy weight
	Horsepower			
500	0.60	0.90	1.51	2.12
750	0.90	1.36	2.27	3.18
1,000	1.21	1.81	3.03	4.24
1,250	1.51	2.27	3.79	5.30
1,500	1.81	2.72	4.55	6.36
1,750	2.12	3.18	5.30	7.42
2,000	2.42	3.63	6.06	8.48
2,250	2.72	4.09	6.82	9.54
2,500	3.03	4.54	7.58	10.60
2,750	3.33	4.99	8.34	11.66
3,000	3.63	5.44	9.10	12.72
3,250	3.93	5.90	9.85	13.78
3,500	4.24	6.36	10.60	14.84
3,750	4.54	6.81	11.36	15.90
4,000	4.84	7.27	12.12	16.96
4,250	5.15	7.72	12.88	18.02
4,500	5.45	8.18	13.64	19.08
4,750	5.75	8.63	14.40	20.14
5,000	6.06	9.08	15.16	21.20

Based on an arc of contact of 165 deg and not corrected for centrifugal force.

SELECTION OF BELTING TYPE

Operating atmospheric conditions to a considerable extent control the type of belting used. The following tabulation is based on actual testing in plant operations and therefore may be used as an authoritative guide:

Table 14-24. Belting-type Selection Based on Atmospheric Conditions

Belting type	Acid spillage	Acid fumes	Abrasive dusts	Steam and moisture	Temp. up to 140°F	Temp. up to 300°F	Oil and grease	Normal
Nonwaterproof oak leather	No	No	Yes	No	No	No	Yes	Yes
Waterproof oak leather	No	No	Yes	Yes	Yes	No	Yes	Yes
Mineral-retanned leather	No	Yes	Yes	Yes	Yes	No	Yes	Yes
Combination-tannage leather	No	No	Yes	Yes	Yes	No	Yes	Yes
Friction-surface rubber	No	No	Yes	Yes	Yes	No	No	Yes
Rubber-covered frictioned duck	No	Yes	No	Yes	Yes	No	No	No
Stitched cotton duck	No	No	Yes	Yes	Yes	No	Yes	Yes
Solid-woven cotton	No	Slight, yes	Yes	Yes	Yes	No	Yes	Yes
Camel's hair	Yes, slight	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Balata	No	Yes, slight	Yes	Yes	No	No	Yes	Yes
Teon	No	Yes	Yes	Yes	Yes	Yes	No	Yes

MAINTENANCE OF POWER BELTING

1. Belting when employed serves as a connecting medium between power source and application. It is equal in importance to the operating machines and the power plant. Production depends upon it.

2. Systematic care and attention are given to the power-source and production machines. Similar care and attention to power belting are absolutely necessary if efficiency of operation and maximum production are to be maintained.

3. The cost of repairs to a belt may be small, but the enforced idleness of men and the production loss based on such repairs are expensive. The value of a machine as a producer exceeds the wages paid the men who operate it. A slipping belt can cause heavy expense through loss of applied power.

4. Consideration of belting should begin with the design and installation of the transmission system. Belting of the proper type should be chosen for the atmospheric and mechanical condition involved. It should be of the proper type and size to transmit its load. A belt larger than necessary will more than repay in increased life and lessened maintenance expense the additional cost over a smaller overloaded belt.

5. Properly installed belting requires little attention, provided it is given regular and systematic care. Neglect and abuse are generally responsible for belting difficulties.

6. Belting maintenance should be centralized. One man should be held responsible. The man designated should make periodical inspections of all belt drives so that breakages, slip, and other difficulties may be anticipated and therefore avoided.

7. Belting should be maintained in proper physical condition so that it can transmit its capacity load without slip.

8. Belting requires dressing or lubricating; therefore it is important that the dressings be of suitable character to accomplish the results desired. Such dressings should be capable of relieving the internal friction of fiber on fiber, mellowing the belt and keeping it pliable, and furnishing a cohesive but not sticky grip on the pulleys.

9 Power is wasted in tearing a sticky belt free from the pulleys, and such tearing action is destructive to the belt.

10 When in proper condition, a belt will grip the pulleys with a clinging suction action and will release without adhesion. When so maintained the belt will carry its load without high initial tension. High tension has no value or purpose other than to eliminate slip.

11 Belting lubricating must be considered from the standpoint of the two general classes of belting. One class, such as leather, stitched canvas, solid-woven cotton, and camel's hair, consists of a porous structure which will absorb lubricants; while the other class, such as rubber, balata, and teon, are impervious to the absorption of lubricants. The dressings or lubricants for both classes of belting can be the same, but the method of application must be different. Belting having absorbing qualities should be treated sparingly at regular intervals both front and back. Approximately 60 per cent should be through the back. Belting of the nonabsorbent class should be given surface treatment only. This treatment should be very light and applied only as required to maintain a good friction and prevent polishing and glazing.

12 Belting of the nonabsorbing class is subject to cracks and pinholes. When operating in dampness, moisture will enter, become trapped, and will eventually rot the cotton duck. A proper surface dressing will prevent the entrance of moisture.

ESSENTIALS FOR EFFICIENT BELTING SERVICE

1 Pulleys should be slightly wider than the belt that they are to carry. This will prevent the belt's running over the rim and striking adjacent objects. The pulley should not have a sharp or too much crown.

2. Belting joints should be carefully made. The belt ends should be absolutely squared, so that the belt can run true and not stretch unevenly. Heavy lacing, which results in "bunches" at the joints, and oversized metallic fasteners cause loss of pulley contact, breaking of the belt at the joint, stiffness, and thumping of the belt against the pulley. A projecting fastener striking an obstruction or a badly adjusted shifter fork will cause rapid destruction of any belt.

3. Pulleys and shafting should be kept in perfect alignment. Much belting is ruined by neglect of this particular factor.

4. The laps of leather belting should be inspected frequently. As soon as the points of the laps show signs of lifting, they should be immediately cemented down.

5. Any defects developing at the joints should receive prompt attention. If lacings, hooks, or plates have damaged the ends of the belt, immediate attention should be given; otherwise the belt will give way and a shutdown result.

6. Belting should be kept clean. A coating of dirt will destroy its gripping quality. Dirt working into a belt is similar to sand in a bearing. It causes frictional heat which destroys the belting fibers.

7. Leather, stitched canvas, solid-woven cotton, and camel's-hair belting can be washed with a mixture of two parts gasoline to one of turpentine to loosen the dirt and then scraped with a dull knife. After drying, the proper lubricant or dressing must be applied.

8. Rubber, balata, and teon belting should be washed with soap and water only. After washing they should be carefully scraped and the proper surface dressings applied.

9. An oil-soaked leather belt should be immersed in the gasoline and turpentine cleaning mixture as above and allowed to soak for several hours. After drying, the proper lubricant or dressings should be applied.

10. Continuous stretch signifies that the belt is not strong enough for the load. A wider belt should be used. If this is not possible, use a thicker one, provided the thickness conforms to pulley diameter and speeds.

11. Any belt when overloaded and slipping is likely to slide off the pulleys. A belt will slide off a crowned pulley more easily than from a flat pulley. The remedy is to stop the belt's slipping by cleaning, dressing, tightening, or providing a wider or thicker belt.

12. If a belt is uniform with joint square and the pulleys are true and in line, it will not tend to work to one side of the pulleys on a horizontal shaft.

If a belt which is not overloaded is causing trouble by working to one side of the pulley, first make sure that the shafts and pulleys are perfectly in line. If the defects still persist when perfect alignment is obtained, the belt can be made to move to the desired side by drawing the shafts together on that side.

FASTENING OF POWER BELTING

Joining a power-transmission belt in an improper and careless manner results in high belting-maintenance costs, short belting life, and production loss.

Belt-joining Fundamentals

1. The belt should not be strained at the joint.
2. The fastener should not cut or weaken the belt.
3. The fastener should conform to the curvature of the pulley.
4. The fastener should resist wear caused by pulley contact.
5. The correct size and type of fastener should be used for pulley diameters and speeds.
6. The fastener should not prevent the belt being square and close fitting at the joint.
7. If metallic, the minimum amount of metal should be at the joint.
8. The fastener should be quick of application.

General Methods of Belt Joining

1. Endless
2. Laced
3. Metallic

The Endless Method

The endless method is the most efficient belt-joining method, since the joint is integral with the belt. Initial stretch is evident in all types of belting; therefor center adjusting facilities should be provided when this method is employed. Th leather belt offers the greatest possibility for the endless method because it is a multiple of cemented joints.

Making the Leather Belt Endless. *Regular Oak Nonwaterproof.* Scarf or scrape the ends down to a thin edge 3, 4, or 6 in. back, depending upon the width and thickness of the belt, and apply the hot glue to the two scarfed surfaces. Place the glued surfaces together and subject to continuous pressure for at least $2\frac{1}{2}$ hr.

Waterproof Oak, Mineral Retanned, and Combination Tannage. Scarf or scrape the ends down to a thin edge as described above and size the two scarfed surfaces first with a waterproof pyroxylin cement; allow this to dry. Apply another coating of the cement to the sized surfaces, place them together, and subject them to continuous pressure for at least 6 hr.

Uses of the Endless Belt

1. On motor drives when the motor is provided with slide rails
2. On pivoted motor drives
3. On motor drives, automatic-idler-pulley controlled
4. On high-speed spindle operations of machine tools when idler pulley control or center adjustment is possible
5. On adjustable mule-pulley drives
6. On large-powered engine drives

Directions for the Use of Regular Nonwaterproof Leather Belt Cement. Place the can in hot water and heat it. Thin to the consistency of thick syrup, and apply while hot in a thin coat, well rubbed in with a brush. Put the surfaces together quickly before the cement chills, rub the outside of the joint with a block of wood, and tap lightly with a hammer. *Have the cement hot, but do not allow it to boil.*

Directions for the Use of Waterproof Leather Belt Cement. *Caution: Keep this cement away from flames or cigarettes, etc.* Before applying cement, any grease or oil should be removed with gasoline or carbon tetrachloride, and all the old cement must be cleaned off. After preparing the splice, make sure that the laps are absolutely of even thickness in order to obtain a smoothly running belt. Roughen the surface with card cloth if available. Apply two coats of cement, using a brush, letting each dry 15 to 30 min before applying the next. After applying the third coat, immediately clamp the splice between boards with C clamps and let it stay 6 hr. Hammering the laps down instead of clamping is not acceptable.

The Laced Method

The laced method of joining belting employs rawhide or wire and is applicable to all types of belting.

Rawhide lacing is applied manually and results in an efficient and strong joint. The time required for application must be considered based on other equally efficient methods available. Rawhide lacing should always be employed on hand-shifted belting. It cannot injure the hands.

Wire lacing is applied either by machine or by hand and results in an efficient, strong, and flexible joint. There are two machine methods: one actually sews the ends of the belt together, while the other forms wire loops through which a pin of either rawhide or fiber is inserted.

Metallic Fasteners

Wire Hook with Rawhide or Fiber Pin. *Characteristics*

1. Easy to apply by either portable or stationary machine.
2. Results in an accurate and flexible joint.
3. Proper-size selection gives minimum amount of metal at joint.
4. Pressed into ends of belt so as to form a series of loops.
5. Loops are meshed together, and either rawhide or fiber pin is inserted.
6. Hooks are furnished on cards.
7. Hooks are of plain steel, stainless steel, bronze, or monel metal.

Pressed Steel, Hinged. *Characteristics*

1. Manual application.
2. No special machine or tools required.
3. Applied with a hammer.
4. Results in a strong and accurate joint.
5. Furnished in strips with jaws open.

Table 14-25. Minimum Pulley Diameters and Belt Thicknesses for Wire Hooks*

Standard hook size	Belt type and thickness	Minimum pulley diameter, in.
2	2-ply fabric and light single leather	2–3
3	3-ply fabric and medium single leather	3–4
4	4-ply fabric, heavy single and light double leather	4–5
5	5-ply fabric, light and medium double leather	6–8
6	6-ply fabric and heavy double leather	8–12

* Courtesy of the Clipper Belt Lacer Co.

Table 14-26. Minimum Pulley Diameters and Belt Thicknesses for the Pressed-steel Hinge-pin Fastener*

Manufacturers' size	Belt type and thickness	Minimum pulley diameter, in.	Standard length of lacing, in.
20	3-ply fabric	3–5	12
L-27	4-ply fabric and heavy single-oak leather	6–7	12
M-35	5-ply fabric and light double-oak leather	8–11	8
U-45	6-ply fabric and medium and heavy double-oak leather	12–14	12
X-65	7- and 8-ply rubber, cotton, or balata	14 16	12

* Courtesy of the Alligator Belt Lacer Co.

6. Grips belt from both sides without weakening the leather or fabric.
7. Two hinge pins employed, designed to furnish rolling action at joint.

Bolted-plate Fastener. *Characteristics*

1. Manual application.
2. Punch, hammer, and wrench required.
3. Light canvas or leather between belt and plates recommended.
4. Specially designed bolts and nuts used for securing plates to belt.
5. Results in a strong and efficient joint.
6. Plates are designed to conform to curvature of pulley.
7. Pulley-surface washer designed to embed itself in belt.
8. When applied correctly, washer does not come in contact with pulley face.
9. Screw the bolt nuts tight.

Table 14-27. Plate Size, Based on Belt Width

Size of plate	Belt width, in.	Diameter of bolt, in.
0	1½–2	7/32
1 small	2½–4	¼
1 large	5–6	9/32
2 small	7	5/16
3 small	10–16	⅜
3 large	17–20	13/32
4	21–24	7/16
5	Above 24	½

Steel-prong Hook. *Characteristics*

1. Manual application.
2. No special machine or tools required.
3. Applied with hammer.
4. Results in strong and efficient joint.
5. Prongs bend flush with pulley side of belt.
6. Belt must be against hard surface when hook is driven through.
7. Is quick of application.

Riveted Steel-plate Fasteners. *Characteristics*

1. Manual application.
2. No special machine or tools required.
3. Plates secured to belt by specially designed rivets.

Table 14-28. Plate Sizes and Number to Use for Transmission, Conveyor, and Elevator Belting*

Belt width in	Light high-speed work	Light work	General work		Heavy work	Extremely heavy work
	2 in diameter of smaller pulley	3 in diameter of smaller pulley	6 in diameter of smaller pulley	9 in diameter of smaller pulley	12 in. diameter of smaller pulley	24-in diameter of smaller pulley
	High-speed plates	Short grip plates	Medium grip plates	Special-grip plates	Long-grip plates	Jumbo plates
½	One 20					
¾	One 20					Used also on conveyor and elevator belts on pulleys smaller than 24 in where speed is not excessive
1	One 40	One 25				
1½	One 44†	One 45	One 67			
2	One 60	One 65	One 607	One 63		
2½	Two 44	One 85	One 87	One 83		
3	Two 44	One 805 or two 45	Two 67 or one 107	One 103		
3½		Two 45	One 127	One 123		
4		Two 65	Two 607 or one 147	Two 63	One 189	
5			Two 87	Two 83	Two 109	
6			Four 67 or two 107	Two 103	Two 149	
7			Two 127	Two 123	Two 1 409	
8			Four 607 or two 147	Four 63	Two 189	Four 1 011
9			Three 107	Three 103	Three 149	Four 1 011
10			Four 87	Four 83	Four 109	Five 1 011
12				Four 103	Four 149	Four 1 611
14				Four 123	Four 1 409	Four 1 611
16		For belt widths wider than 24 in use sufficient plates of the same grip series to cover the width of the belt completely			Four 189	Four 2 211
18					Six 149	Six 1 611
20					Eight 109	Five 2 211
24					Six 189	Six 2 211

* Courtesy of the Crescent Belt Lacer Co

† Where smaller pulley is 4½ in diameter or larger use No 06, covers 1½ in of belt width

4 Easily applied by a hammer
5 Results in a strong and efficient joint
6 Combines strength with light weight
7 Distributes power strain evenly across belt
8 Rivets spread flush with pulley side of belt when driven through.
9 Belt can be joined without removal from pulley
10 Prongs of rivets easily penetrate leather or fabric belting
11 Plate is designed to conform to curvature of pulley (see Table 14-28)

Note The plate fastener owing to its construction forms a nonflexible joint therefore the plate used must conform to the curvature or circumference of the pulley If the plate does not conform, premature breakage of the belt will occur immediately back of the plate

POWER TRANSMISSION BY BELTING

Initial Tension. Initial tension is measured in pounds per inch width and maintains the contact between pulley and belt necessary to pull the load with a minimum of slip with given friction properties of belt and pulley This tension must be increased as the power load increases, as the friction coefficient is reduced by foreign substances, as the stretch in the belt produced by the load and centrifugal

force loosens the belt on the pulley, and as the arc of contact decreases by high ratios, or where a stiff or thick belt operates at high speed over a small-diameter pulley.

Effective tension, signified in belting formula as ($T_1 - T_2$), is also an important factor in belting engineering. Figure 14-6 illustrates a belt weighted at each end and hung over a pulley. When the weights T_1 and T_2 are equal, the pulley will not revolve. This is equivalent to a power-transmission belt at rest. When the weight T_1 is heavier than the weight T_2, the pulley will revolve in the direction shown by the arrow. This is equivalent to a belt transmitting power. It is the difference in the weight of T_1 and T_2 which measures the turning force or torque applied to the pulley. This weight or tension difference is the effective power tension.

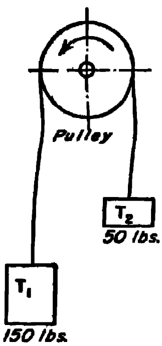

Fig. 14-6.

Computation of T_1 and T_2. To obtain the values for tight side T_1, slack side T_2, and total initial tension for a given load and speed, assume a 25-hp belt drive with a belt speed of 3,000 fpm, drive horizontal.

$$\text{Belt-power pull} = \frac{(25\text{ hp} \times 33{,}000\text{ lb})}{3{,}000\text{ fpm}} = 275\text{ lb}$$

The belt-power pull, or in this case 275 lb, is not the tight-side or T_1 tension because the coefficient of friction of the belt and the arc of contact have an influence.

To obtain the tight-side tension T_1, the power pull must be multiplied by a factor based on the coefficient of friction and the arc of contact given in Table 14-29. These values are based on oak-tanned leather on cast-iron pulleys.

Table 14-29. Tension Factors

Arc of contact, deg Smaller pulley	130	140	150	160	170	180	190	200	210	220	230
Horizontal drive	2.00	1.90	1.80	1.70	1.65	1.60	1.55	1.50	1.47	1.44	1.40
Vertical drive	2.10	2.00	1.90	1.80	1.75	1.70	1.65	1.60	1.57	1.54	1.50

In the above example assume the arc of contact on the smaller pulley to be 160 deg; then

$$T_1 = 275 \times 1.70 = 467\text{ lb}$$
$$T_2 = 467 - 275 = 192\text{ lb}$$

$$\text{Total static or initial tension} = 467 + 192 = 659\text{ lb}$$

Note: The preceding tension factors in Table 14-29 are calculated on a basis of approximately 0.31 coefficient of friction, which is a fair average for a good oak-leather belt on two moderately large cast-iron pulleys. If the pulleys were small and both of similar diameter resulting in 180 deg arc of contact, the horsepower capacity of the belt would be somewhat reduced. Inversely, if the two pulleys were of large diameter, it would be increased. Composition pulleys, such as paper, fiber, or leather, will increase the pulling power of the belt about 10 per cent. The nature of the drive itself can influence greatly the ability of the belt to pull the load. If the load is jerky and highly fluctuating, the total pulling capacity of the belt is proportionally less than for drives where the load is constant and steady. For these reasons a factor of safety should be used over theoretical calculations, and the determination of such a factor, which may vary from 25 to 100 per cent, is a matter of judgment.

Arc of Contact. This is an important factor, since the more wrap or contact a belt has on the driving pulley, the more power it will transmit. Regardless of the fact that 165 deg is generally assumed as a fair average for industrial driving and that belting-horsepower tables are calculated on this basis, the arc of contact must be considered where accurate calculations are necessary. The greater the arc of contact, the lower the belt tension and belt size. The horsepower ratings given in Tables 14-2 to 14-7 are suitable for the average driving, but where accuracy is necessary they should be corrected as follows:

The factor *R* is found by dividing the diameter difference of the pulleys in inches by their center distance in feet. By referring to the curve chart, Fig. 14-7, and with the factor *R* found, the arc of contact in degrees on the smaller pulley can be determined. With the arc determined, the correction factor for the table ratings can be found in Table 14-30.

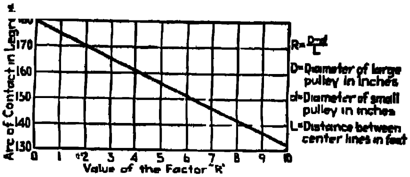

FIG. 14-7. Arc-factor curve chart.

EXAMPLE: Assume the arc of contact from the curve chart (Fig. 14-7) to be 130 deg; then the arc factor from Table 14-30, based on an arc rating of 130 deg would be 0.85. Assume 8 in. medium double-oak leather belt at 3,000 fpm. From Table 14-3 a belt of this character at this speed will transmit 47 hp at 165 deg arc of contact. On an arc of contact of 130 deg, it will transmit 47 × 0.85, or 39.9 hp.

Coefficient of Friction. All belting fundamentally transmits power by frictional contact between the belt and the pulley. This value varies with the types of belting and pulleys employed.

Table 14-30

Arc of contact, deg	Arc factor	Arc of contact, deg	Arc factor
120	0.81	170	1.02
125	0.83	175	1.03
130	0.85	180	1.05
135	0.87	185	1.07
140	0.90	190	1.09
145	0.92	195	1.10
150	0.94	200	1.12
155	0.96	205	1.13
160	0.98	210	1.15
165	1.00	215	1.17

Centrifugal Tension on Belting. Each particle of a belt as it moves in a curved path around a pulley exerts a centrifugal force which acts straight away from the center of the pulley. These forces combine to create a tension which is felt throughout the length of the belt. It is termed "centrifugal tension" and is in addition to any other existing belt tension. A diagram illustrating the action of centrifugal tension on a horizontal drive is shown by Fig. 14-8.

This action stretches and subjects the belt to a tension which lifts it away from the pulleys and therefore decreases its tractive grip upon them. On heavy belting at high speeds the centrifugal tension may equal the effective tension with the result that power transmission is negligible.

Formula for Centrifugal Tension:

$$T = \frac{W \times S^2}{116,000}$$

where T = centrifugal tension
W = weight of belt, lb per ft
S = speed of belt, fpm

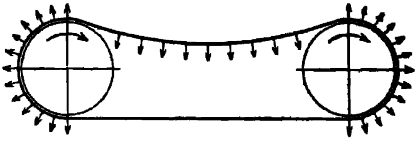

FIG. 14-8. Effect of centrifugal tension.

NOTE: Centrifugal tension is governed by the square of the speed. Doubling the speed increases it four times, and tripling it increases it nine times. Therefore very high-speed belts act at a disadvantage. Since the centrifugal tension is also governed by the weight of the belt, it is evident that, where high belt speeds are unavoidable, it is important to use the lightest weight belt that will do the work.

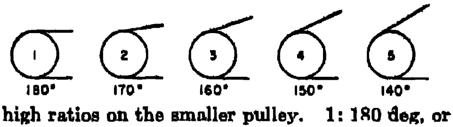

FIG. 14-9. Effect of high ratios on the smaller pulley. 1: 180 deg, or 100 per cent power rating on this arc. 2: 170 deg, or 4 per cent off power rating of 180 deg. 3: 160 deg, or 8 per cent off power rating of 180 deg. 4: 150 deg, or 13 per cent off power rating of 180 deg. 5: 140 deg, or 17 per cent off power rating of 180 deg.

Table 14-31. Horsepower Losses Due to Centrifugal Tension

Speed of belt, fpm	*Percentage of rated horsepower*
1,000	1
2,000	4
3,000	8
4,000	15
5,000	23
6,000	34
7,000	46
8,000	60
9,000	76
10,000	95

NOTE: These losses are applicable to belting running without automatic idler or pivoted motor base control.

Table 14-32. Minimum Center Distances for Belt Drives Based on Power and Ratios

Horsepower to be transmitted	Ratio of driver to driven	Center distance, ft
1-4	2-1	4
	3-1	5
	4-1	7 5
	5-1	10
5-9	2-1	
	3-1	6
	4-1	16 5
	5-1	12
10-14	2-1	8
	3-1	10
	4-1	12
	5-1	14
15-24	2-1	9
	3-1	11
	4-1	13
	5-1	15
25-39	2-1	10
	3-1	12
	4-1	14
	5-1	16
40-49	2-1	12
	3-1	13
	4-1	15
	5-1	17
50-74	2-1	13
	3-1	14
	4-1	16
	5-1	18
75-99	2-1	18
	3-1	20
	4-1	22
	5-1	24
100-124	2-1	20
	3-1	22
	4-1	24
	5-1	26
125-149	2-1	22
	3-1	24
	4-1	26
	5-1	28
150-200	2-1	26
	3-1	28
	4-1	30
	5-1	32

NOTE: These center distances are based on actual experience. The figures should be employed on belted motor drives and all manner of belting installations where reasonable pulley diameters are used. Belt drives from engine flywheels and powers above 300 hp by belting are now exceptional; therefore for such service operating conditions must be considered.

Driving Ratios. The relation of the diameter of the driving pulley to the diameter of the driven one controls the speed ratio and the arc of contact of the belt on the driving pulley or the smaller pulley of the two. Without the use of an idler pulley a ratio of 1:1 furnishes a maximum arc of contact of 180 deg on open drives and relatively more on a crossed-belt drive. For efficient service a ratio of 6:1 should not be exceeded. Figure 14-9 illustrates the effect of various ratios on the arc of contact of the belt on the small pulley.

Center Distance. Belt drives without the use of automatic idler or pivoted-base motor control should have center distance proportioned to the pulley ratios, speed, and load requirements Short centers necessitate high-tensioned or "tight" belting, resulting in hot bearings, belt slippage, short belting life, and loss of power (see Table 14-32).

Vertical Drives. A belt operating vertically without center adjustment or idler control is a source of high maintenance and power loss. The stretch of the belting substance causes the belt to drop away from the bottom pulley. If a vertical drive is necessary and take-up facilities are not possible, the center distance should be comparatively short, based on the least amount of accumulated stretch per foot. Adjustable centers should be used, or a controllable idler pulley placed against the loose side of the belt, or a vertical-type pivoted motor base used. An angle of 15 deg from the vertical will help to eliminate vertical-drive difficulties.

Crossed-belt Drives. Crossed belting is used to reverse the direction of rotation and to increase the arc of contact. It is practicable, but the following factors must be considered:

1. **The ratio of a crossed drive should not exceed 4:1.**
2. **The cross should occur as close to the center between the pulleys as possible.**
3. **The width of the belting should not exceed 8 in.**
4. **The fastener must not interfere at the point of crossing.**
5. **Extremely short center drives should not be crossed.**

Quarter-turn Drives. A quarter-turn belt drive is employed to transmit power at right angles in a vertical, horizontal, or angular direction. It is a source of trouble and should not be used unless absolutely necessary. For best results a leather belt specially constructed for quarter-turn driving should be used. The quarter-turn drive should be installed correctly.

Rule for Installing Quarter-turn Drives. The center of the face of the loose side of the driver must line with the center of the face of the tight side of the driven.

Alignment of Shafting and Pulleys. In belt driving of any character, misaligned shafting and pulleys cause loss of power, high maintenance, and short belting life. Initially an installation may be in perfect alignment. During operation it should be checked periodically, based on the settling of foundations, expansion and contraction of woodwork, intermittent application of load, and machinery wear. Misalignment causes crooked-running belting and therefore slip, resulting in power loss and premature belt destruction, hot bearings, and excessive friction loads. The centers of the faces of the pulleys should be perfectly in line, and the shafting should be parallel.

Belt Slippage. Loss of power and belt destruction are primarily caused by belt slip. The following factors must be considered to avoid this action:

1. Type and thickness of belt not suitable for pulley diameter and speeds
2. Overload and underload
3. Insufficient initial belt tension
4. Dry, glazed, and hard belting; lack of proper dressing
5. Formation of lumps on pulley face by use of resinous dressings
6. Atmospheric effects
7. Misalignment
8. Improper joining

Shifting of Belting. Frequent or periodic sliding of a belt on tight and loose pulleys subjects it to edge wear and considerable transverse stress and crumpling action. Such strains and crumpling are more pronounced in wide belting regardless of thickness. To avoid such action, single leather and four-ply or equal-thickness fabric should not exceed 5 in. in width; and double leather or six-ply or equal-thickness fabric should not exceed a width of 8 in. when employed on shifting installations. Edge wear is primarily caused by wrongly adjusted and rough shifter forks. The fork should be smooth and straight and come in contact with the belt only when shifting. The roller-type shifter fork eliminates edge wear.

Pulley Diameters and Revolutions

Problem 1. Diameter and rpm of driver and diameter of driven known; to find rpm of driven.

Formula: $$\frac{\text{Diameter of driver} \times \text{rpm}}{\text{Diameter of driven}}$$

Problem 2. Diameter and rpm of driven and diameter of driver known; to find rpm of driver.

Formula: $$\frac{\text{Diameter of driven} \times \text{rpm}}{\text{Diameter of driver}}$$

Problem 3. Rpm of driver and diameter and rpm of driven known; to find diameter of driver.

Formula: $$\frac{\text{Diameter of driven} \times \text{rpm}}{\text{Rpm of driver}}$$

Problem 4. Rpm of driven and diameter and rpm of driver known; to find diameter of driven.

Formula: $$\frac{\text{Diameter of driver} \times \text{rpm}}{\text{Rpm of driven}}$$

Note: When pulley diameters result in fractions of an inch, the nearest whole number should be used.

FLAT-BELT PULLEYS

For efficient and economical power transmission by belting, the proper type and design of pulley should be used for the operating mechanical and atmospheric conditions.

Pulley Types. Six types of pulleys are available: paper, cast iron, fabricated steel, all wood, wood rim with cast-iron center, and metallic center with paper rim. Each type is manufactured in a number of designs to suit industrial requirements.

Pulley Essentials. The following factors should be considered when selecting type and design:

1. Frictional value of face surface.
2. Face must be smooth.
3. Ability to resist overload and shock.
4. Ability to conduct heat.
5. Solid or split design; split design facilitates installation and removal.
6. The pulley should be true in diameter.
7. Minimum air resistance.
8. Ability of material to resist steam, moisture, acid drip, and acid fumes.
9. Whether flywheel effect is required.
10. Should face be crowned or flat?
11. Do power and speed require a keyed or clamped pulley?
12. Interchangeability of bore.

Pulley Specifications. Pulley ordering will be facilitated by observance of the following:

1. *Service.* Specify character of service, horsepower requirements, and revolutions per minute.

2. *Description.* State whether solid, split, clamp hub or keyed, bushed or straight bored, tight or loose, flanged or special.

3. *Diameter.* Specify diameter in inches. This should be the first dimension given. If exact diameter is required, mention this and state whether measurements shall be made at crown or edge of rim. Unless otherwise specified, the diameter of a pulley is the diameter at the top of the crown.

4. *Face.* Specify face in inches. This should be second dimension given and should be in accordance with the width of the belt unless an exact width of face is desired. If exact face is required, the word "exact" must follow the face-width dimension.

5. *Bore.* Specify exact diameter of shaft in inches. This should be the third dimension given.

6. *Crown or Straight Face.* Specify whether crown or straight face. Crown face will be furnished if no specification is given. Drum pulleys for shifting belts have straight face. Tight and loose pairs have crowned faces.

7. *Key Seat or Setscrew.* Specify whether pulley is to be keyseated or setscrewed, or both.

Relation of Width of Pulley Face to Width of Belt. The face width of a pulley is determined by the required belt width. A pulley specification of 24 by 6 in. indicates a pulley having a crown-point diameter of 24 in. and sufficient face width to accommodate a 6-in.-wide belt. A 24- by 6-in. pulley will actually measure approximately 6½ in. width of face. This is standard practice for the purpose of securing full face belt contact. Up to and including 10-in. belt width, ⅜ to ½ in. wider pulley face is provided. From 12-in. belt width up, the pulley face is approximately 1 to 1½ in. wider than the belt.

Relative Speed Capacities of Pulley Types in Feet per Minute. *Cast-iron Type.* The cast-iron pulley is designed for rim speeds of 3,500 to 4,000 fpm. Rim speeds of 4,500 to 5,500 fpm are permissible and possible, but cast-iron pulleys to run at such speeds must be specially built and perfectly balanced. The cast-iron pulley can be employed for any practical power requirement.

Pressed-steel Type. Fabricated construction of the pressed-steel pulley permits with safety rim speeds of 4,000 to 5,000 fpm. This type, under test, has been operated at 14,000 fpm for 30-hr periods without failure. Such speeds, however, are not practicable for industrial driving. The pressed-steel pulley can be employed for any practical power requirements.

Wood Type. The all-wood pulley or the wood-rim pulley with cast-iron hubs and arms can be run with safety at rim speeds of 5,000 to 6,000 fpm. This type can be employed for any practical power requirement.

STEEL SHAFTING

Shafting is an important factor in mechanical power transmission. All methods employ shafting; therefore success or failure of the installation depends largely upon the proper selection of shaft size and the grade of steel employed.

Shafting for "group" power-transmission purposes, such as head-, line-, counter-, and jackshafting, is cold-finished, low-carbon steel of screwstock quality. Sizes up to about 2¾ in. are cold-drawn, while larger sizes are turned and polished. The physical properties of such steels vary and cannot be guaranteed; therefore the working stresses recommended are comparatively low. Shafting of large size or those requiring uniform physical properties are hot-rolled or forged and turned to finished size. The limits and methods vary by different manufacturers.

Cold-finished Steel Shafting. Shafting designated as "cold-finished" indicates the method of obtaining the finish and not the type of steel or its physical properties. To produce the shafting in sizes 2¾ in. diameter and under, the hot-rolled bars, after pickling for scale removal, are cold-drawn to size through tool-steel dies. The compression of the surface in cold drawing strengthens the steel to a certain depth. As the proportion of this strengthened surface to the total area increases, so will the tensile properties improve with decreasing size. When cold-finished steels are machined, thereby cutting the strengthened surface, the tensile increase is lost to a considerable extent. This tensile improvement is small on sizes from 2 to 2¾ in. in diameter. Shafting above 2¾ in. in diameter is usually turned and polished from hot-rolled bars, therefore the physical properties of such increased sizes are the same as for hot-rolled steel. The expression cold-finished may apply to alloy or straight-carbon steels as well as to screwstock. Specifications therefore should state not only the character of the finish but also the type of steel. Cold-finished medium-carbon and alloy steels are stronger than screwstock in the same proportion as hot-rolled medium-carbon and alloy steels are stronger than hot-rolled screw or machine steel (see Table 14-33 for physical properties).

Hot-rolled Steel Shafting. This type owes its superior strength to analysis and thermal treatment. Plain carbon steels such as SAE 1020, 1045, etc., become stronger in tensile strength with increasing carbon content. Above approximately 0.50 carbon, steels of this character are difficult to machine without annealing. Furthermore, if these steels are annealed, their strength is lower, and their cost is more than that of lower carbon steels or those of the medium-carbon, high-manganese type. This also applies to the alloy steels such as nickel, chrome nickel, chrome molybdenum, or chrome vanadium. Alloy steels such as these should never be employed in the annealed or untreated condition, since untreated alloy steels are more expensive, more difficult to machine, and their physical properties are no better than medium-carbon steels of the high-manganese type. Alloy steels when used should be quenched and drawn (see Table 14-34 for comparative physical properties of the hot-rolled steels).

A characteristic that is particularly applicable to hot-rolled and untreated steels whether carbon or alloy is that with increasing size, tensile strength and yield point drop with no increase in ductility. Therefore, when a size selection is being made, allowance should also be made for this fact. When physical properties of such steels are specified they should be made in connection with size from which test specimens are taken. When physical properties of heat-treated alloy steels are stated, they must be allied with definite sizes. For example, assume that a 5-in.-diameter shaft is desired with a tensile strength of 140,000 lb; the steel selected must possess the required tensile strength in 5 in. round and not as specified for ¾ or 1-in.-diameter stock-test specimens.

Tensile strength indicates the static strength of the steel. The amount of repeated stress endured without breakage (fatigue strength) is proportional to tensile strength. The shear resistance, torsional strength and hardness by Brinell, Rockwell, or Shore are also proportional to this property. Hardness tests by any of the foregoing methods have not provèd reliable as an index for the two important qualities of machinability and abrasion resistance. These qualities are apparently dependent on structure rather than hardness.

Alloy Steels. Alloy steels should always be heat-treated, either before or after machining. For transmission shafting it is not practicable to heat-treat after machining; therefore when alloy steel is necessary for this service, the preheated type should be employed. An attribute of the alloy steels is that they can be hardened after machining; thus they are adaptable for parts requiring a greater surface hardness than can be machined. For all purposes, with the exception of carbonizing, the alloy steels should be of 0.30 to 0.55 carbon content, since such a carbon content enables them to respond properly to heat-treatment. In these steels carbon content usually increases with size.

Yield point indicates the actual working limit of the steel and is the figure to employ when determining shaft size or section required to resist a given load or stress It is usually considered comparable to compressive strength and therefore is particularly adaptable for calculations of high-pressure or shock-resisting parts such as gearing, machine shafting, and sprockets.

Elongation and Reduction. Elongation and reduction are measures of ductility, shock resistance, and ability to stand sudden overload. If these figures are low, the steel may be termed "brittle."

Modulus of elasticity indicates the degree of deflection under a given load. The modulus 29,000,000 to 31,000,000 psi is essentially the same for any kind or condition of steel. Soft machine steel possesses a modulus similar to that of heat-treated alloy steel; therefore the latter with identical cross section will deflect to the same extent with similar loading This factor is of great importance in the design of transmission shafting to carry heavy loads in combination with belt or chain pull between bearings The deflection capacity of the shaft must be based on diameter and not on the use of steels having high tensile strength.

Table 14-33. Physical Properties of Cold-finished Steel Shafting, SAE Numbers

Type	Tensile strength, psi	Yield point, psi	Elongation in 2 in., per cent	Reduction of area, per cent	Size diameter in.
Hot-rolled soft steel or screwstock	50/60,000	30/35 000	20/30	35/45	1–2
Cold-finished screwstock, SAE 1112 or 1120	70/100,000	55/90,000	10/20	32/41	Up to 1
Cold-finished screwstock, SAE 1112 or 1120	50/60,000	30/40,000	20/35	35/45	2¾–4
Cold-finished, 35/45 carbon, SAE 1040	70/90,000	35/50,000	20/30	35/50	1½–2
Cold-finished annealed, 35/45 carbon alloy such as SAE 3140, 2340, 6140, etc	90/110,000	60/70,000	20/30	55/65	Up to 1½

Shaft Failures. Transmission shafting usually fails from abnormal deflection, torsional fatigue, or progressive fracture. The usual term applied is "crystallization." This term is a misnomer because all steel is crystalline, and the structure is not changed by breakage. Breakages result in a crystalline appearance of a portion of the fracture, while other portions of the fracture have been smoothed or rubbed down by abrasion or wear of the parted contacting surfaces prior to complete rupture.

Table 14-34. Physical Properties of Hot-rolled Steel Shafting, SAE Numbers

Type	Tensile strength, psi	Yield point, psi	Elongation in 2 in., per cent	Reduction of area, per cent	Brinell hardness	diameter, in.
Soft steel, SAE 1020	50/60,000	30/35,000	20/30	35/45	150/200	1–2
Medium carbon, SAE 1045	80/90,000	45/50,000	15/26	32/42	170/210	1–2
Medium carbon, high, manganese SAE X1340	90/110,000	55/60,000	20/30	35/45	163/207	1–2
Machinable (annealed) alloy, SAE 3145, 2345, 6145	80/100,000	50/60,000	25/35	30/40	180/220	1–2
Machinable heat-treated alloy, SAE 6150	140/150,000	120/130,000	14/20	52/60	302/340	1¾–3
Machinable heat-treated alloy, SAE 6150	145/160,000	120/135,000	15/20	42/52	302/340	3–6

Shafting failures can be minimized by the proper filleting of cross-section changes, keyways, etc., and by particular attention to torsional resistance and deflection capacity. These factors are more important as the strength of the steel employed increases.

Shaft-size Determination, Primary Factors

Service:

1. Source of power and kind of driven machines
2. Normal and maximum power to be transmitted
3. Steady or fluctuating load
4. Magnitude and duration of shock loads
5. Revolutions per minute and direction of rotation
6. Center distance and relative elevation of driver and driven

Kind of Drive:

1. Flat or V belt, rope, chain, gears, friction, or direct-connected by coupling or clutch to reduction gear or motor
2. Size and type of belt or ropes

Pulleys, Sprockets, or Gears:

1. Size and weight
2. Number of arms, length and location of hubs
3. Location on shaft in relation to bearings

Bearings:

1. Rigid or self-aligning, roller, ball, or sleeve type
2. Type of lubrication
3. Length of bearing surface on shaft
4. Method of supporting bearings

In heavy or important installations an approximate shaft weight should be included in first calculation and checked when shaft size is definitely determined. All stress factors acting upon the shaft should be considered, such as weight of shaft, belt, chain attachment, material carried, and any unusual erection or operating condition. Shaft sizes are generally large when speeds are low, when bending loads are heavy, or when bearings are not close to the bending loads. Smaller shafting can often be used with safety and economy by increased speeds and by proper mounting of the shaft in antifriction bearings. Shafts that are too small can be relieved of excessive stress by an additional bearing or by relocating bearings in relation to loads.

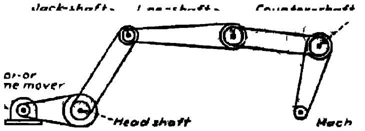

FIG. 14-10. Identification of group method of shafting.

Shafting for Group Method. Cold-finished shafting is usually employed. Power requirements are comparatively low, and shaft speeds in rpm are such that high stresses are not involved. In view of this, empirical formulas by Thurston have been arranged for quickly solving shafting problems of this character.

Identification of "Group-method" Shafting. Figure 14-10 indicates the name and location of the various shafts employed in the group system of driving by belting, chain, or a combination of these mediums.

Note. This arrangement is not necessarily standard for the group method, since in many cases the motor is directly belted or chain connected to the line shaft. The line shaft is then known as the "head line shaft." The jackshaft is used for ratio purposes and also to break up long center distances. Frequently the use of a countershaft is unnecessary, the machines being direct belted or chain connected from the line shaft.

Head Shafts. *Location.* First shaft from motor or prime mover.

Service. Reduces speed between motor and line or jackshaft and must have load capacity for entire installation.

Stresses. Torsional and bending.

Supports. Usually two bearings of the rigid type on adjustable base plates. The shaft is short, based on the usual provision for one power-receiving pulley, sprocket, or sheave from the motor and one power-delivery pulley or sprocket from the head shaft. Three bearings may be used, but for the average head shaft this is the maximum.

Speeds. Industrial range 25 to 550 rpm.

Formula:

$$\mathrm{Hp} = D^3R/125$$
$$D = \sqrt[3]{(125 \times \mathrm{Hp})/R}$$

where Hp = horsepower
D = shaft diameter
R = rpm
125 = constant for a combined torsional and bending safe-working stress of 2,800 psi

Note: Use constant of 125 only when head shaft is subjected to heavy strains, such as slow speeds and intermittent loads, or where clutches, shifters, or gearing are necessary. For the average head-shaft installation use 110 as the constant.

Line Shafts. *Location.* Can be first shaft from motor or first or second shaft from head shaft.

Service. Acts as a power-distributing shaft to the various pieces of apparatus or machines of the group.

Stresses. Torsional and bending.

Supports. Usually three or more bearings, spacing depending upon power pulls by belt or chain and the dead load of pulleys, gears, sprockets, sheaves, clutches, and couplings. Self-aligning adjustable bearings should be employed wherever possible. If the rigid type of bearing is necessary, provide adjustable base plates.

Speeds. Industrial range 70 to 400 rpm.

Formula. For line shafts 75 to 100 ft long, heavily loaded, and bearings spaced on 8-ft. centers

$$\mathrm{Hp} = D^3R/100$$
$$D = \sqrt[3]{(100 \times \mathrm{Hp})/R}$$

where Hp = horsepower
D = shaft diameter
R = rpm
100 = constant for a combined torsional and bending safe-working stress of 3,200 psi

Note: For line shafts 50 to 75 ft long, medium loaded, with bearings spaced on 8-ft centers, use constant of 90. For line shafts 20 to 50 ft long, lightly loaded, with bearings spaced on 6-ft centers, use constant of 75.

Jackshafts. *Location.* Either between head and line shafts, or between line and countershafts.

Service. For the increase or decrease of velocity ratio or to reduce center distance between principal shafts.

Stress. Torsional and bending.

Supports. If short, two or three bearings of either the rigid or the self-aligning adjustable type; if long, three or more bearings of the self-aligning adjustable type.

Speeds. Industrial range 100 to 400 rpm.

Formula. Same as for line shafting, with similar constants based on loading.

Countershafts. *Location.* Usually employed between line shaft and driven apparatus or machine.

Service. For increase or decrease of velocity ratio to machine or as a power-input control medium to driven machine.

Stresses. Torsional and bending.

Supports. Two or three bearings of either the rigid or the self-aligning adjustable type. When used as a component part of the driven machine, two bearings are usually employed.

Speeds. Industrial range, 100 to 600 rpm

Formula. For average service and loading

$$\mathrm{Hp} = D^3R/80$$

$$D = \sqrt[3]{(80 \times \mathrm{Hp})/R}$$

where Hp = horsepower
D = shaft diameter
R = rpm
80 = constant for combined torsional and bending safe-working stress of 3,600 psi

Specification Numbers for SAE Steels. A numeral index system is used for the specification of SAE steels, which facilitates the specifying of these steels on drawings and in the field. Such numerals are partially descriptive of the quality of material covered by these numbers. The first numeral of the number indicates the class to which the steel belongs; thus the numeral 1 indicates a carbon steel, 2 a nickel steel, and 3 a nickel-chromium steel. In the case of the alloy steels the second numeral of the number generally indicates the approximate percentage of the predominant alloying element. Usually the last two or three numerals of the number indicate the average carbon content in "points" or hundredths of 1 per cent. Thus 2340 indicates a nickel steel of approximately 3 per cent nickel (3.25 to 3.75) and 0.40 per cent carbon (0.35 to 0.45), and 71360 indicates a tungsten steel of about 13 per cent tungsten (12 to 15) and 0.60 per cent carbon (0.50 to 0.70). The basic numerals for the various qualities of steels specified are shown in Table 14-35.

Table 14-35

Steel	*Basic numeral*
Carbon	1
Nickel	2
Nickel-chromium	3
Molybdenum	4
Chromium	5
Chromium-vanadium	6
Tungsten	7
Silicomanganese	9

STEEL-CABLE TRANSMISSION BELTING

BY PAUL SULOFF

Mechanical Goods Design Staff, The Goodyear Tire & Rubber Company

Steel-cable transmission belting is assembled with a single or double helix of steel cable embedded in rubber or synthetic compound and encased in one or more envelopes, usually of cotton fabric, which give lateral strength to the cable helix and protect it from abrasion (see Fig. 14-11). The cable itself is ordinarily 0.035 to 0.050 in. in over-all diameter, with individual wires 0.006 to 0.008 in. in diameter. The ultimate strength of the wire is usually on the order of 300,000 to 350,000 psi, and the strength of the cable, 200 to 350 lb. Assembled into the belt, this results in belt strength on the order of 4,000 to 6,000 lb per in. of width, with final elongation of 1.5 to 2.0 per cent. The belt thickness may be from $\frac{3}{32}$ to $\frac{3}{8}$ in., depending on its width and the number and weight of envelopes required by the dictates of lateral strength and good proportion.

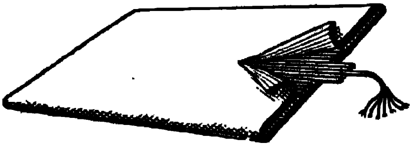

FIG. 14-11. Section of steel-cable transmission belting.

This type of belting has been designed to provide horsepower capacity per unit of width in excess of other types of belting. Because it can be kept extremely thin and free of internal bending stresses, its high ratings can be carried down into the small-pulley drives. Its extremely high elastic constant keeps the amount of take-up required to a minimum, and freedom from any hygroscopic effect on the length eliminates any shrinkage problems.

Generally, steel-cable belts are built endless by the manufacturer, but where necessary they can be vulcanized endless in the field. Standard metallic fasteners are not recommended since they will not hold at the ratings given this type of belt.

For horsepower ratings of steel-cable belting see Table 14-36.

Table 14-36. Maximum Horsepower Rating for Compass No. 250 Steel-cable Belts per Inch of Belt Width at 180 deg Arc of Contact to Be Applied against Peak Load

Small-pulley diam, in.	Belt speed, fpm																
	100	500	1,000	1,500	2,000	2,500	3,000	3,500	4,000	4,500	5,000	5,500	6,000	7,000	8,000	9,000	10,000
4	0.29	1.4	2.6	3.4	4.1	4.7	5.2	5.6	5.9	6.2	6.3	6.3	6.2	5.2	3.3		
5	0.35	1.6	2.9	4.0	4.8	5.7	6.5	7.2	7.8	8.4	8.8	9.1	9.0	8.2	6.6	3.8	
6	0.41	1.8	3.4	4.6	5.8	6.8	7.7	8.6	9.5	10.2	10.8	11.3	11.6	11.6	10.6	8.6	5.5
7	0.44	2.0	3.7	5.1	6.6	7.8	9.0	10.0	11.0	11.8	12.4	12.9	13.2	13.4	12.8	11.8	10.0
8	0.49	2.2	4.1	5.9	7.5	9.2	10.7	12.0	13.2	14.0	14.4	14.7	14.8	15.0	15.0	14.8	14.7
9	0.54	2.3	4.5	6.3	8.1	10.0	11.8	12.4	14.8	16.0	17.0	17.4	17.6	18.2	18.0	17.8	17.7
10	0.57	2.4	4.8	6.9	9.0	11.0	13.0	15.0	16.8	18.5	19.6	20.2	20.5	20.8	21.0	20.8	20.6
12	0.65	2.7	5.4	8.0	10.6	13.2	15.8	18.2	20.4	22.6	24.5	25.7	26.2	26.6	27.0	27.0	26.6
14	0.74	3.1	6.2	9.4	12.5	15.4	18.4	21.4	24.2	26.8	29.2	30.8	31.6	32.0	32.5	32.5	32.0
16 and over	0.86	3.6	7.2	10.8	14.3	17.8	21.2	24.5	27.8	30.7	33.5	35.7	36.6	37.1	37.5	37.6	37.2

These ratings reach values that are approximately twice the highest ratings given cotton-rubber cord belting.

Steel-cable belting requires no more and no less tension than other rubber-surfaced belting. It has the advantage that, because narrower belts can be used, the bending moment on the overhanging motor shaft is less, and bearing force is lower even though the belt tension is unchanged.

Crowns recommended for pulleys for steel-cable belts are given in Table 14-37.

Tests conducted on steel-cable belts for response to crown show that they respond to a given crown to much the same degree as cotton-rubber belts. In general, shaft misalignment in excess of 1 deg cannot be corrected for by crown.

The high elastic constant (low elongation) of the steel-cable belt, which is an advantage on ordinary open drives, prevents its use on most quarter-turn and twisted-belt drives, since the uneven distribution of strain across the belt width results in highly localized tensions destructive to the belt.

Table 14-37. Recommended Maximum Crown

(Difference between diameter at center and at edge)

Pulley-face width, in.	Large pulley, in.	Small pulley
2– 5	$\frac{1}{32}$	Flat–$\frac{1}{32}$ in.
6– 9	$\frac{3}{64}$	Flat–$\frac{1}{32}$ in.
10–15	$\frac{1}{16}$	Flat–$\frac{1}{32}$ in.
16–30	$\frac{3}{32}$	$\frac{1}{32}$ in.

While some extraordinary precautions must be taken to use successfully steel-cable belts, the high rating and length stability of this belt make it a power-transmission medium for many applications.

These same characteristics of high strength and rating coupled with length stability have been taken advantage of in designing steel-cable V belts.

The high elastic constant of steel-cable belting has also made possible the current development of positive-drive flat belting. In the past, efforts to design a positive-drive flat belt employing engaging teeth on belt and pulleys have been handicapped by the difficulty of maintaining pitch in the belt in the face of elastic and inelastic elongation. Steel cable has reduced these factors to an amount probably less than the pitch change resulting from wear in a chain.

Positive-drive steel-cable belts in widths from ¼ to 2 in. are being developed around the steel-cable flat-belt carcass to maintain a fixed ratio between shafts for loads up to 10 hp. These have teeth molded on the inner surface of the belt and engaging recesses machined in the pulleys. This type of belt offers a quiet lubrication-free method of maintaining a fixed speed relationship between shafts for light and moderate loads. At present, such drives are being custom designed for original equipment uses since expensive manufacturing equipment precludes the design of individual drives. Future design may include standard-pitch belts and pulleys for selection of drives for individual applications.

Steel-cable belting has been in successful operation for several years, and while other strong fibers are coming up to challenge the position of both steel and cotton, there is still nothing with the concentrated power capacity and length stability of steel.

MULTIPLE AND VARIABLE-SPEED V-BELT DRIVES

BY H. A. WILSON

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The V-belt drive consists of one or more endless belts of trapezoidal cross section, run on sheaves with V-shaped grooves on the periphery. The endless belts are constructed of cotton or rayon cords and fabric impregnated with rubber. V belts are also made with endless steel cables as tension members. Horsepower ratings for belts are given in Tables 14-39 to 14-43.

Table 14-38

Belt section	K	t	n	W	Kn	tn	Maximum sheave diam
A	0.213	0.020	22	0.07	4.686	0.440	5.0
B	0.225	0.034	37	0.12	8.325	1.258	7.0
C	0.241	0.061	72	0.19	17.350	4.393	12.0
D	0.250	0.106	149	0.39	37.250	15.790	17.0
E	0.250	0.160	227	0.59	56.750	36.320	28.0

V-belt drives are primarily short center drives and provide a smooth, even flow of power, with a minimum of slip and without noise. These drives can be operated in either direction with slack side of belts on top or bottom. They are especially adapted to reversing service.

OPERATION OF V-BELT DRIVES

Alignment of V-belt Drives. The driver and driven shafts should be parallel, and the belts should be at right angles to the shafts. Any appreciable misalignment may cause undue wear on the belts.

Adjustment of V-belt Drives for Proper Tension. Every installation of V-belt drives should include some form of movement to permit maintaining proper tension on the drive. This can be accomplished through the use of slide rails or a base on the motor or a pivoted motor base. An idler may be used where no other form of adjustment is possible. A grooved idler operating on the inside of the belts is to be preferred to a flat-face idler operation on the back of the belts. V belts (cotton or rayon) may stretch approximately 4 per cent of their original pitch length during their entire useful operating life. Provision should be made for this amount of take-up so that proper tension can be maintained at all times.

Replacement of V Belts. All tension should be removed from the drive and as much slack as possible obtained on the belts. The belts can then be removed easily and a complete new set placed on the drive. A complete new set of belts is recommended because of the difficulty of matching new belts with the old belts which have been used and have stretched an unknown amount. V belts should never be pried onto the sheave. A bar or any object used for this purpose may rupture the cord construction of the belt causing it to break.

Storage of Spare Sets of V Belts. Belts should be stored in a cool, dry place. Light, heat, and moisture will cause deterioration and greatly reduce the useful life of the belts.

Belt Dressing. No belt dressing of any kind should ever be used on any V-belt drive. The application of belt dressing will cause deterioration and shorten the useful life of the belts.

Table 14-39. Horsepower Ratings for A-section V Belts

Velocity, fpm	Sheave diam						
	2.6 in.	3.0 in.	3.4 in.	3.8 in.	4.2 in.	4.6 in.	5.0 in. and larger
1,000	0.5	0.7	0.8	0.9	0.9	1.0	1.0
1,100	0.6	0.7	0.9	1.0	1.0	1.1	1.1
1,200	0.6	0.8	1.0	1.0	1.1	1.2	1.2
1,300	0.7	0.9	1.0	1.1	1.2	1.3	1.3
1,400	0.7	0.9	1.1	1.2	1.3	1.4	1.4
1,500	0.8	1.0	1.1	1.3	1.4	1.4	1.5
1,600	0.8	1.0	1.2	1.3	1.4	1.5	1.6
1,700	0.9	1.1	1.3	1.4	1.5	1.6	1.7
1,800	0.9	1.2	1.4	1.5	1.6	1.7	1.8
1,900	0.9	1.2	1.4	1.6	1.7	1.8	1.9
2,000	1.0	1.3	1.5	1.6	1.8	1.8	2.0
2,100	1.0	1.3	1.5	1.7	1.9	2.0	2.1
2,200	1.1	1.4	1.6	1.8	1.9	2.0	2.2
2,300	1.1	1.4	1.6	1.8	2.0	2.1	2.3
2,400	1.1	1.4	1.7	1.9	2.0	2.2	2.3
2,500	1.1	1.5	1.7	2.0	2.1	2.3	2.4
2,600	1.2	1.6	1.8	2.0	2.2	2.3	2.5
2,700	1.2	1.6	1.8	2.1	2.3	2.4	2.6
2,800	1.2	1.6	1.9	2.1	2.3	2.5	2.6
2,900	1.2	1.6	1.9	2.2	2.4	2.6	2.7
3,000	1.2	1.7	2.0	2.2	2.4	2.6	2.7
3,100	1.3	1.7	2.0	2.3	2.5	2.6	2.8
3,200	1.3	1.7	2.0	2.3	2.5	2.7	2.9
3,300	1.3	1.7	2.1	2.4	2.6	2.7	2.9
3,400	1.3	1.8	2.1	2.4	2.7	2.8	3.0
3,500	1.3	1.8	2.1	2.5	2.7	2.8	3.0
3,600	1.3	1.8	2.2	2.5	2.7	2.9	3.1
3,700	1.3	1.8	2.2	2.5	2.8	2.9	3.1
3,800	1.3	1.8	2.2	2.5	2.8	3.0	3.2
3,900	1.3	1.8	2.2	2.5	2.8	3.0	3.2
4,000	1.2	1.8	2.2	2.6	2.8	3.0	3.3
4,100	1.2	1.8	2.2	2.6	2.9	3.1	3.3
4,200	1.2	1.8	2.2	2.6	2.9	3.1	3.3
4,300	1.2	1.8	2.2	2.6	2.9	3.1	3.3
4,400	1.2	1.8	2.2	2.6	2.9	3.2	3.4
4,500	1.1	1.7	2.2	2.6	2.9	3.2	3.4
4,600	1.1	1.7	2.2	2.6	2.9	3.2	3.4
4,700	1.0	1.7	2.2	2.6	2.9	3.2	3.4
4,800	1.0	1.7	2.2	2.6	2.9	3.2	3.4
4,900	0.9	1.6	2.2	2.6	2.9	3.2	3.4
5,000	0.9	1.6	2.1	2.6	2.9	3.2	3.4

SELECTING A V-BELT DRIVE

Selection of a V-belt drive for any application should be based on the nature of the load, the type of driving unit, horsepower (name-plate rating) and speed. To obtain the horsepower capacity to be used, multiply the rated horsepower of the driving unit by the recommended service factor for the type of driven unit (see Table 14-44). The belt section to be used for a drive is dependent on the rating and speed (see Table 14-46). Belt velocities should be under 5,000 fpm for maximum belt life. Sheave diameters used should not be less than recommended minimum diameters as shown in Table 14-47. Corrections for arc of contact on small sheave are determined by

Table 14-40. Horsepower Ratings for B-section V Belts

Velocity, fpm	Sheave diam					
	5.0 in.	5.4 in.	5.8 in.	6.2 in.	6.6 in.	7.0 in. and larger
1,000	1.3	1.4	1.5	1.6	1.7	1.8
1,100	1.4	1.5	1.6	1.7	1.8	1.9
1,200	1.5	1.6	1.7	1.8	1.9	2.0
1,300	1.6	1.8	1.9	2.0	2.1	2.2
1,400	1.7	1.9	2.0	2.1	2.2	2.3
1,500	1.8	2.0	2.1	2.3	2.4	2.5
1,600	1.9	2.1	2.3	2.4	2.5	2.6
1,700	2.0	2.2	2.4	2.5	2.7	2.8
1,800	2 2	2.4	2.5	2.7	2.8	2.9
1,900	2.3	2.5	2.6	2.8	3.0	3.1
2,000	2.4	2.6	2.8	2.9	3.1	3.2
2,100	2.4	2.7	2.9	3.1	3.2	3.4
2,200	2.5	2.8	3.0	3.2	3.4	3.5
2,300	2.6	2.9	3.1	3.3	3.5	3.6
2,400	2.7	3.0	3.2	3.4	3.6	3.7
2,500	2.8	3.1	3.3	3.5	3.7	3.9
2,600	2.9	3.2	3.4	3.6	3 8	4.0
2,700	3.0	3.3	3.5	3.7	3.9	4.1
2,800	3.0	3.3	3.6	3.8	4.1	4.3
2,900	3.1	3.4	3.7	3.9	4.2	4.4
3,000	3.2	3.5	3.8	4.0	4.3	4.5
3,100	3.2	3.6	3.9	4.1	4.4	4.6
3,200	3.3	3.6	4.0	4.2	4.5	4.7
3,300	3.3	3.7	4.0	4.3	4.6	4.8
3,400	3.4	3.7	4.1	4.4	4.7	4.9
3,500	3.4	3.8	4.1	4.4	4.7	4.9
3,600	3.4	3.8	4.2	4.5	4.8	5.0
3,700	3.5	3.9	4.3	4.5	4.8	5.1
3,800	3.5	3.9	4.3	4.6	4.9	5.2
3,900	3.5	3.9	4.3	4.6	4.9	5.2
4,000	3.5	4.0	4.4	4.7	5.0	5.3
4,100	3.5	4.0	4.4	4.7	5.0	5.3
4,200	3.5	4.0	4.4	4.7	5 1	5.4
4,300	3.5	4.0	4.4	4.7	5.1	5.4
4,400	3.5	4.0	4.4	4.8	5.1	5.4
4,500	3.5	4.0	4.4	4.8	5.1	5.4
4,600	3.4	4.0	4.4	4.8	5.1	5.5
4,700	3.4	3.9	4.4	4.8	5.2	5.5
4,800	3.4	3.9	4.4	4.8	5.2	5.5
4,900	3 3	3.9	4.3	4 8	5 1	5.5
5,000	3 3	3.8	4.3	4.7	5.1	5.5

formula

$$\text{Arc of contact} = 180 - [(D - d)60]/C$$

where D = diameter of large sheave
d = diameter of small sheave
C = center distance of drive

See Table 14-48 for correction factors.

Center Distance. Knowing the sheave sizes required for a drive and the approximate center distance, the belt length can be estimated by the use of the formula

$$L = 2C + (\pi/2)(D + d)$$

Select the nearest belt length from Table 14-45 (page 1003).

Table 14-41. Horsepower Ratings for C-section V-Belts

Velocity, fpm	Sheave diam					
	7.0 in.	8.0 in.	9.0 in.	10.0 in.	11.0 in.	12.0 in. and larger
1,000	2.0	2.5	2.8	3.1	3.4	3.6
1,100	2.2	2.7	3.1	3.5	3.7	3.9
1,200	2.4	2.9	3.4	3.8	4.0	4.3
1,300	2.6	3.2	3.7	4.1	4.4	4.6
1,400	2.7	3.4	3.9	4.4	4.7	5.0
1,500	2.9	3.6	4.2	4.6	5.0	5.3
1,600	3.1	3.9	4.5	4.9	5.3	5.7
1,700	3.3	4.1	4.7	5.3	5.6	6.0
1,800	3.4	4.3	5.0	5.5	5.9	6.3
1,900	3.6	4.5	5.2	5.8	6.2	6.6
2,000	3.7	4.7	5.5	6.1	6.5	7.0
2,100	3.9	4.9	5.7	6.3	6.8	7.3
2,200	4.0	5.1	5.9	6.6	7.1	7.6
2,300	4.2	5.3	6.2	6.8	7.4	7.9
2,400	4.3	5.5	6.4	7.1	7.7	8.2
2,500	4.5	5.7	6.6	7.3	8.0	8.5
2,600	4.6	5.8	6.8	7.6	8.2	8.8
2,700	4.7	6.0	7.0	7.8	8.4	9.0
2,800	4.8	6.2	7.2	8.0	8.7	9.3
2,900	4.9	6.3	7.4	8.3	9.0	9.6
3,000	5.0	6.5	7.6	8.5	9.2	9.8
3,100	5.1	6.6	7.8	8.7	9.5	10.1
3,200	5.2	6.7	7.9	8.9	9.7	10.3
3,300	5.3	6.9	8.1	9.1	9.9	10.6
3,400	5.4	7.0	8.3	9.3	10.1	10.8
3,500	5.4	7.1	8.4	9.4	10.3	11.0
3,600	5.5	7.2	8.5	9.6	10.5	11.2
3,700	5.5	7.3	8.7	9.8	10.7	11.4
3,800	5.6	7.4	8.8	9.9	10.9	11.6
3,900	5.6	7.5	8.9	10.1	11.0	11.8
4,000	5.6	7.5	9.0	10.2	11.2	12.0
4,100	5.6	7.6	9.1	10.3	11.3	12.2
4,200	5.6	7.6	9.2	10.4	11.5	12.3
4,300	5.6	7.7	9.3	10.5	11.6	12.5
4,400	5.6	7.7	9.3	10.6	11.7	12.6
4,500	5.5	7.7	9.4	10.7	11.8	12.7
4,600	5.5	7.7	9.4	10.8	11.9	12.9
4,700	5.4	7.7	9.5	10.8	12.0	13.0
4,800	5.4	7.7	9.5	10.9	12.1	13.2
4,900	5.3	7.7	9.5	11.0	12.1	13.2
5,000	5.2	7.6	9.5	11.0	12.2	13.2

The exact center distance can be calculated from formula

$$C = \frac{b + \sqrt{b^2 - 32(D - d)^2}}{16}$$

where $b = 4L - 6.28(D + d)$

L = pitch length of belt

For small ratios and long centers, reasonably accurate centers can be found by the formula

$$C = [L - 1.57(D + d)]/2$$

Table 14-42. Horsepower Ratings for D-section V Belts

Velocity, fpm	Sheave diam					
	12.0 in.	13.0 in.	14.0 in.	15.0 in.	16.0 in.	17.0 in. and larger
1,000	4.5	5.1	5.6	6.1	6.5	6.8
1,100	4.9	5.8	6.2	6.7	7.2	7.5
1,200	5.3	6.1	6.7	7.3	7.7	8.2
1,300	5.7	6.5	7.2	7.8	8.4	8.8
1,400	6.1	7.0	7.8	8.4	9.0	9.4
1,500	6.5	7.5	8.2	9.0	9.6	10.1
1,600	6.9	7.9	8.8	9.5	10.2	10.7
1,700	7.3	8.4	9.3	10.1	10.8	11.4
1,800	7.7	8.8	9.8	10.6	11.3	12.0
1,900	8.0	9.2	10.2	11.1	11.9	12.6
2,000	8.4	9.7	10.7	11.7	12.5	13.2
2,100	8.8	10.1	11.2	12.2	13.0	14.4
2,200	9.1	10.5	11.7	12.7	13.6	14.9
2,300	9.5	10.9	12.1	13.2	14.1	14.9
2,400	9.8	11.3	12.6	13.7	14.6	15.5
2,500	10.1	11.6	12.9	14.1	15.1	16.0
2,600	10.4	12.0	13.4	14.6	15.6	16.6
2,700	10.6	12.3	13.8	15.0	16.1	17.1
2,800	10.9	12.7	14.1	15.5	16.6	17.6
2,900	11.2	13.0	14.5	15.9	17.0	18.1
3,000	11.4	13.3	14.9	16.3	17.5	18.6
3,100	11.7	13.6	15.2	16.7	17.9	19.0
3,200	11.9	13.9	15.6	17.1	18.4	19.5
3,300	12.1	14.1	15.9	17.4	18.9	19.9
3,400	12.2	14.4	16.2	17.7	19.1	20.3
3,500	12.4	14.6	16.5	18.1	19.5	20.7
3,600	12.6	14.8	16.7	18.4	19.8	21.1
3,700	12.7	15.0	17.0	18.7	20.2	21.5
3,800	12.8	15.2	17.2	18.9	20.4	22.2
3,900	12.9	15.3	17.4	19.2	20.7	22.3
4,000	13.0	15.5	17.6	19.4	21.0	22.5
4,100	13.0	15.6	17.8	19.6	21.3	22.8
4,200	13.1	15.7	17.9	19.8	21.5	23.0
4,300	13.1	15.7	18.0	20.0	21.8	23.0
4,400	13.1	15.8	18.1	20.1	21.9	23.3
4,500	13.0	15.8	18.2	20.3	22.1	23.5
4,600	13.0	15.8	18.3	20.4	22.3	23.9
4,700	12.9	15.8	18.3	20.5	22.4	24.1
4,800	12.8	15.8	18.4	20.6	22.5	24.2
4,900	12.7	15.7	18.3	20.6	22.6	24.3
5,000	12.5	15.7	18.3	20.6	22.6	24.4

The pitch length of the belt given in Table 14-45 is used in all calculations.

Example. A fan operating continuously at 420 rpm is to be driven by a 10-hp 1,750-rpm motor. Center distance between fan and motor shaft is approximately thirty-two inches

Solution: From Table 14-44 the overload factor for fans is 1.4 to 2.0 depending on the type of motor used. With a squirrel-cage motor the service factor will be 1.6 (heavy-duty fan) Horsepower of drive, therefore, is 1.6 × 10 = 16 hp Speed ratio 1,750/420 = 4.16 From Table 14-46 a B belt is recommended. Since B belts will be used, the minimum-size sheave recommended is 5.4 in. (see Table 14-47). The driven sheave required would be 22.46 which is not a stock size (See Table 14-58). Therefore, the sheaves which should be selected would be 6 and 25 in. which are stock sizes. Next determine the length of belt which will give nearest the center distance required. This length can be estimated by

Table 14-43. Horsepower Ratings for E-section V Belts

Velocity, fpm	Sheave diam								
	20.0 in.	21.0 in.	22.0 in.	23.0 in.	24.0 in.	25.0 in.	26.0 in.	27.0 in.	28.0 in. and larger
1,000	8 0	8 5	9.0	9.4	9.8	10 2	10.5	10.9	11.1
1,100	8.7	9.3	9.8	10.3	10.8	11.2	11.6	11.9	12.2
1,200	9.5	10.1	10.7	11.2	11.7	12.2	12.6	13.0	13.3
1,300	10.2	10.9	11.6	12.1	12.7	13.1	13.6	14.0	14.4
1,400	11.0	11.7	12.4	13.0	13.6	14.1	14.6	15.0	15.4
1,500	11.7	12.5	13.2	13.9	14.5	15.1	15.6	16.1	16.5
1,600	12.4	13.3	14.1	14.8	15.4	16.0	16.6	17.1	17.5
1,700	13.1	14.0	14.9	15.6	16.3	16.9	17.5	18.1	18.6
1,800	13.8	14.8	15.7	16.5	17.2	17.9	18.5	19.0	19.6
1,900	14.5	15.5	16.4	17.3	18.0	18.8	19.4	20.0	20.6
2,000	15.2	16.2	17.2	18.1	18.9	19.6	20.3	21.0	21.6
2,100	15.8	17.0	18.0	18.9	19.7	20.5	21.2	21.9	22.5
2,200	16.5	17.6	18.7	19.7	20.6	21.4	22.1	22.8	23.5
2,300	17.1	18.3	19.4	20.4	21.4	22.2	23.0	23.7	24.4
2,400	17.7	18.9	20.1	21.2	22.2	23.1	23.9	24.6	25.3
2,500	18.3	19.6	20.8	21.9	23.0	23.9	24.7	25.5	26.3
2,600	18.9	20.2	21.5	22.6	23.7	24.7	25.5	26.4	27.1
2,700	19.4	20.8	22.1	23.3	24.4	25.4	26.4	27.2	28.0
2,800	19.9	21.4	22.8	24 0	25.1	26 2	27.1	28.0	29.9
2,900	20.4	22.0	23.4	24.7	25.8	26.9	27.9	28.8	29.7
3,000	20.9	22 7	24.0	25.3	26.5	27.6	28.6	29.6	30.5
3,100	21 4	23.0	24 5	26 0	27.2	28 3	29.4	30.4	31.3
3,200	21.8	23.5	25.1	26.5	27.8	29.0	30.1	31.1	32.0
3,300	22.3	24.0	25.6	27.1	28.4	29.6	30.7	31.8	32.8
3,400	22.6	24.5	26.1	27.6	29.0	30.2	31.4	32.5	33.5
3,500	23.0	24.9	26.6	28.1	29.5	30.8	32.0	33.1	34.2
3,600	23.4	25.3	27.0	28.6	30.0	31.4	32.6	33.8	34.8
3,700	23.7	25.6	27.4	29 1	30.6	31.9	33.2	34.4	35.5
3,800	24.0	26.0	27.8	29.5	31.0	32.4	33.7	34.9	36.1
3,900	24.2	26.3	28.2	29.9	31.5	32.9	34.3	35.5	36.6
4,000	24.5	26.6	28.5	30.3	31.9	33.4	34.7	36.0	37.2
4,100	24.7	28.8	28.8	30.6	32.3	33.8	35.2	36.5	37.7
4,200	24.8	27.1	29.1	30.9	32.6	34.2	35.6	37.0	38.2
4,300	25.0	27.3	29.3	31.2	33 0	34.6	36.0	37.4	38.7
4,400	25.1	27.4	29.5	31.5	33.3	34.9	36.4	37.8	39.1
4,500	25.2	27.5	29.7	31.7	33.5	35.2	36.7	38.2	39.5
4,600	25.2	27.6	29.9	31.9	33.7	35.5	37.0	38.5	39.9
4,700	25.2	27.7	30.0	32.0	33.9	35.7	37.3	38.8	40.2
4,800	25.2	27.7	30.0	32.2	34.1	35.9	37.5	39.1	40.5
4,900	25.1	27.7	30.1	32.2	34 2	36.0	37.7	39.3	40.7
5,000	25.0	27.7	30 1	32 3	34.3	36.2	37.9	39.5	40.9

$L = 2C + (\pi/2)(D + d)$. In this case $L = 2(32) + (3.14/2)(25 + 6) = 112.8$. Therefore, a B112 belt should be used. Using the formula for center distance

$$C = \frac{b + \sqrt{b^2 - 32(D - d)^2}}{16}$$

the center distance will be found to be 31.2 in. A 6-in. sheave operating at 1,750 rpm will have a belt velocity of 2,745 fpm. From Table 14-40 the horsepower per belt will be found to be 3.6. Correcting horsepower for arc of contact on small sheave

$$\text{Arc of contact} = 180 - [(D - d)60]/C$$

From Table 14-48 correction is 0.90; 90 × 36 = 3.24. Finally (16/3.24) = 4.94. Therefore five belts will be required, and the drive will be 5-B112-60-250.

Table 14-44. Service Factors

Applications	Electric motors										Engines		
	Alternating current								Direct current		Gas and diesel		
	Squirrel cage				Syn-chro-nous		Single phase						
	Normal torque line start	Normal torque compensator start	High torque	Wound rotor (slip-ring)	Normal torque	High torque	Repulsion and split phase	Capacitor	Shunt wound	Compound wound	4 or more cylinders above 700 rpm	Steam	Line shafts and clutch starting
Agitators—paddle-propeller	1.2	1.0	1.4	1.2									
Brick and clay machinery	1.5	1.3	1.8	1.5	...	...	...	...	1.4	...	...	...	2.0
Bakery machinery	1.2	...	...	...	...	...	1.2	1.0					
Compressors	1.4	1.4	...	1.5	1.5	...	1.2	1.2	1.2	...	1.2		
Conveyors	...	1.6	1.8	...	...	...	...	...	1.6	...	...	...	1.8
Crushing machinery	...	1.4	1 6	1.4	1.4	1.6	...	...	...	1.6	1.6	...	1.6
Fans and blowers	1.6	1.6	2.0	2.0	2.0	2.0	...	...	1.4	...	1.6	1.5	1.5
Flour, feed, and cereal mill machinery	1.4	1.4	1.6	1.4	1.4	...	...	...	...	...	1.8		
Generators and exciters	1.2	...	...	...	...	...	...	...	1.2	...	2.0	1.4	1.4
Laundry machinery	1.2	...	...	...	...	...	...	...	...	1.2			
Line shafts	1.4	1.4	...	1 4	1.4	2.0	1.4	1.4	1.4	1.4	1.6	1.6	1.6
Machine tools	1.2	...	...	1.4	...	...	1.2	1.2	1.2	1.2			
Mills	...	1.6	1.6	1.4	...	...	...	...	...	1.4	...	...	1.6
Oil-field machinery	1.2	1.2	1.4	...	...	...	...	...	1.4	1.4	1.4	1.4	1.6
Paper machinery	1.2	1.4	1.8	1.5	1.6	1.8	...	...	1.5	1.5	...	...	1 8
Printing machinery	1.2	1.2	..	1.2	...	...	...	...	1.2	1.2			
Pumps	1.4	1.4	1.4	1 6	1.6	1.8	1.2	1,2	1.2	...	2.0	2.0	
Rubber-plant machinery	1 4	1.4	1.4	1.4	...	1.8	...	...	...	...	...	2.0	
Screens	1 2	1.2	1.4										
Textile machinery	1.6	...	1.8										

Horsepower Transmitted by V Belts. The following formula was developed for horsepower transmitted by V belts by Robert F. Vogt, former Chief Consulting Engineer for Allis-Chalmers Manufacturing Company

$$\text{Hp/belt} = \left[KT_0n - \frac{4Etn}{3N_id} - \frac{W}{g}\left(\frac{V}{60}\right)^2 \right] \frac{(T_1 - T_2)}{(T_1 - T_c)} \frac{V}{33{,}000}$$

where Hp/belt = horsepower recommended per belt
K = load safety factor
T_0 = breaking strength per cord
N = number of cords in belt
E = modulus of elasticity of cords
t = outer cord distance from neutral axis of cord section
N_i = number of cords per square inch
d = diameter of small sheave
W = weight of belt per foot of length
g = gravity constant
V = velocity of belt, fpm

T_1 = tension on tight side
T_2 = tension on slack side
T_c = tension due to centrifugal force
KT_0N = maximum allowable tension
$4Etn = T_b$ = bending-stress term
$\frac{W}{g}\left(\frac{V}{60}\right)^2 = T_c$ = centrifugal-force term

$$\frac{E}{N_t} = 190$$

$$T_0 = 14 \text{ lb}$$

$$\frac{(T_1 - T_2)}{(T_1 - T_c)} = \frac{R - 1}{R} = 8$$

where R = tension ratio = 5

The general horsepower formula can be converted to a more simplified form by using the constants given in Table 14-38 (page 996). The simplified formula is

$$\text{Hp/belt} = [0.339Kn - 6.141tn - 0.209WS^2]S$$

where $S = \frac{V}{1{,}000}$

Table 14-45. Standard V Belts

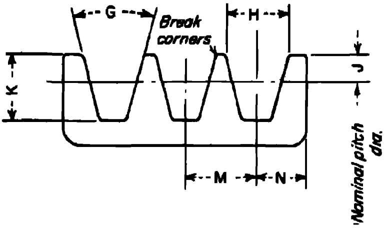
Section A, $\frac{1}{2} \times \frac{11}{32}$ in.		Section B, $\frac{21}{32} \times \frac{7}{16}$ in.		Section C, $\frac{7}{8} \times \frac{17}{32}$ in.		Section D, $1\frac{1}{4} \times \frac{3}{4}$ in.		Section E, $1\frac{1}{2} \times 1$ in.	
Serial No	Pitch length	Serial No.	Pitch length	Serial No	Pitch length	Serial No.	Pitch length	Serial No.	Pitch length
A26	26.9	B35	36.1	C51	52.6	D120	122.4	E180	183.3
A31	31.9	B38	39.1	C60	61.4	D128	130.4	E195	198.5
A35	35.9	B42	43.1	C68	69.4	D144	146.4	E210	213.6
A38	38.9	B46	47.1	C75	76.6	D158	160.4	E240	240.0
A42	42.9	B51	52.1	C81	82.6	D162	164.4	E270	270.0
A46	46.9	B55	56.4	C85	86.2	D173	175.4	E300	300.0
A51	51.9	B60	61.0	C90	91.9	D180	182.6	E330	330.0
A60	60.9	B68	69.3	C96	97.9	D195	197.6	E360	360.0
A68	68.9	B75	76.3	C105	106.9	D210	213.1		
A75	75.9	B81	82.3	C112	113.9	D240	240.0		
A80	81.1	B85	86.5	C120	122.0	D270	270.0		
A85	86.1	B90	91.5	C128	130.0	D300	300.0		
A90	91.1	B97	99.0	C144	146.0	D330	330.0		
A96	97.1	B105	106.5	C158	160.0	D360	360.0		
A105	106.1	B112	114.0	C162	164.0				
A112	113.1	B120	121.5	C173	175.0				
A120	121.1	B128	129.5	C180	182.0				
A128	129.1	B144	145.5	C195	197.0				
		B158	159.5	C210	212.0				
		B173	174.5	C240	240.0				
		B180	181.6	C270	270.0				
		B195	196.3	C300	300.0				
		B210	211.3	C330	330.0				
		B240	240.0	C360	360.0				
		B270	270.0						
		B300	300.0						

Table 14-46. General Belt Selection

Motor horsepower	Motor speed, rpm				
	1800	1200	900	720	600 and below
½–¾	A	A	A		
1–1½	A	A	A		
2–3	A or B	A or B	A or B	A or B	
5	A or B	A or B	A or B	B	
7½	A or B	A or B	B	B or C	B or C
10	A or B	B	B	B or C	C
15	B	B or C	B or C	C	C
20	B or C	C	C	C	C
25	C	C	C	C	C
30	C	C	C	C	C
40	C	C or D	C or D	C or D	D
50	C	C or D	C or D	D	D
60	C	C or D	C or D	D	D
75	C	C or D	C or D	D or E	D or E
100	C	D	D	D or E	D or E
125		D	D	D or E	D or E
150		D	D	D or E	E
200		D	D	E	E
250 and over				E	E

When two cross sections are shown:
If sheave pitch diameters are limited, use the smaller cross section.
If sheave face widths are limited, use the larger cross section.
If no limitations, use cross section providing the most economic drive.

Table 14-47. Groove Dimensions



Belt section	Minimum recommended pitch diam	Nominal pitch diam	Dimensions					
			G	H	J	K	M	N
A	3.00	2.6 to 5.4 over 5.4	34° 38°	0.494 0.504	0.125	0.490	⅝	⅜
B	5.40	4.6 to 7.0 over 7.0	34° 38°	0.637 0.650	0.175	0.580	¾	½
C	9.00	7.0 to 7.99 8.0 to 12.0 over 12.0	34° 36° 38°	0.879 0.887 0.895	0.200	0.780	1	$1\frac{1}{16}$
D	13.00	12.0 to 12.99 13.0 to 17.0 over 17.0	34° 36° 38°	1.259 1.271 1.283	0.300	1.050	$1\frac{7}{16}$	⅞
E	21.0	18.0 to 24.0 over 24.0	36° 38°	1.527 1.542	0.400	1.300	1¾	1⅛

Table 14-48. Correction Factors Corresponding to the Various Degrees of Contact on the Small Sheave

Arc of contact	*Correction factor*
180	1 00
170	0.98
160	0 95
150	0 92
140	0 89
130	0 86
120	0 83
110	0 79
100	0 74
90	0 69

Table 14-49. Difference between Pitch Diameter and Outside Diameter of Sheaves

Section	Nominal size of belt	Add to *PD* to get *OD*, in.
A	½ × 11/32 in.	250
B	21/32 × 7/16 in.	350
C	⅞ × 17/32 in.	.400
D	1¼ × ¾ in.	600
E	1½ × 1 in.	800

Table 14-50. Face Width of Stock and Standard Sheaves, In.

Size of belt	No. of grooves																								
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
A	¾	1⅜	2	2⅝	3¼	3⅞	4½	5⅛	5¾	6⅜	7	7⅝	8¼	8⅞	9½	10⅛	10¾	11⅜	12	12⅝	13¼	13⅞	14½	15⅛	15¾
B	1	1¾	2½	3¼	4	4¾	5½	6¼	7	7¾	8½	9¼	10	10¾	11½	12¼	13	13¾	14½	15¼	16	16¾	17½	18¼	19
C	1⅜	2⅜	3⅜	4⅜	5⅜	6⅜	7⅜	8⅜	9⅜	10⅜	11⅜	12⅜	13⅜	14⅜	15⅜	16⅜	17⅜	18⅜	19⅜	20⅜	21⅜	22⅜	23⅜	24⅜	25⅜
D	1¾	3 3/16	4⅝	6 1/16	7½	8 15/16	10⅜	11 13/16	13¼	14 11/16	16⅛	17 9/16	19	20 7/16	21⅞	23 5/16	24¾	26 3/16	27⅝	29 1/16	30½	31 15/16	33⅜	34 13/16	36¼
E	2¼	4	5¾	7½	9¼	11	12¾	14½	16¼	18	19¾	21½	23¼	25	26¾	28½	30¼	32	33¾	35½	37¼	39	40¾	42½	44¼

VARIABLE-SPEED V-BELT DRIVES

Variable speeds for V-belt drives can be obtained by the use of **variable-pitch-diameter sheaves.** Vari-Pitch variable-pitch-diameter sheaves for A-, B-, C-, D-, E-, Q-, and R-section belts are made. Two types of Vari-Pitch sheaves are available: **stationary control,** in which the adjustment for speed must be made with the drive stopped, and **motion control,** in which the adjustment for speed can be made with the drive running.

Table 14-51. Wide-range Vari-Pitch Texrope Drives

Horsepower ratings of wide-range Texrope belts

Q section							R section								
Velocity, fpm	Pitch diam, in.						Velocity, fpm	Pitch diam, in.							
	3.0	4.0	5.0	6.0	7.0	8.0		5.0	6.0	7.0	8.0	9.0	10.0	11.0	12.0
1,000	1.3	1.7	2.1	2.4	2.6	2.7	1,000	2.6	3.0	3.3	3.7	3.9	4.1	4.2	4.3
1,100	1.4	1.9	2.3	2.6	2.9	3.0	1,100	2.9	3.2	3.7	4.1	4.3	4.5	4.6	4.7
1,200	1.5	2.1	2.5	2.9	3.1	3.3	1,200	3.2	3.5	4.0	4.4	4.6	4.8	5.0	5.1
1,300	1.6	2.2	2.7	3.1	3.3	3.5	1,300	3.4	3.8	4.3	4.8	5.0	5.2	5.4	5.5
1,400	1.7	2.4	2.9	3.3	3.6	3.8	1,400	3.6	4.0	4.6	5.1	5.3	5.6	5.8	5.9
1,500	1.8	2.5	3.1	3.6	3.8	4.0	1,500	3.8	4.3	4.9	5.4	5.7	6.0	6.2	6.3
1,600	1.9	2.7	3.3	3.8	4.0	4.2	1,600	4.0	4.5	5.1	5.7	6.0	6.3	6.6	6.7
1,700	2.0	2.8	3.4	4.0	4.2	4.5	1,700	4.2	4.7	5.4	6.0	6.4	6.7	6.9	7.0
1,800	2.0	2.9	3.6	4.2	4.4	4.7	1,800	4.4	5.0	5.6	6.3	6.7	7.0	7.2	7.4
1,900	2.1	3.0	3.8	4.4	4.7	4.9	1,900	4.6	5.2	5.9	6.5	7.0	7.3	7.6	7.7
2,000	2.2	3.1	3.9	4.6	4.9	5.1	2,000	4.8	5.4	6.1	6.8	7.3	7.6	7.9	8.0
2,100	...	3.3	4.1	4.8	5.1	5.3	2,100	5.0	5.6	6.3	7.0	7.6	7.9	8.2	8.4
2,200	...	3.4	4.2	4.9	5.2	5.5	2,200	5.1	5.8	6.5	7.2	7.8	8.2	8.5	8.7
2,300	...	3.5	4.4	5.1	5.4	5.7	2,300	5.2	6.0	6.7	7.5	8.1	8.5	8.8	9.0
2,400	...	3.6	4.5	5.2	5.6	5.9	2,400	5.4	6.1	6.9	7.7	8.3	8.8	9.0	9.3
2,500	...	3.6	4.6	5.4	5.8	6.0	2,500	5.5	6.2	7.1	7.9	8.6	9.0	9.3	9.5
2,600	...	3.7	4.7	5.5	5.9	6.2	2,600	5.6	6.4	7.3	8.1	8.8	9.3	9.5	9.8
2,700	...	3.8	4.9	5.7	6.1	6.4	2,700	5.6	6.5	7.4	8.3	9.0	9.5	9.8	10.0
2,800	...	...	5.0	5.8	6.3	6.5	2,800	5.7	6.6	7.6	8.5	9.2	9.8	10.0	10.3
2,900	...	...	5.1	6.0	6.4	6.7	2,900	5.7	6.7	7.7	8.6	9.4	10.0	10.2	10.5
3,000	...	...	5.1	6.1	6.5	6.8	3,000	5.8	6.8	7.8	8.8	9.6	10.2	10.4	10.6
3,100	...	...	5.2	6.2	6.7	7.0	3,100	...	6.9	7.9	9.0	9.8	10.4	10.6	10.9
3,200	...	...	5.3	6.3	6.8	7.1	3,200	...	6.9	8.0	9.1	9.9	10.5	10.8	11.0
3,300	...	...	5.4	6.4	6.9	7.2	3,300	...	7.0	8.0	9.2	10.0	10.7	11.0	11.2
3,400	...	...	...	6.5	7.0	7.4	3,400	..	7.0	8.1	9.2	10.1	10.8	11.1	11.4
3,500	...	...	...	6.5	7.1	7.5	3,500	...	...	8.1	9.3	10.2	10.9	11.2	11.5
3,600	...	...	...	6.6	7.2	7.6	3,600	...	...	8.1	9.3	10.3	11.0	11.3	11.6
3,700	...	...	...	6.6	7.3	7.7	3,700	...	...	8.1	9.3	10.3	11.0	11.4	11.6
3,800	...	...	...	6.6	7.3	7.7	3,800	...	...	8.1	9.3	10.3	11.0	11.5	11.7
3,900	...	...	...	6.7	7.4	7.8	3,900	...	...	...	9.3	10.3	11.1	11.5	11.7
4,000	...	...	...	6.8	7.4	7.9	4,000	...	...	...	9.2	10.3	11.1	11.5	11.8
4,100	...	...	...	...	7.4	7.9	4,100	...	...	...	9.2	10.3	11.0	11.5	11.8
4,200	...	...	...	...	7.4	7.9	4,200	...	...	...	9.1	10.2	11.0	11.5	11.7
4,300	...	...	...	...	7.4	7.9	4,300	...	...	...	...	10.1	10.9	11.4	11.7
4,400	...	...	...	...	7.4	7.9	4,400	...	...	...	...	10.0	10.8	11.4	11.6
4,500	...	...	.	...	7.4	7.9	4,500	...	...	...	...	9.9	10.7	11.3	11.5
4,600	...	...	...	...	7.3	7.8	4,600	...	...	...	...	9.8	10.6	11.2	11.4
4,700	...	...	...	...	7.3	7.7	4,700	...	...	...	...	....	10.4	11.0	11.3
4,800	...	...	..	...	...	7.7	4,800	...	...	...	...	....	10.3	10.9	11.2
4,900	...	...	...	...	...	7.6	4,900	...	...	...	...	....	10.0	10.7	11.0
5,000	...	...	...	...	...	7.5	5,000	...	...	...	...	....	9.8	10.4	10.8

Table 14-52. Stock Wide-range Texrope Belts

Q section (1³⁄₁₆ × ½ in.)			R section (1½ × ¾ in.)		
Serial No.	Pitch length, in.	Approx. weight, lb	Serial No.	Pitch length, in.	Approx. weight, lb
Q-33	34.18	0.6	R-50	53.0	2.0
Q-42	39.93	0.8	R-61	58.52	2.2
Q-44	42.59	0.9	R-64	62.02	2.3
Q-50	47.93	1.0	R-68	66.02	2.5
Q-55	53.43	1.1	R-74	72.02	2.7
Q-60	57.93	1.2	R-76	74 00	2.8
Q-65	63.43	1.3	R-80	77.84	3.0
Q-70	68.43	1.4	R-85	82 84	3.1
Q-76	74.43	1.6	R-90	87.84	3.3
Q-80	78.43	1.6	R-94	92.21	3.5
Q-90	88.43	1.8	R-100	97 84	3.7
Q-100	98.43	2.1	R-110	107.84	4.1
Q-110	108.43	2.3	R-120	117 84	4.4
			R-130	127.84	4.8
			R-150	147.84	5.6
			R-170	167.84	6.3

Table 14-53. Face Widths of Companion Sheaves

(Used with wide-range Vari-Pitch sheaves)

Size of Texrope belt	No. of grooves			
	1	2	3	4
Q section	1½ in.	3⅞ in.	6¼ in.	8⅝ in.
R section	....	4⅞ in.	7⅞ in.	10⅞ in.

Variable-speed drives using one Vari-Pitch sheave provide speed adjustments from 14 to 28 per cent for standard A-, B-, C-, D-, and E-section belts and 150 to 190 per cent for Q- and R-section belts. The use of two Vari-Pitch sheaves on a drive increases the variation proportionally, and the amount of variation depends on the sizes of sheaves used. Tables 14-61 to 14-64 show sizes of Vari-Pitch sheaves.

Companion sheaves are required for drives using Vari-Pitch sheaves owing to the construction of the sheave. The groove spacing of Vari-Pitch sheaves is not the same as for standard fixed-diameter V-belt sheaves. Table 14-65 (page 1025) gives face width of Companion sheaves.

Selecting a Vari-Pitch Drive. The data required for selecting a Vari-Pitch drive are the same as for a standard V-belt drive, and the rating is the same except when the drive is used for a constant horsepower application. In this case, the horsepower rating must be calculated using smallest pitch diameter.

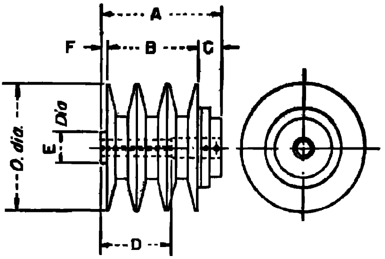
Horsepower tables for Q and R belts are given in Table 14-51. Serial numbers and pitch lengths of Q and R belts are given in Table 14-52. Face widths of Companion sheaves for Q and R belts are given in Table 14-53.

V-FLAT DRIVES

A small grooved sheave used with a large flat pulley and a set of V belts are most successful when the following conditions are satisfied:

1. The speed ratio is 3:1 or more.
2. The center distance is less than the diameter of the large pulley. Centers of ¾ the diameter of the large pulley are desirable and should not exceed 1½ times the diameter of the large pulley.

Table 14-54. Dimensions of Wide-range Vari-Pitch Sheaves



Belt size	Range of pitch diam	No. of grooves	Outside diam	Dimensions					
				A	B	C	D	E	F
Q	3.50– 7.50	1	7 7/16	4 7/8	2 3/8	2	3	1 7/16	1/2
Q	3.50– 7.50	2	7 7/16	7 1/4	4 3/4	2	3 3/4	1 7/16	1/2
Q	3.50– 7.50	3	7 7/16	9 5/8	7 1/8	2	4 1/4	1 7/16	1/2
Q	4.50– 8.00	1	8 7/16	4 7/8	2 3/8	2	3	2 1/4	1/2
Q	4.50– 8.00	2	8 7/16	7 1/4	4 3/4	2	3 3/4	2 1/4	1/2
Q	4.50– 8.00	3	8 7/16	9 5/8	7 1/8	2	5	2 1/4	1/2
Q	5.50– 9.00	2	9 7/16	7 1/4	4 3/4	2	3 3/4	2 1/4	1/2
Q	5.50– 9.00	3	9 7/16	9 5/8	7 1/8	2	5	2 1/4	1/2
Q	5.50– 9.00	4	9 7/16	12	9 1/2	2	6 1/2	2 1/4	1/2
R	5.25–10.00	2	10 3/8	9	6	2 3/8	5	3 1/8	5/8
R	5.25–10.00	3	10 3/8	12	9	2 3/8	7	3 1/8	5/8
R	5.25–10.00	4	10 3/8	15	12	2 3/8	7 3/4	3 1/8	5/8
R	6.25–11.00	2	11 3/8	9	6	2 3/8	5	3 1/8	5/8
R	6.25–11.00	3	11 3/8	12	9	2 3/8	7	3 1/8	5/8
R	6.25–11.00	4	11 3/8	15	12	2 3/8	7 3/8	3 1/8	5/8
R	7.25–12.00	2	12 3/8	9	6	2 3/8	5	3 1/8	5/8
R	7 25–12.00	3	12 3/8	12	9	2 3/8	7	3 1/8	5/8
R	7.25–12.00	4	12 3/8	15	12	2 3/8	7 3/4	3 1/8	5/8

Table 14-55. Correction Factors for V-flat Drives

Arc of contact	*Correction factor*
180	0.74
170	0.77
160	0.79
150	0.82
140	0.84
130	0.86
120	0.83
110	0.79
100	0.74
90	0.69

Table 14-56. V-flat Drives

Belt section	*Add to diam of flat pulley, in.*
A	0 35
B	0.45
C	0.50
D	0.75
E	1.00

Table 14-57. Deep-groove Sheave Face Width

No. of grooves	Sheave face width				
	A belt	B belt	C belt	D belt	E belt
1	3/4	1	1 3/8	1 7/8	2 1/8
2	1 3/8	1 7/8	2 9/16	3 1/2	4 1/8
3	2	2 3/4	3 3/4	5 1/8	6 1/8
4	2 5/8	3 5/8	4 15/16	6 3/4	8 1/8
5	3 1/4	4 1/2	6 1/8	8 3/8	10 1/8
6	3 7/8	5 3/8	7 5/16	10	12 1/8
7	4 1/2	6 1/4	8 1/2	11 5/8	14 1/8
8	5 1/8	7 1/8	9 11/16	13 1/4	16 1/8
9	5 3/4	8	10 7/8	14 7/8	18 1/8
10	6 3/8	8 7/8	12 1/16	16 1/2	20 1/8

3. The large pulley has a flat face (no crown).
4. The face width of the flat pulley is at least 1 in. wider than the face width of the grooved sheave. For large drives a 10 per cent increase in face width should be used.

Flat pulleys of cast iron or pressed steel are generally used. If crown-face pulleys are used, there will be unequal tension between the center and outer belts which may cause the belts to "crowd" toward the center of the crown.

Selecting a V-flat Drive. The correction factors for arc contact in Table 14-55 should be used. The pitch diameter of the flat pulley (for calculation purposes) is found by adding to the diameter of the flat pulley the amount shown in Table 14-56. All other data required are the same as for a V-belt drive.

QUARTER-TURN V-BELT DRIVE

Quarter-turn drives (Fig. 14-12) are used to transmit power from a horizontal shaft to a vertical shaft or vice versa. Speed ratios should not exceed 2½ to 1. Horsepower ratings of belts are 75 per cent of those listed in Tables 14-39 to 14-43. The arc of contact correction factor is disregarded. The center distance should not be less than $5(P + W)$ where P = pitch diameter of larger sheave, and W = face width of large sheave. The center line of the horizontal shaft should be half the diameter of the large sheave above the center line of the vertical shaft, and the vertical shaft should be offset half the diameter of the small sheave. Special deep-groove sheaves are required. Face widths of deep-groove sheaves are given in Table 14-57.

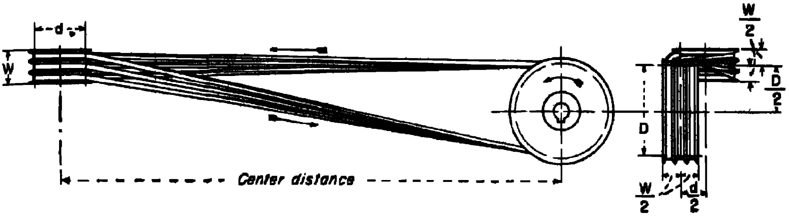


Fig. 14-12. Quarter-turn V-belt drive.

FLYWHEEL EFFECT (WR^2) OF V-BELT SHEAVES

In Fig. 14-13 the lines on the left, designated *B*, *C*, *D*, and *E* represent the standard sections of belts. To use the chart, find the place where the lines representing the pitch diameters of sheaves intersect the lines (WR^2 standard sheave) for the correct section of belt, and read on the right the value (WR^2) per inch of width. Multiply the face width of sheave in inches (Table 14-50) by the value read on chart, and the result will be the WR^2 of a standard sheave.

To find WR^2 for heavy flywheels where the value WR^2 is given, determine the WR^2 per inch by dividing the total WR^2 by the face width in inches, and read to left to intersect with diagonal line (pitch diameter) for size of sheave required. Then read at bottom of chart the extra weight to be added to a standard sheave (per inch of width). The total extra weight required to be added to a standard sheave will be found by multiplying the value read at bottom of Fig. 14-13 by the face width of sheave in inches.

"Magic-Grip" sheaves are of cast-iron construction and are made in two general types so far as location of hub is concerned. These two types are illustrated by Fig. 14-14. Tables 14-58 to 14-60 list the type in which each pitch diameter is furnished. In general sheaves up to 8.6 in. pitch diameter in B section, 10.6 in. pitch diameter in C section, and 18.0 in. in pitch diameter section are made in web style; larger sheaves have arm-style construction.

Note: The terms "Texrope," "Vari-Pitch" and "Magic-Grip" are trade-marks of Allis-Chalmers Manufacturing Company.

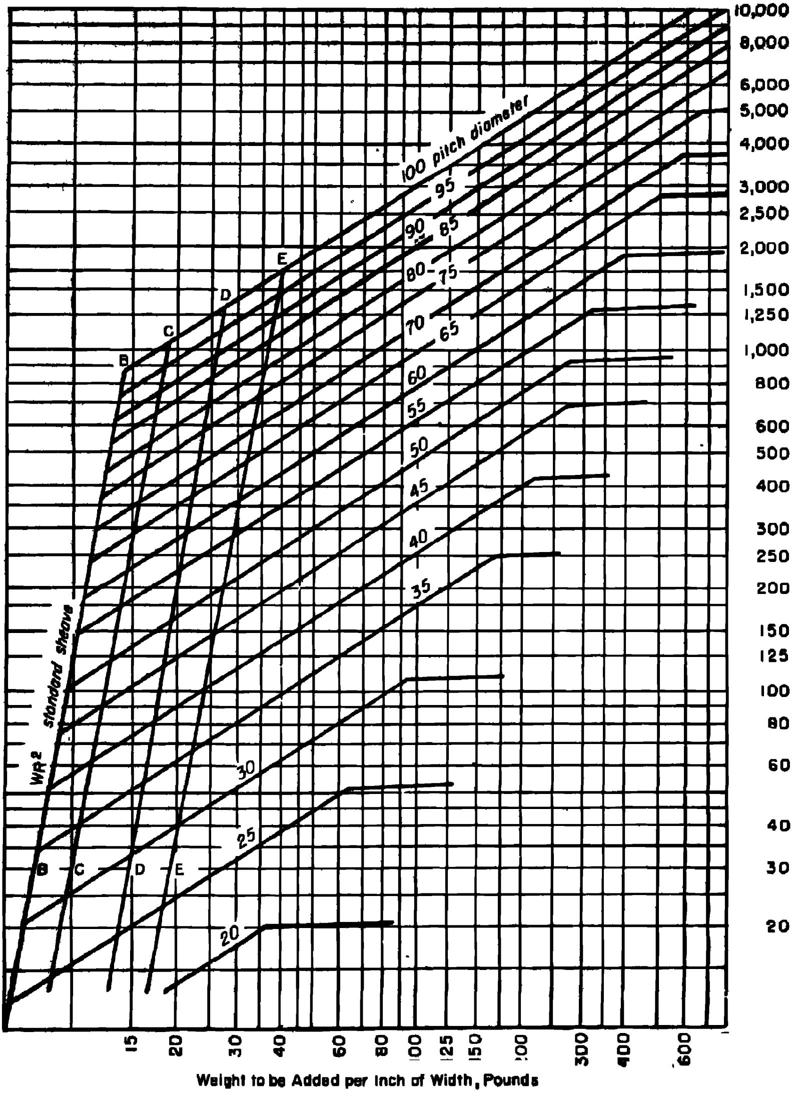

100 pitch diameter
95
90
85
80
75
70
65
60
55
50
45
40
35
30
25
20
B
C
D
E
WR² standard sheave
10,000
8,000
6,000
5,000
4,000
3,000
2,500
2,000
1,500
1,250
1,000
800
600
500
400
300
250
200
150
125
100
80
60
40
30
20
15
20
30
40
60
80
100
125
150
200
300
400
600
Weight to be Added per Inch of Width, Pounds

FIG. 14-13.

Table 14-58. Dimensions of Magic-Grip Sheaves, B Section, 2 to 30 Hp*

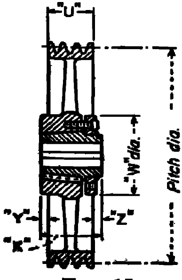
(Stock sheaves for B-section belts available in 2 to 10 grooves only)

All dimensions are given in inches. Sheaves above line are web style. Sheaves below line are arm style.

Pitch diam	2 grooves, $U = 1\frac{3}{4}$-in. face							3 grooves, $U = 2\frac{1}{2}$-in. face							4 grooves, $U = 3\frac{1}{4}$-in. face						
	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†
5.4	15	1	3 5/16	1 7/8	Flush	1/8	9	16	1	3 5/16	1 7/8	3/8	1/4	11	16	2	3 7/8	2 5/8	5/8	1/4	16
5.6	15	1	3 5/16	1 7/8	Flush	1/8	10	16	1	3 5/16	1 7/8	3/8	1/4	12	16	2	3 7/8	2 5/8	5/8	1/4	17
5.8	15	1	3 5/16	1 7/8	Flush	1/8	11	16	1	3 5/16	1 7/8	3/8	1/4	13	16	2	3 7/8	2 5/8	5/8	1/4	18
6.0	15	1	3 5/16	1 7/8	Flush	1/8	11	15	2	3 7/8	2 5/8	Flush	1/8	15	16	2	3 7/8	2 5/8	5/8	1/4	18
6.2	15	1	3 5/16	1 7/8	Flush	1/8	12	15	2	3 7/8	2 5/8	Flush	1/8	16	16	2	3 7/8	2 5/8	5/8	1/4	19
6.4	15	1	3 5/16	1 7/8	Flush	1/8	12	15	2	3 7/8	2 5/8	Flush	1/8	16	16	2	3 7/8	2 5/8	5/8	1/4	19
6.6	15	1	3 5/16	1 7/8	Flush	1/8	13	15	2	3 7/8	2 5/8	Flush	1/8	17	16	2	3 7/8	2 5/8	5/8	1/4	20
6.8	15	1	3 5/16	1 7/8	Flush	1/8	13	15	2	3 7/8	2 5/8	Flush	1/8	17	16	2	3 7/8	2 5/8	5/8	1/4	21
7.0	15	2	3 7/8	2 5/8	3/8	1/2	14	15	2	3 7/8	2 5/8	Flush	1/8	18	16	3	4 1/4	3	1/8	1/8	23
8.6	15	3	4 1/4	3	5/8	5/8	15	15	3	4 1/4	3	1/4	1/4	20	16	3	4 1/4	3	1/8	1/8	24
11.0	15	3	4 1/4	3	5/8	5/8	16	15	3	4 1/4	3	1/4	1/4	25	16	3	4 1/4	3	1/8	1/8	29
13.6	15	3	4 1/4	3	5/8	5/8	19	15	3	4 1/4	3	1/4	1/4	30	16	3	4 1/4	3	1/8	1/8	39
16.0	15	3	4 1/4	3	5/8	5/8	27	15	4	4 3/4	3 1/2	1/2	1/2	44	15	4	4 3/4	3 1/2	1/8	1/8	50
20.0	15	4	4 3/4	3 1/2	7/8	7/8	44	15	4	4 3/4	3 1/2	1/2	1/2	58	15	4	4 3/4	3 1/2	1/8	1/8	67
25.0	15	4	4 3/4	3 1/2	7/8	7/8	58	15	4	4 3/4	3 1/2	1/2	1/2	74	15	4	4 3/4	3 1/2	1/8	1/8	85
30.0	15	4	4 3/4	3 1/2	7/8	7/8	72	15	4	4 3/4	3 1/2	1/2	1/2	90	15	4	4 3/4	3 1/2	1/8	1/8	102
38.0	15	4	4 3/4	3 1/2	7/8	7/8	109	15	4	4 3/4	3 1/2	1/2	1/2	116	15	4	4 3/4	3 1/2	1/8	1/8	141

* Courtesy Allis-Chalmers Manufacturing Co.

† Approximate net weight of sheave and bushing of medium-size bore.



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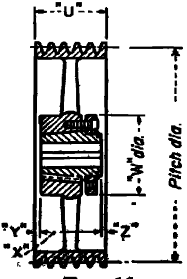

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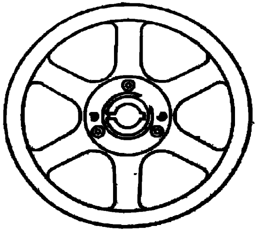

Table 14-58. Dimensions of Magic-Grip Sheaves, B Section, 2 to 30 Hp* (*Cont.*)

Pitch diam	5 grooves, *U* = 4-in. face							6 grooves, *U* = 4¾-in. face							7 grooves, *U* = 5½-in. face						
	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†
5.4	16	2	3⅞	2⅝	⅜	1	19	16	2	3⅞	2⅝	½	1⅝	21	16	2	3⅞	2⅝	¾	2⅛	23
5.6	16	2	3⅞	2⅝	⅜	1	20	16	2	3⅞	2⅝	½	1⅝	22	16	2	3⅞	2⅝	¾	2⅛	24
5.8	16	2	3⅞	2⅝	⅜	1	21	16	2	3⅞	2⅝	½	1⅝	23	16	2	3⅞	2⅝	¾	2⅛	25
6.0	16	2	3⅞	2⅝	⅜	1	22	16	3	4¼	3	½	1¼	24	16	3	4¼	3	¾	1¾	27
6.2	16	2	3⅞	2⅝	⅜	1	23	16	3	4¼	3	½	1¼	25	16	3	4¼	3	¾	1¾	28
6.4	16	2	3⅞	2⅝	⅜	1	24	16	3	4¼	3	½	1¼	26	16	3	4¼	3	¾	1¾	30
6.6	16	3	4¼	3	⅜	⅝	25	16	3	4¼	3	½	1¼	27	16	4	4¾	3½	¾	1¼	35
6.8	16	3	4¼	3	⅜	⅝	26	16	3	4¼	3	½	1¼	28	16	4	4¾	3½	¾	1¼	37
7.0	16	3	4¼	3	⅜	⅝	27	16	3	4¼	3	½	1¼	29	16	4	4¾	3½	¾	1¼	39
8.6	16	4	4¾	3½	¼	¼	36	16	4	4¾	3½	½	¾	38	16	4	4¾	3½	¾	1¼	41
11.0	16	4	4¾	3½	¼	¼	43	16	4	4¾	3½	½	¾	45	16	4	4¾	3½	¾	1¼	50
13.6	16	4	4¾	3½	¼	¼	49	16	4	4¾	3½	½	¾	54	16	4	4¾	3½	¾	1¼	60
16.0	16	4	4¾	3½	¼	¼	56	16	5	5¾	4½	⅛	⅛	76	16	5	5¾	4½	½	½	84
20.0	16	4	4¾	3½	¼	¼	74	16	5	5¾	4½	⅛	⅛	86	16	5	5¾	4½	½	½	104
25.0	16	4	4¾	3½	¼	¼	95	16	5	5¾	4½	⅛	⅛	120	16	5	5¾	4½	½	½	130
30.0	15	5	5¾	4½	¼	¼	131	16	5	5¾	4½	⅛	⅛	146	16	5	5¾	4½	½	½	164
38.0	15	5	5¾	4½	¼	¼	172	16	5	5¾	4½	⅛	⅛	195	16	5	5¾	4½	½	½	214

Pitch diam	8 grooves, *U* = 6¼-in. face							9 grooves, *U* = 7-in. face							10 grooves, *U* = 7¾-in. face						
	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†
5.4	16	2	3⅞	2⅝	1	2⅝	25	16	2	3⅞	2⅝	1¾	2⅝	27	16	2	3⅞	2⅝	2½	2⅝	28
5.6	16	2	3⅞	2⅝	1	2⅝	26	16	2	3⅞	2⅝	1¾	2⅝	29	16	2	3⅞	2⅝	2½	2⅝	30
5.8	16	2	3⅞	2⅝	1	2⅝	27	16	2	3⅞	2⅝	1¾	2⅝	30	16	2	3⅞	2⅝	2½	2⅝	32
6.0	16	3	4¼	3	1	2¼	30	16	3	4¼	3	1¾	2¼	32	16	3	4¼	3	2⅜	2⅜	35
6.2	16	3	4¼	3	1	2¼	31	16	3	4¼	3	1¾	2¼	33	16	3	4¼	3	2⅜	2⅜	36
6.4	16	3	4¼	3	1	2¼	32	16	3	4¼	3	1¾	2¼	35	16	3	4¼	3	2⅜	2⅜	38
6.6	16	4	4¾	3½	1	1¾	38	16	4	4¾	3½	1¾	1¾	42	16	4	4¾	3½	2⅛	2⅛	46
6.8	16	4	4¾	3½	1	1¾	40	16	4	4¾	3½	1¾	1¾	44	16	4	4¾	3½	2⅛	2⅛	47
7.0	16	4	4¾	3½	1	1¾	42	16	4	4¾	3½	1¾	1¾	46	16	4	4¾	3½	2⅛	2⅛	49
8.6	16	4	4¾	3½	1	1¾	45	16	4	4¾	3½	1¾	1¾	49	16	4	4¾	3½	2⅛	2⅛	52
11.0	16	4	4¾	3½	1	1¾	53	16	5	5¾	4½	1¼	1¼	71	16	5	5¾	4½	1⅝	1⅝	76
13.6	16	4	4¾	3½	1	1¾	65	16	5	5¾	4½	1¼	1¼	82	16	5	5¾	4½	1⅝	1⅝	88
16.0	16	5	5¾	4½	⅞	⅞	90	16	5	5¾	4½	1¼	1¼	98	16	5	5¾	4½	1⅝	1⅝	104
20.0	16	5	5¾	4½	⅞	⅞	113	16	5	5¾	4½	1¼	1¼	122	16	5	5¾	4½	1⅝	1⅝	131
25.0	16	5	5¾	4½	⅞	⅞	141	16	6	7	6	½	½	180	16	6	7	6	⅞	⅞	190
30.0	16	5	5¾	4½	⅞	⅞	176	16	6	7	6	½	½	227	16	6	7	6	⅞	⅞	250
38.0	16	5	5¾	4½	⅞	⅞	237	16	6	7	6	½	½	305	16	6	7	6	⅞	⅞	320

† Approximate net weight of sheave and bushing of medium-size bore

Table 14-59. Dimensions of Magic-Grip Sheaves, C Section, 15 to 125 Hp*

(Stock sheaves for C-section belts available in 3 to 10 grooves only.)

All dimensions are given in inches. Sheaves above line are web style. Sheaves below line are arm style.

Pitch diam	3 grooves, U = 3⅝-in. face							4 grooves, U = 4⅝-in. face						
	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†
9.0	15	4	4¾	3½	1/16	1/16	36	16	4	4¾	3½	7/16	7/16	42
9.2	15	4	4¾	3½	1/16	1/16	37	16	4	4¾	3½	7/16	7/16	43
9.4	15	4	4¾	3½	1/16	1/16	38	16	4	4¾	3½	7/16	7/16	44
9.6	15	4	4¾	3½	1/16	1/16	38	16	4	4¾	3½	7/16	7/16	45
9.8	15	4	4¾	3½	1/16	1/16	39	16	4	4¾	3½	7/16	7/16	46
10.0	15	4	4¾	3½	1/16	1/16	40	16	4	4¾	3½	7/16	7/16	47
10.2	15	4	4¾	3½	1/16	1/16	41	16	4	4¾	3½	7/16	7/16	48
10.6	15	4	4¾	3½	1/16	1/16	43	16	4	4¾	3½	7/16	7/16	49
13.0	15	4	4¾	3½	1/16	1/16	49	16	4	4¾	3½	7/16	7/16	59
16.0	15	5	5¾	4½	9/16	9/16	73	15	5	5¾	4½	1/16	1/16	84
20.0	15	5	5¾	4½	9/16	9/16	90	15	5	5¾	4½	1/16	1/16	105
24.0	15	5	5¾	4½	9/16	9/16	110	15	5	5¾	4½	1/16	1/16	127
30.0	15	5	5¾	4½	9/16	9/16	132	15	6	7	6	1 3/16	1 3/16	170
36.0	15	5	5¾	4½	9/16	9/16	175	15	6	7	6	1 3/16	1 3/16	230
44.0	15	5	5¾	4½	9/16	9/16	210	15	6	7	6	1 3/16	1 3/16	250
50.0	15	5	5¾	4½	9/16	9/16	260	15	6	7	6	1 3/16	1 3/16	325

* Courtesy Allis-Chalmers Manufacturing Co.

† Approximate net weight of sheave and bushing of medium-size bore.

Table 14-59. Dimensions of Magic-Grip Sheaves, C Section, 15 to 125 Hp* (*Cont.*)

Pitch diam	5 grooves, U = 5 5/8-in. face							6 grooves, U = 6 5/8-in. face							7 grooves, U = 7 5/8-in. face						
	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†
9.0	16	4	4 3/4	3 1/2	1/2	1 3/8	48	16	5	5 3/4	4 1/2	1/2	1 3/8	70	16	5	5 3/4	4 1/2	3/4	2 1/8	76
9.2	16	4	4 3/4	3 1/2	1/2	1 3/8	49	16	5	5 3/4	4 1/2	1/2	1 3/8	72	16	5	5 3/4	4 1/2	3/4	2 1/8	78
9.4	16	4	4 3/4	3 1/2	1/2	1 3/8	50	16	5	5 3/4	4 1/2	1/2	1 3/8	73	16	5	5 3/4	4 1/2	3/4	2 1/8	79
9.6	16	4	4 3/4	3 1/2	1/2	1 3/8	51	16	5	5 3/4	4 1/2	1/2	1 3/8	74	16	5	5 3/4	4 1/2	3/4	2 1/8	80
9.8	16	4	4 3/4	3 1/2	1/2	1 3/8	52	16	5	5 3/4	4 1/2	1/2	1 3/8	75	16	5	5 3/4	4 1/2	3/4	2 1/8	81
10.0	16	4	4 3/4	3 1/2	1/2	1 3/8	53	16	5	5 3/4	4 1/2	1/2	1 3/8	76	16	5	5 3/4	4 1/2	3/4	2 1/8	82
10.2	16	4	4 3/4	3 1/2	1/2	1 3/8	54	16	5	5 3/4	4 1/2	1/2	1 3/8	77	16	5	5 3/4	4 1/2	3/4	2 1/8	84
10.6	16	4	4 3/4	3 1/2	1/2	1 3/8	56	16	5	5 3/4	4 1/2	1/2	1 3/8	80	16	5	5 3/4	4 1/2	3/4	2 1/8	87
13.0	16	5	5 3/4	4 1/2	7/16	7/16	80	16	5	5 3/4	4 1/2	1 5/16	1 5/16	90	16	6	7	6	1 1/16	1 1/16	125
16.0	16	5	5 3/4	4 1/2	7/16	7/16	96	16	6	7	6	3/16	3/16	135	16	6	7	6	1 1/16	1 1/16	147
20.0	16	5	5 3/4	4 1/2	7/16	7/16	120	16	6	7	6	3/16	3/16	160	16	6	7	6	1 1/16	1 1/16	175
24.0	16	5	5 3/4	4 1/2	7/16	7/16	145	16	6	7	6	3/16	3/16	186	16	6	7	6	1 1/16	1 1/16	204
30.0	15	6	7	6	5/16	5/16	204	16	6	7	6	3/16	3/16	226	15	7	8 1/4	7 1/2	1/16	1/16	297
36.0	15	6	7	6	5/16	5/16	255	16	6	7	6	3/16	3/16	282	15	7	8 1/4	7 1/2	1/16	1/16	356
44.0	15	6	7	6	5/16	5/16	302	16	6	7	6	3/16	3/16	336	15	7	8 1/4	7 1/2	1/16	1/16	435
50.0	15	6	7	6	5/16	5/16	364	16	6	7	6	3/16	3/16	400	15	7	8 1/4	7 1/2	1/16	1/16	518

Pitch diam	8 grooves, U = 8 5/8-in. face							9 grooves, U = 9 5/8-in. face							10 grooves, U = 10 5/8-in. face						
	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†
9.0	16	5	5 3/4	4 1/2	1	2 7/8	83	16	6	7	6	1	2 3/8	88	16	6	7	6	1	3 3/8	107
9.2	16	5	5 3/4	4 1/2	1	2 7/8	85	16	6	7	6	1	2 3/8	91	16	6	7	6	1	3 3/8	110
9.4	16	5	5 3/4	4 1/2	1	2 7/8	86	16	6	7	6	1	2 3/8	93	16	6	7	6	1	3 3/8	112
9.6	16	5	5 3/4	4 1/2	1	2 7/8	88	16	6	7	6	1	2 3/8	95	16	6	7	6	1	3 3/8	115
9.8	16	5	5 3/4	4 1/2	1	2 7/8	90	16	6	7	6	1	2 3/8	98	16	6	7	6	1	3 3/8	118
10.0	16	5	5 3/4	4 1/2	1	2 7/8	92	16	6	7	6	1	2 3/8	100	16	6	7	7	1	3 3/8	121
10.2	16	5	5 3/4	4 1/2	1	2 7/8	93	16	6	7	6	1	2 3/8	103	16	6	7	6	1	3 3/8	123
10.6	16	5	5 3/4	4 1/2	1	2 7/8	97	16	6	7	6	1	2 3/8	130	16	6	7	6	1	3 3/8	129
13.0	16	6	7	6	1 3/16	1 3/16	134	16	6	7	6	1 11/16	1 11/16	176	16	6	7	6	2 3/16	2 3/16	153
16.0	16	6	7	6	1 3/16	1 3/16	159	16	7	8 1/4	7 1/2	1 5/16	1 5/16	208	16	7	8 1/4	7 1/2	1 7/16	1 7/16	220
20.0	16	6	7	6	1 3/16	1 3/16	190	16	7	8 1/4	7 1/2	1 5/16	1 5/16	245	16	7	8 1/4	7 1/2	1 7/16	1 7/16	260
24.0	16	6	7	6	1 3/16	1 3/16	220	16	7	8 1/4	7 1/2	1 5/16	1 5/16	282	16	7	8 1/4	7 1/2	1 7/16	1 7/16	300
30.0	16	7	8 1/4	7 1/2	7/16	7/16	318	16	7	8 1/4	7 1/2	1 5/16	1 5/16	340	16	7	8 1/4	7 1/2	1 7/16	1 7/16	362
36.0	16	7	8 1/4	7 1/2	7/16	7/16	384	16	7	8 1/4	7 1/2	1 5/16	1 5/16	410	16	7	8 1/4	7 1/2	1 7/16	1 7/16	436
44.0	16	7	8 1/4	7 1/2	7/16	7/16	470	16	7	8 1/4	7 1/2	1 5/16	1 5/16	500	16	7	8 1/4	7 1/2	1 7/16	1 7/16	535
50.0	16	7	8 1/4	7 1/2	7/16	7/16	556	16	7	8 1/4	7 1/2	1 5/16	1 5/16	594	16	7	8 1/4	7 1/2	1 7/16	1 7/16	620

† Approximate net weight of shea and bushing of medium-size bore.

Table 14-60. Dimensions of Magic-Grip Sheaves, D Section, 60 to 250 Hp*

(Stock sheaves for D-section belts available in 5 to 12 grooves only.)

All dimensions are given in inches. Sheaves above line are web style. Sheaves below line are arm style.

Pitch diam	5 grooves, U = 7½-in. face							6 grooves, U = 8¹⁵⁄₁₆-in. face						
	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†
13.0	16	6	7	6	¾	¾	153	16	6	7	6	¾	2³⁄₁₆	170
13.4	16	6	7	6	¾	¾	160	16	6	7	6	¾	2³⁄₁₆	176
13 8	16	6	7	6	¾	¾	165	16	6	7	6	¾	2³⁄₁₆	182
14.2	16	6	7	6	¾	¾	172	16	6	7	6	¾	2³⁄₁₆	190
14.6	16	6	7	6	¾	¾	174	16	6	7	6	¾	2³⁄₁₆	192
15.0	16	6	7	6	¾	¾	177	16	6	7	6	¾	2³⁄₁₆	197
15 4	16	6	7	6	¾	¾	185	16	6	7	6	¾	2³⁄₁₆	203
18 0	16	6	7	6	¾	¾	194	16	7	8¼	7½	¾	1¹⁄₁₆	274
22 0	16	6	7	6	¾	¾	236	16	7	8¼	7½	¾	1¹⁄₁₆	336
27 0	16	7	8¼	7½	Flush	Flush	325	16	7	8¼	7½	²³⁄₃₂	²³⁄₃₂	362
33 0	16	7	8¼	7½	Flush	Flush	416	16	7	8¼	7½	²³⁄₃₂	²³⁄₃₂	460
40 0	16	7	8¼	7½	Flush	Flush	516	16	7	8¼	7½	²³⁄₃₂	²³⁄₃₂	572
48 0	16	7	8¼	7½	Flush	Flush	672	16	7	8¼	7½	²³⁄₃₂	²³⁄₃₂	750

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* Courtesy Allis-Chalmers Manufacturing Co.

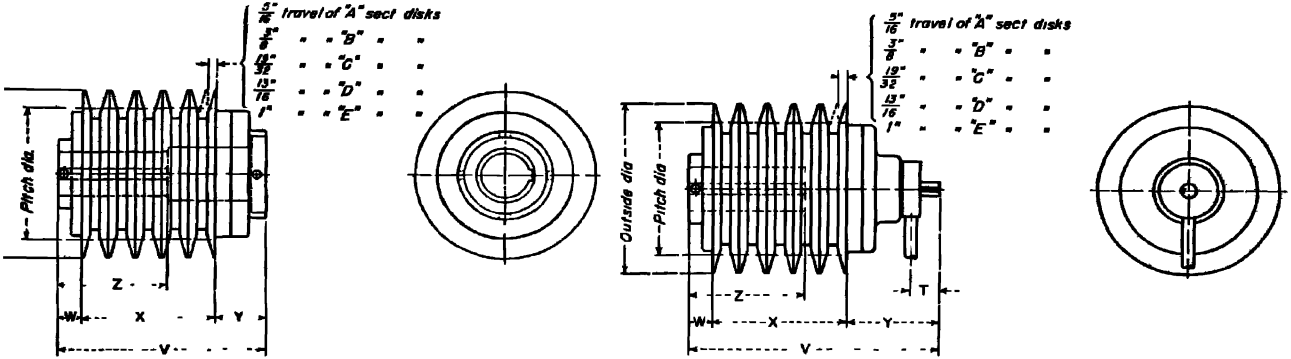
† Approximate net weight of sheave and bushing of medium-size bore.

le 14-60. Dimensions of Magic-Grip Sheaves, D Section, 50 to 250 Hp* (*Cont.*)

Pitch diam	7 grooves, *U* = 10⅝-in. face							8 grooves, *U* = 11¹³⁄₁₆-in. faces							9 grooves, *U* = 13¼-in. face						
	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†
13.0	16	6	7	6	¾	3⅝	184	16	7	8¼	7½	1½	2¹³⁄₁₆	230	16	7	8¼	7½	1½	4¼	244
13.4	16	6	7	6	¾	3⅝	192	16	7	8¼	7½	1½	2¹³⁄₁₆	238	16	7	8¼	7½	1½	4¼	254
13.8	16	6	7	6	¾	3⅝	200	16	7	8¼	7½	1½	2¹³⁄₁₆	246	16	7	8¼	7½	1½	4¼	264
14.2	16	6	7	6	¾	3⅝	208	16	7	8¼	7½	1½	2¹³⁄₁₆	256	16	7	8¼	7½	1½	4¼	274
14.6	16	6	7	6	¾	3⅝	210	16	7	8¼	7½	1½	2¹³⁄₁₆	258	16	7	8¼	7½	1½	4¼	275
15.0	16	6	7	6	¾	3⅝	215	16	7	8¼	7½	1½	2¹³⁄₁₆	265	16	7	8¼	7½	1½	4¼	282
15.4	16	6	7	6	¾	3⅝	223	16	7	8¼	7½	1½	2¹³⁄₁₆	275	16	7	8¼	7½	1½	4¼	294
18.0	16	7	8¼	7½	¾	2⅛	296	16	7	8¼	7½	1½	2¹³⁄₁₆	318	16	8	9¾	9	1½	2¾	390
22.0	16	7	8¼	7½	¾	2⅛	361	16	7	8¼	7½	1½	2¹³⁄₁₆	390	16	8	9¾	9	1½	2¾	415
27.0	16	7	8¼	7½	1⁷⁄₁₆	1⁷⁄₁₆	420	16	8	9¾	9	1¹³⁄₃₂	1¹³⁄₃₂	512	16	8	9¾	9	2⅛	2⅛	560
33.0	16	7	8¼	7½	1⁷⁄₁₆	1⁷⁄₁₆	556	16	8	9¾	9	1¹³⁄₃₂	1¹³⁄₃₂	660	16	8	9¾	9	2⅛	2⅛	705
40.0	16	7	8¼	7½	1⁷⁄₁₆	1⁷⁄₁₆	704	16	8	9¾	9	1¹³⁄₃₂	1¹³⁄₃₂	820	16	8	9¾	9	2⅛	2⅛	895
48.0	16	7	8¼	7½	1⁷⁄₁₆	1⁷⁄₁₆	900	16	8	9¾	9	1¹³⁄₃₂	1¹³⁄₃₂	1024	16	8	9¾	9	2⅛	2⅛	1120

Pitch diam	10 grooves, *U* = 14¹¹⁄₁₆-in. face							11 grooves, *U* = 16⅛-in. face							12 grooves, *U* = 17⁹⁄₁₆-in. face						
	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†	Type	Bush. No.	W	X	Y	Z	Net wt†
13.0	16	7	8¼	7½	1½	5¹¹⁄₁₆	260	16	8	9¾	9	2	5⅛	274	16	8	9¾	9	2	6⁹⁄₁₆	320
13.4	16	7	8¼	7½	1½	5¹¹⁄₁₆	270	16	8	9¾	9	2	5⅛	290	16	8	9¾	9	2	6⁹⁄₁₆	336
13.8	16	7	8¼	7½	1½	5¹¹⁄₁₆	280	16	8	9¾	9	2	5⅛	305	16	8	9¾	9	2	6⁹⁄₁₆	352
14.2	16	7	8¼	7½	1½	5¹¹⁄₁₆	290	16	8	9¾	9	2	5⅛	322	16	8	9¾	9	2	6⁹⁄₁₆	370
14.6	16	7	8¼	7½	1½	5¹¹⁄₁₆	293	16	8	9¾	9	2	5⅛	324	16	8	9¾	9	2	6⁹⁄₁₆	372
15.0	16	7	8¼	7½	1½	5¹¹⁄₁₆	300	16	8	9¾	9	2	5⅛	338	16	8	9¾	9	2	6⁹⁄₁₆	385
15.4	16	7	8¼	7½	1½	5¹¹⁄₁₆	315	16	8	9¾	9	2	5⅛	354	16	8	9¾	9	2	6⁹⁄₁₆	415
18.0	16	8	9¾	9	1½	4³⁄₁₆	410	16	8	9¾	9	2	5⅛	440	16	8	9¾	9	2	6⁹⁄₁₆	465
22.0	16	8	9¾	9	1½	4³⁄₁₆	492	16	8	9¾	9	2	5⅛	535	16	8	9¾	9	2	6⁹⁄₁₆	565
27.0	16	8	9¾	9	2²⁷⁄₃₂	2²⁷⁄₃₂	657	16	8	9¾	9	3⁹⁄₁₆	3⁹⁄₁₆	650	16	8	9¾	9	4⁹⁄₃₂	4⁹⁄₃₂	655
33.0	16	8	9¾	9	2²⁷⁄₃₂	2²⁷⁄₃₂	750	16	8	9¾	9	3⁹⁄₁₆	3⁹⁄₁₆	810	16	8	9¾	9	4⁹⁄₃₂	4⁹⁄₃₂	865
40.0	16	8	9¾	9	2²⁷⁄₃₂	2²⁷⁄₃₂	950	16	8	9¾	9	3⁹⁄₁₆	3⁹⁄₁₆	1025	16	8	9¾	9	4⁹⁄₃₂	4⁹⁄₃₂	1085
48.0	16	8	9¾	9	2²⁷⁄₃₂	2²⁷⁄₃₂	1190	16	8	9¾	9	3⁹⁄₁₆	3⁹⁄₁₆	1300	16	8	9¾	9	4⁹⁄₃₂	4⁹⁄₃₂	1370

Table 14-61. General Dimensions of Stationary-control Vari-Pitch Sheaves, Combination Groove, for A- or B-section Texrope Belts*



All dimensions are given in inches

Range of pitch diam†		Outside diam	2 grooves						3 grooves						4 grooves					
A	B		Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z
2 9–4 0	3 0– 4 4	4 375	1¼	5	1	2⅛	1⅞	3	1¼	6⅛	1	3¼	1⅞	3	1¼	7¼	1	4⅜	1⅞	3½
3 4–4 5	3 5– 4 9	5 187	1¼	5¼	1	2¼	2	3	1¼	6⅜	1	3⅜	2	3	1¼	7¼	1	4½	2	3½
3 9–5 0	4 0– 5 4	5 687	1⅝	5¼	1	2¼	2	3	1⅝	6⅜	1	3⅜	2	3	1⅝	7½	1	4½	2	3½
4 4–5 5	4 5– 5 9	6 187	1⅝	5¼	1	2¼	2	3	1⅝	6⅜	1	3⅜	2	3	1⅝	7½	1	4½	2	3½
4 9–6 0	5 0– 6 4	6 687	2⅜	5 7/16	1	2¼	2 3/16	3	2⅜	6 9/16	1	3⅜	2 3/16	3⅜	2⅜	7 11/16	1	4½	2 3/16	4
5 4–6 5	5 5– 6 9	7 187	2⅜	5 7/16	1	2¼	2 3/16	3	2⅜	6 9/16	1	3⅜	2 3/16	3⅜	2⅜	7 11/16	1	4½	2 3/16	4
5 9–7 0	6 0– 7 4	7 687	2⅜	5 7/16	1	2¼	2 3/16	3	2⅜	6 9/16	1	3⅜	2 3/16	3⅜	2⅜	7 11/16	1	4½	2 3/16	4
6 9–8 0	7 0– 8 4	8 687	2⅜	5 7/16	1	2¼	2 3/16	3	2⅜	6 9/16	1	3⅜	2 3/16	3⅜	2⅜	7 11/16	1	4½	2 3/16	4
7 9–9 0	8 0– 9 4	9 687	2⅜	5 7/16	1	2¼	2 3/16	3	2⅜	6 9/16	1	3⅜	2 3/16	3⅜	2⅜	7 11/16	1	4½	2¾	4
	9 0–10 4	10 687	3½	6 7/16	1½	2¼	2 11/16	3¾	3½	7 9/16	1½	3⅜	2 11/16	4	3½	8 11/16	1½	4½	2 11/16	4¼
	10 0–11 4	11 687	3½	6 7/16	1½	2¼	2 11/16	3¾	3½	7 9/16	1½	3⅜	2 11/16	4	3½	8 11/16	1½	4½	2 11/16	4¼
	11 0–12 4	12 687	3½	6 7/16	1½	2¼	2 11/16	3¾	3½	7 9/16	1½	3⅜	2 11/16	4	3½	8 11/16	1½	4½	2 11/16	4¼

Table 14-61. General Dimensions of Stationary-control Vari-Pitch Sheaves, Combination Groove, for A- or B-section Texrope Belts* *(Cont.)*

Range of pitch diam		Outside diam	5 grooves						6 grooves					
A	B		Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z
2 9–4 0	3 0– 4 4													
3 4–4 5	3 5– 4 9	5 187	1¼	8⅝	1	5⅝	2	4	1¼	9¾	1	6¾	2	4½
3 9–5 0	4 0– 5 4	5 687	1⅝	8⅝	1	5⅝	2	4	1⅝	9¾	1	6¾	2	4½
4 4–5 5	4 5– 5 9	6 187	1⅝	8⅝	1	5⅝	2	4	1⅝	9¾	1	6¾	2	4½
4 9–6 0	5 0– 6 4	6 687	2⅜	8 13/16	1	5⅝	2 3/16	4⅝	2⅜	9 15/16	1	6¾	2 3/16	5¼
5 4–6 5	5 5– 6 9	7 187	2⅜	8 13/16	1	5⅝	2 3/16	4⅝	2⅜	9 15/16	1	6¾	2 3/16	5¼
5 9–7 0	6 0– 7 4	7 687	2⅜	8 13/16	1	5⅝	2 3/16	4⅝	2⅜	9 15/16	1	6¾	2 3/16	5¼
6 9–8 0	7 0– 8 4	8 687	2⅜	8 13/16	1	5⅝	2 3/16	4⅝	2⅜	9 15/16	1	6¾	2 3/16	5¼
7 9–9 0	8 0– 9 4	9 687	2⅜	8 13/16	1	5⅝	2 3/16	4⅝	2⅜	9 15/16	1	6¾	2 3/16	5¼
	9 0–10 4	10 687	3½	9 13/16	1½	5⅝	2 11/16	4¾	3½	10 15/16	1½	6¾	2 11/16	5⅝
	10 0–11 4	11 687	3½	9 13/16	1½	5⅝	2 11/16	4¾	3½	10 15/16	1½	6¾	2 11/16	5⅝
	11 0–12 4	12 687	3½	9 13/16	1½	5⅝	2 11/16	4¾	3½	10 15/16	1½	6¾	2 11/16	5⅝

Range of pitch diam		Outside diam	7 grooves						8 grooves					
A	B		Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z
2 9–4 0	3 0– 4 4													
3 4–4 5	3 5– 4 9	5 187	1¼	10⅞	1	7⅞	2	5⅛	1¼	12	1	9	2	5¼
3 9–5 0	4 0– 5 4	5 687	1⅝	10⅞	1	7⅞	2	5	1⅝	12	1	9	2	5¼
4 4–5 5	4 5– 5 9	6 187	1⅝	10⅞	1	7⅞	2	5	1⅝	12	1	9	2	5¼
4 9–6 0	5 0– 6 4	6 687	2⅜	11 1/16	1	7⅞	2 3/16	5⅞	2⅜	12 3/16	1	9	2 3/16	6½
5 4–6 5	5 5– 6 9	7 187	2⅜	11 1/16	1	7⅞	2 3/16	5⅞	2⅜	12 3/16	1	9	2 3/16	6½
5 9–7 0	6 0– 7 4	7 687	2⅜	11 1/16	1	7⅞	2 3/16	5⅞	2⅜	12 3/16	1	9	2 3/16	6½
6 9–8 0	7 0– 8 4	8 687	2⅜	11 1/16	1	7⅞	2 3/16	5⅞	2⅜	12 3/16	1	9	2 3/16	6½
7 9–9 0	8 0– 9 4	9 687	2⅜	11 1/16	1	7⅞	2 3/16	5⅞	2⅜	12 3/16	1	9	2 3/16	6½
	9 0–10 4	10 687	3½	12 1/16	1½	7⅞	2 11/16	6	3½	13 3/16	1½	9	2 11/16	6⅞
	10 0–11 4	11 687	3½	12 1/16	1½	7⅞	2 11/16	6	3½	13 3/16	1½	9	2 11/16	6⅞
	11 0–12 4	12 687	3½	12 1/16	1½	7⅞	2 11/16	6	3½	13 3/16	1½	9	2 11/16	6⅞

* Courtesy Allis-Chalmers Manufacturing Co

† Number of turns of adjusting nut to make a complete change in pitch diameter A section, 3¾ turns, B section, 4½ turns

Table 14-62. General Dimensions of Stationary-control Vari-Pitch Sheaves for Texrope Belts*

C-section Texrope belts

All dimensions are given in inches

Range of pitch diam†	Outside diam	2 grooves						3 grooves						4 grooves						5 grooves					
		Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z
8 5–10 7	11 063	3½	7 13/16	1⅜	3 9/32	2¾	3¾	3½	9 1/16	1⅜	4 15/16	2¾	4½	3½	10 23/32	1⅜	6 19/32	2¾	5¼	3½	12⅜	1⅜	8¼	2¾	6
9 5–11 7	12 063	3½	7 13/32	1⅜	3 9/32	2¾	3¾	3½	9 1/16	1⅜	4 15/16	2¾	4½	3½	10 23/32	1⅜	6 19/32	2¾	5¼	3½	12⅜	1⅜	8¼	2¾	6
10 5–12 7	13 063	3½	7 13/32	1⅜	3 9/32	2¾	3¾	3½	9 1/16	1⅜	4 15/16	2¾	4½	3½	10 23/32	1⅜	6 19/32	2¾	5¼	3½	12⅜	1⅜	8¼	2¾	6
11 5–13 7	14 063	3½	7 13/32	1⅜	3 9/32	2¾	3¾	2½	9 1/16	1⅜	4 15/16	2¾	4½	3½	10 23/32	1⅜	6 19/32	2¾	5¼	3½	12⅜	1⅜	8¼	2¾	6
12 5–14 7	15 063	3½	7 13/32	1⅜	3 9/32	2¾	3¾	3½	9 1/16	1⅜	4 15/16	2¾	4½	3½	10 23/32	1⅜	6 19/32	2¾	5¼	3½	12⅜	1⅜	8¼	2¾	6
13 5–15 7	16 063	3½	7 13/32	1⅜	3 9/32	2¾	3¾	3½	9 1/16	1⅜	4 15/16	2¾	4½	3½	10 23/32	1⅜	6 19/32	2¾	5¼	3½	12⅜	1⅜	8¼	2¾	6

Range of pitch diam	Outside diam	6 grooves						7 grooves						8 grooves					
		Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z
8 5–10 7	11 063	3½	14 1/32	1⅜	9 29/32	2¾	6 5/16	3½	15 11/16	1⅜	11 9/16	2¾	7	3½	17 11/32	1⅜	13 7/32	2¾	8
9 5–11 7	12 063	3½	14 1/32	1⅜	9 29/32	2¾	6 5/16	3½	15 11/16	1⅜	11 9/16	2¾	7	3½	17 11/32	1⅜	13 7/32	2¾	8
10 5–12 7	13 063	3½	14 1/32	1⅜	9 29/32	2¾	6 5/16	3½	15 11/16	1⅜	11 9/16	2¾	7	3½	17 11/32	1⅜	13 7/32	2¾	8
11 5–13 7	14 063	3½	14 1/32	1⅜	9 29/32	2¾	6 5/16	3½	15 11/16	1⅜	11 9/16	2¾	7	3½	17 11/32	1⅜	13 7/32	2¾	8
12 5–14 7	15 063	3½	14 1/32	1⅜	9 29/32	2¾	6 5/16	3½	15 11/16	1⅜	11 9/16	2¾	7	3½	17 11/32	1⅜	13 7/32	2¾	8
13 5–15 7	16 063	3½	14 1/32	1⅜	9 29/32	2¾	6 5/16	3½	15 11/16	1⅜	11 9/16	2¾	7	3½	17 11/32	1⅜	13 7/32	2¾	8

D-section Texrope belts

Range of pitch diam‡	Outside diam	3 grooves								4 grooves								5 grooves								6 grooves							
		Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z
			Min	Max							Min	Max							Min	Max							Min	Max					
12 0–15 0	15 625	4½	1¾	2 9/16	14 1/16	2	6⅞	5 13/16	5 15/16	4½	1¾	2 9/16	16⅜	2	9 9/16	5 5/16	8¼	4½	1¾	2 9/16	19 3/16	2	11½	5 11/16	8 13/16	4½	1¾	2 9/16	21½	2	13 13/16	5 11/16	11⅛
13 0–16 0	16 625	4½	1¾	2 9/16	14 1/16	2	6⅞	5 13/16	5 15/16	4½	1¾	2 9/16	16⅜	2	9 9/16	5 5/16	8¼	4½	1¾	2 9/16	19 3/16	2	11½	5 11/16	8 13/16	4½	1¾	2 9/16	21½	2	13 13/16	5 11/16	11⅛
14 0–17 0	17 625	4½	1¾	2 9/16	14 1/16	2	6⅞	5 13/16	5 15/16	4½	1¾	2 9/16	16⅜	2	9 9/16	5 5/16	8¼	4½	1¾	2 9/16	19 3/16	2	11½	5 11/16	8 13/16	4½	1¾	2 9/16	21½	2	13 13/16	5 11/16	11⅛

Table 14-62. General Dimensions of Stationary-control Vari-Pitch Sheaves for Texrope Belts* (*Cont.*)

D-section Texrope belts (*Cont.*)

Range of pitch diam	Outside diam	7 grooves								8 grooves								9 grooves								10 grooves							
		Max bore	T		V	W	X	Y[†]	Z	Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z
			Min	Max							Min	Max							Min	Max							Min	Max					
12 0-15 0	15 625	4½	1¾	2 9/16	23 13/16	2	16⅛	5 11/16	13 7/16	4½	1¾	2 9/16	26⅛	2	18 7/16	5 11/16	15¾	4½	1¾	2 9/16	28 7/16	2	20¾	5 11/16	18 1/16	4½	1¾	2 9/16	30¾	2	23 1/16	5 11/16	20⅝
13 0-16 0	16 625	4½	1¾	2 9/16	23 13/16	2	16⅛	5 11/16	13 7/16	4½	1¾	2 9/16	26⅛	2	18 7/16	5 11/16	15¾	4½	1¾	2 9/16	28 7/16	2	20¾	5 11/16	18 1/16	4½	1¾	2 9/16	30¾	2	23 1/16	5 11/16	20⅝
14 0-17 0	17 625	4½	1¾	2 9/16	23 13/16	2	16⅛	5 11/16	13 7/16	4½	1¾	2 9/16	26⅛	2	18 7/16	5 11/16	15¾	4½	1¾	2 9/16	28 7/16	2	20¾	5 11/16	18 1/16	4½	1¾	2 9/16	30¾	2	23 1/16	5 11/16	20⅝

E-section Texrope belts

Range of pitch diam§	Outside diam	5 grooves						6 grooves						7 grooves						8 grooves					
		Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z
20 0-23 75	24 5	5½	23¼	2⅝	14 15/16	5 11/16	12½	5½	26¼	2⅝	17 15/16	5 11/16	14	5½	29¼	2⅝	20 15/16	5 11/16	15½	5½	32¼	2⅝	23 15/16	5 11/16	17

Range of pitch diam	Outside diam	9 grooves						10 grooves						11 grooves						12 grooves					
		Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z
20 0-23 75	24 5	5½	35¼	2⅝	26 15/16	5 11/16	18½	5½	38¼	2⅝	29 15/16	5 11/16	20	5½	41¼	2⅝	32 15/16	5 11/16	21½	5½	44¼	2⅝	35 15/16	5 11/16	23

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* Courtesy Allis-Chalmers Manufacturing Co.

† Number of turns of adjusting nut to make a complete change in pitch diameter, 7⅛ turns.

‡ Number of turns of adjusting nut to make a complete change in pitch diameter, 9¾ turns.

§ Number of turns of adjusting nut to make a complete change in pitch diameter, 12 turns.

Table 14-63. General Dimensions of Stub-shaft Type Motion-control Vari-Pitch Sheaves for A- or B-section Texrope Belts*

All dimensions are given in inches.

Range of pitch diam†		Out-side diam	2 grooves								3 grooves								4 grooves							
A	B		Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z
				Min	Max							Min	Max							Min	Max					
3.4–4.5	3.5– 4.9	5.187	1 1/4	1 3/16	1 9/16	8	2 1/8	2 1/4	3 5/8	2 7/8	1 1/4	1 3/16	1 9/16	8	1	3 3/8	3 5/8	2 7/8	1 1/4	1 3/16	1 9/16	9 1/8	1	4 1/2	3 5/8	4
3.9–5.0	4.0– 5.4	5.687	1 5/8	1 3/16	1 9/16	8	2 1/8	2 1/4	3 5/8	2 7/8	1 5/8	1 3/16	1 9/16	8	1	3 3/8	3 5/8	2 7/8	1 5/8	1 3/16	1 9/16	9 1/8	1	4 1/2	3 5/8	4
4.4–5.5	4.5– 5.9	6.187	1 5/8	1 3/16	1 9/16	8	2 1/8	2 1/4	3 5/8	2 7/8	1 5/8	1 3/16	1 9/16	8	1	3 3/8	3 5/8	2 7/8	1 5/8	1 3/16	1 9/16	9 1/8	1	4 1/2	3 5/8	4
4.9–6.0	5.0– 6.4	6.687	2 3/8	1 3/16	1 9/16	8 1/16	2 1/8	2 1/4	3 11/16	3 3/16	2 3/8	1 3/16	1 9/16	8 1/16	1	3 3/8	3 11/16	3 3/16	2 3/8	1 3/16	1 9/16	9 5/16	1	4 1/2	3 11/16	4 5/16
5.4–6.5	5.5– 6.9	7.187	2 3/8	1 3/16	1 9/16	8 1/16	2 1/8	2 1/4	3 11/16	3 3/16	2 3/8	1 3/16	1 9/16	8 1/16	1	3 3/8	3 11/16	3 3/16	2 3/8	1 3/16	1 9/16	9 5/16	1	4 1/2	3 11/16	4 5/16
5.9–7.0	6.0– 7.4	7.687	2 3/8	1 3/16	1 9/16	8 1/16	2 1/8	2 1/4	3 11/16	3 3/16	2 3/8	1 3/16	1 9/16	8 1/16	1	3 3/8	3 11/16	3 3/16	2 3/8	1 3/16	1 9/16	9 5/16	1	4 1/2	3 11/16	4 5/16
6.9–8.0	7.0– 8.4	8.687	2 3/8	1 3/16	1 9/16	8 1/16	2 1/8	2 1/4	3 11/16	3 3/16	2 3/8	1 3/16	1 9/16	8 1/16	1	3 3/8	3 11/16	3 3/16	2 3/8	1 3/16	1 9/16	9 5/16	1	4 1/2	3 11/16	4 5/16
7.9–9.0	8.0– 9.4	9.687	2 3/8	1 3/16	1 9/16	8 1/16	2 1/8	2 1/4	3 11/16	3 3/16	2 3/8	1 3/16	1 9/16	8 1/16	1	3 3/8	3 11/16	3 3/16	2 3/8	1 3/16	1 9/16	9 5/16	1	4 1/2	3 11/16	4 5/16
	9.0–10.4	10.687	3 1/4	1 5/8	2	9 13/16	2 5/8	2 1/4	4 15/16	3 3/8	3 1/4	1 5/8	2	9 13/16	1 1/2	3 3/8	4 15/16	3 3/8	3 1/4	1 5/8	2	10 15/16	1 1/2	4 1/2	4 15/16	4 1/2
	10.0–11.4	11.687	3 1/4	1 5/8	2	9 13/16	2 5/8	2 1/4	4 15/16	3 3/8	3 1/4	1 5/8	2	9 13/16	1 1/2	3 3/8	4 15/16	3 3/8	3 1/4	1 5/8	2	10 15/16	1 1/2	4 1/2	4 15/16	4 1/2
	11.0–12.4	12.687	3 1/4	1 5/8	2	9 13/16	2 5/8	2 1/4	4 15/16	3 3/8	3 1/4	1 5/8	2	9 13/16	1 1/2	3 3/8	4 15/16	3 3/8	3 1/4	1 5/8	2	10 15/16	1 1/2	4 1/2	4 15/16	4 1/2

Range of pitch diam		Outside diam	5 grooves								6 grooves							
A	B		Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z
				Min	Max							Min	Max					
3.4–4.5	3.5– 4.9	5.187	1 1/4	1 3/16	1 9/16	10 1/4	1	5 5/8	3 5/8	5 1/8	1 1/4	1 3/16	1 9/16	11 5/8	1	6 3/4	3 5/8	6 1/4
3.9–5.0	4 0– 5.4	5.687	1 5/8	1 3/16	1 9/16	10 1/4	1	5 5/8	3 5/8	5 1/8	1 5/8	1 3/16	1 9/16	11 5/8	1	6 3/4	3 5/8	6 1/4
4.4–5.5	4.5– 5.9	6.187	1 5/8	1 3/16	1 9/16	10 1/4	1	5 5/8	3 5/8	5 1/8	1 5/8	1 3/16	1 9/16	11 5/8	1	6 3/4	3 5/8	6 1/4
4.9–6.0	5 0– 6.4	6 687	2 3/8	1 3/16	1 9/16	10 1/2	1	5 5/8	3 7/8	4 7/8	2 3/8	1 3/16	1 9/16	11 5/8	1	6 3/4	3 7/8	6
5.4–6.5	5.5– 6.9	7.187	2 3/8	1 3/16	1 9/16	10 1/2	1	5 5/8	3 7/8	4 7/8	2 3/8	1 3/16	1 9/16	11 5/8	1	6 3/4	3 7/8	6
5.9–7.0	6 0– 7.4	7.687	2 3/8	1 3/16	1 9/16	10 1/2	1	5 5/8	3 7/8	4 7/8	2 3/8	1 3/16	1 9/16	11 5/8	1	6 3/4	3 7/8	6
6.9–8.0	7.0– 8.4	8.687	2 3/8	1 3/16	1 9/16	10 1/2	1	5 5/8	3 7/8	4 7/8	2 3/8	1 3/16	1 9/16	11 5/8	1	6 3/4	3 7/8	6
7.9–9.0	8.0– 9.4	9.687	2 3/8	1 3/16	1 9/16	10 1/2	1	5 5/8	3 7/8	4 7/8	2 3/8	1 3/16	1 9/16	11 5/8	1	6 3/4	3 7/8	6
	9.0–10.4	10.687	3 1/4	1 5/8	2	12 1/16	1 1/2	5 5/8	4 15/16	5 5/8	3 1/4	1 5/8	2	13 3/16	1 1/2	6 3/4	4 15/16	5 1/4
	10.0–11 4	11.687	3 1/4	1 5/8	2	12 1/16	1 1/2	5 5/8	4 15/16	5 5/8	3 1/4	1 5/8	2	13 3/16	1 1/2	6 3/4	4 15/16	5 1/4
	11.0–12.4	12.687	3 1/4	1 5/8	2	12 1/16	1 1/2	5 5/8	4 15/16	5 5/8	3 1/4	1 5/8	2	13 3/16	1 1/2	6 3/4	4 15/16	5 1/4

Table 14-63. General Dimensions of Stub-shaft Type Motion-control Vari-Pitch Sheaves for A- or B-section Texrope Belts* (*Cont.*)

Range of pitch diam		Outside diam	7 grooves								8 grooves							
A	B		Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z
				Min	Max							Min	Max					
3.4–4.5	3.5– 4.9	5.187	1 1/4	1 3/16	1 9/16	12 1/2	1	7 7/8	3 5/8	7 3/8	1 1/4	1 3/16	1 9/16	13 5/8	1	9	3 5/8	8 1/2
3.9–5.0	4.0– 5.4	5.687	1 5/8	1 3/16	1 9/16	12 1/2	1	7 7/8	3 5/8	7 3/8	1 5/8	1 3/16	1 9/16	13 5/8	1	9	3 5/8	8 1/2
4.4–5.5	4.5– 5.9	6.187	1 5/8	1 3/16	1 9/16	12 1/2	1	7 7/8	3 5/8	7 3/8	1 5/8	1 3/16	1 9/16	13 5/8	1	9	3 5/8	8 1/2
4.9–6.0	5.0– 6.4	6.687	2 3/8	1 3/16	1 9/16	12 3/4	1	7 7/8	3 7/8	7 1/8	2 3/8	1 3/16	1 9/16	13 7/8	1	9	3 7/8	8 1/4
5.4–6.5	5.5– 6.9	7.187	2 3/8	1 3/16	1 9/16	12 3/4	1	7 7/8	3 7/8	7 1/8	2 3/8	1 3/16	1 9/16	13 7/8	1	9	3 7/8	8 1/4
5.9–7.0	6.0– 7.4	7.687	2 3/8	1 3/16	1 9/16	12 3/4	1	7 7/8	3 7/8	7 1/8	2 3/8	1 3/16	1 9/16	13 7/8	1	9	3 7/8	8 1/4
6.9–8.0	7.0– 8.4	8.687	2 3/8	1 3/16	1 9/16	12 3/4	1	7 7/8	3 7/8	7 1/8	2 3/8	1 3/16	1 9/16	13 7/8	1	9	3 7/8	8 1/4
7.9–9.0	8.0– 9.4	9.687	2 3/8	1 3/16	1 9/16	12 3/4	1	7 7/8	3 7/8	7 1/8	2 3/8	1 3/16	1 9/16	13 7/8	1	9	3 7/8	8 1/4
	9.0–10.4	10.687	3 1/4	1 5/8	2	14 5/16	1 1/2	7 7/8	4 15/16	6 3/8	3 1/4	1 5/8	2	15 7/16	1 1/2	9	4 15/16	7 1/2
	10.0–11.4	11.687	3 1/4	1 5/8	2	14 5/16	1 1/2	7 7/8	4 15/16	6 3/8	3 1/4	1 5/8	2	15 7/16	1 1/2	9	4 15/16	7 1/2
	11.0–12.4	12.687	3 1/4	1 5/8	2	14 5/16	1 1/2	7 7/8	4 15/16	6 3/8	3 1/4	1 5/8	2	15 7/16	1 1/2	9	4 15/16	7 1/2

* Courtesy Allis-Chalmers Manufacturing Co.

† Number of turns of adjusting screw to make a complete change in pitch diameter: A section, 1 7/8 turns; B section, 2 1/4 turns.

Table 14-64. Motion-control Vari-Pitch Sheaves for Texrope Belts*

C-section Texrope belts

All dimensions are given in inches.

Range of pitch diam†	Outside diam	2 grooves								3 grooves								4 grooves								5 grooves							
		Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z
			Min	Max							Min	Max							Min	Max							Min	Max					
8 5–10 7	11 063	3 1/4	1 13/32	2	11 1/8	3 1/32	3 9/32	4 13/16	4 11/16	3 1/4	1 13/32	2	11 1/8	1 3/8	4 15/16	4 13/16	4 11/16	3 1/4	1 13/32	2	12 25/32	1 3/8	6 19/32	4 13/16	6 3/8	3 1/4	1 13/32	2	14 7/16	1 3/8	8 1/4	4 13/16	6 17/32
9 5–11 7	12 063	3 1/4	1 13/32	2	11 1/8	3 1/32	3 9/32	4 13/16	4 11/16	3 1/4	1 13/32	2	11 1/8	1 3/8	4 15/16	4 13/16	4 11/16	3 1/4	1 13/32	2	12 25/32	1 3/8	6 19/32	4 13/16	6 3/8	3 1/4	1 13/32	2	14 7/16	1 3/8	8 1/4	4 13/16	6 17/32
10 5–12 7	13 063	3 1/4	1 13/32	2	11 1/8	3 1/32	3 9/32	4 13/16	4 11/16	3 1/4	1 13/32	2	11 1/8	1 3/8	4 15/16	4 13/16	4 11/16	3 1/4	1 13/32	2	12 25/32	1 3/8	6 19/32	4 13/16	6 3/8	3 1/4	1 13/32	2	14 7/16	1 3/8	8 1/4	4 13/16	6 17/32
11 5–13 7	14 063	3 1/4	1 13/32	2	11 1/8	3 1/32	3 9/32	4 13/16	4 11/16	3 1/4	1 13/32	2	11 1/8	1 3/8	4 15/16	4 13/16	4 11/16	3 1/4	1 13/32	2	12 25/32	1 3/8	6 19/32	4 13/16	6 3/8	3 1/4	1 13/32	2	14 7/16	1 3/8	8 1/4	4 13/16	6 17/32
12 5–14 7	15 063	3 1/4	1 13/32	2	11 1/8	3 1/32	3 9/32	4 13/16	4 11/16	3 1/4	1 13/32	2	11 1/8	1 3/8	4 15/16	4 13/16	4 11/16	3 1/4	1 13/32	2	12 25/32	1 3/8	6 19/32	4 13/16	6 3/8	3 1/4	1 13/32	2	14 7/16	1 3/8	8 1/4	4 13/16	6 17/32
13 5–15 7	16 063	3 1/4	1 13/32	2	11 1/8	3 1/32	3 9/32	4 13/16	4 11/16	3 1/4	1 13/32	2	11 1/8	1 3/8	4 15/16	4 13/16	4 11/16	3 1/4	1 13/32	2	12 25/32	1 3/8	6 19/32	4 13/16	6 3/8	3 1/4	1 13/32	2	14 7/16	1 3/8	8 1/4	4 13/16	6 17/32

Range of pitch diam	Outside diam	6 grooves								7 grooves								8 grooves							
		Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z
			Min	Max							Min	Max							Min	Max					
8 5–10 7	11 063	3 1/4	1 13/32	2	16 3/32	1 3/8	9 29/32	4 13/16	8 3/16	3 1/4	1 13/32	2	17 3/4	1 3/8	11 9/16	4 13/16	9 27/32	3 1/4	1 13/32	2	19 13/32	1 3/8	17 7/32	4 13/16	11 1/2
9 5–11 7	12 063	3 1/4	1 13/32	2	16 3/32	1 3/8	9 29/32	4 13/16	8 3/16	3 1/4	1 13/32	2	17 3/4	1 3/8	11 9/16	4 13/16	9 27/32	3 1/4	1 13/32	2	19 13/32	1 3/8	17 7/32	4 13/16	11 1/2
10 5–12 7	13 063	3 1/4	1 13/32	2	16 3/32	1 3/8	9 29/32	4 13/16	8 3/16	3 1/4	1 13/32	2	17 3/4	1 3/8	11 9/16	4 13/16	9 27/32	3 1/4	1 13/32	2	19 13/32	1 3/8	17 7/32	4 13/16	11 1/2
11 5–13 7	14 063	3 1/4	1 13/32	2	16 3/32	1 3/8	9 29/32	4 13/16	8 3/16	3 1/4	1 13/32	2	17 3/4	1 3/8	11 9/16	4 13/16	9 27/32	3 1/4	1 13/32	2	19 13/32	1 3/8	17 7/32	4 13/16	11 1/2
12 5–14 7	15 063	3 1/4	1 13/32	2	16 3/32	1 3/8	9 29/32	4 13/16	8 3/16	3 1/4	1 13/32	2	17 3/4	1 3/8	11 9/16	4 13/16	9 27/32	3 1/4	1 13/32	2	19 13/32	1 3/8	17 7/32	4 13/16	11 1/2
13 5–15 7	16 063	3 1/4	1 13/32	2	16 3/32	1 3/8	9 29/32	4 13/16	8 3/16	3 1/4	1 13/32	2	17 3/4	1 3/8	11 9/16	4 13/16	9 27/32	3 1/4	1 13/32	2	19 13/32	1 3/8	17 7/32	4 13/16	11 1/2

Table 14-64. Motion-control Vari-Pitch Sheaves for Texrope Belts* (*Cont.*)

D-section Texrope belts

Range of pitch diam†	Outside diam	3 grooves						4 grooves						5 grooves						6 grooves					
		Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z
12 0–15 0	15 625	4 1/2	13 3/16	2	6 7/8	4 5/16	7	4 1/2	15 1/2	2	9 3/16	4 5/16	7 1/2	4 1/2	17 3/16	2	11 1/2	4 5/16	8	4 1/2	20 1/8	2	13 13/16	4 5/16	8 1/2
13.0–16.0	16 625	4 1/2	13 3/16	2	6 7/8	4 5/16	7	4 1/2	15 1/2	2	9 3/16	4 5/16	7 1/2	4 1/2	17 3/16	2	11 1/2	4 5/16	8	4 1/2	20 1/8	2	13 13/16	4 5/16	8 1/2
14.0–17.0	17.625	4 1/2	13 3/16	2	6 7/8	4 5/16	7	4 1/2	15 1/2	2	9 3/16	4 5/16	7 1/2	4 1/2	17 3/16	2	11 1/2	4 5/16	8	4 1/2	20 1/8	2	13 13/16	4 5/16	8 1/2

Range of pitch diam	Outside diam	7 grooves						8 grooves						9 grooves						10 grooves					
		Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z	Max bore	V	W	X	Y	Z
12.0–15.0	15.625	4 1/2	22 7/16	2	16 1/8	4 5/16	9	4 1/2	24 3/4	2	18 7/16	4 5/16	10	4 1/2	27 1/16	2	20 3/4	4 5/16	11	4 1/2	29 3/8	2	23 1/16	4 5/16	12
13 0–16.0	16 625	4 1/2	22 7/16	2	16 1/8	4 5/16	9	4 1/2	24 3/4	2	18 7/16	4 5/16	10	4 1/2	27 1/16	2	20 3/4	4 5/16	11	4 1/2	29 3/8	2	23 1/16	4 5/16	12
14.0–17.0	17 625	4 1/2	22 7/16	2	16 1/8	4 5/16	9	4 1/2	24 3/4	2	18 7/16	4 5/16	10	4 1/2	27 1/16	2	20 3/4	4 5/16	11	4 1/2	29 3/8	2	23 1/16	4 5/16	12

E-section texrope belts

Range of pitch diam§	Outside diam	5 grooves								6 grooves								7 grooves								8 grooves							
		Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z	Max bore	T		V	W	X	Y	Z
			Min	Max							Min	Max							Min	Max							Min	Max					
20.0–23 75	24.5	4 1/2	1 3/4	2 3/4	23 5/8	2 5/8	14 15/16	6 1/16	13	4 1/2	1 3/4	2 3/4	26 5/8	2 5/8	17 15/16	6 1/16	16	4 1/2	1 3/4	2 3/4	29 5/8	2 5/8	20 15/16	6 1/16	19	4 1/2	1 3/4	2 3/4	32 5/8	2 5/8	23 15/16	6 1/16	22

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* Courtesy Allis-Chalmers Manufacturing Co.

† Number of turns of adjusting screw to make a complete change in pitch diameter, 3 9/16 turns.

‡ Number of turns of adjusting screw to make a complete change in pitch diameter, 4 7/8 turns.

§ Number of turns of adjusting screw to make a complete change in pitch diameter, 6 turns.

Table 14-65. Face Width of Companion Sheaves Used with Vari-Pitch Sheaves

Size of belt	Number of grooves								
	2	3	4	5	6	7	8	9	10
A	1 11/16	2 5/8	3 9/16	4 1/2	5 7/16				
B	2 1/8	3 1/4	4 3/8	5 1/2	6 5/8	7 3/4	8 7/8		
C	2 13/16	4 1/4	5 11/16	7 1/8	8 9/16	10	11 7/16		
D		6 3/8	8 11/16	11	13 5/16	15 5/8	17 15/16		
E				14	16 15/16	19 7/8	22 13/16	25 3/4	28 11/16

PIVOTED AND SPRING AUTOMATIC MOTOR-BASE DRIVES*

BY SAMUEL L. ALLEN

Design Engineer, J. E. Rhoads and Sons

The pivoted motor-base drive is a carefully engineered power-transmission unit adopted by industry for the efficient operation of short center belt drive installations. It utilizes the simplest form of power transmission: a motor, two pulleys, a strong flexible belt, and a pivoted motor base. The motor is mounted on adjustable base arms which are free to rotate about a horizontally mounted pivot shaft (Fig. 14-14*a*) or ball-and-socket bronze bearing (Fig. 14-14*b*).

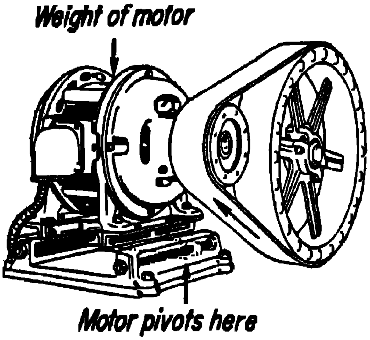

FIG. 14-14*a*. Motor mounted on pivoted motor base.

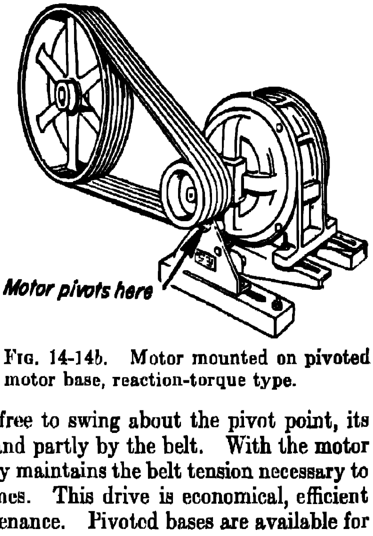

FIG. 14-14*b*. Motor mounted on pivoted motor base, reaction-torque type.

The motor is bolted to the arms and is free to swing about the pivot point, its weight being supported partly by the pivot and partly by the belt. With the motor correctly placed on the arms, it automatically maintains the belt tension necessary to ensure a continuous flow of power at all times. This drive is economical, efficient dependable, and requires little, if any, maintenance. Pivoted bases are available for practically every size of motor, and they can be installed to operate vertically, horizontally at any angle, mounted on the ceiling, on the floor, on the wall, or mounted on the machine itself.

Pivoted bases are adjustable devices which can be used on many types of installations almost too numerous to mention. Some of the more common uses are on fans, compressors, machine tools, pumps, generators, industrial line shafts, textile spinning frames, beaters, etc.

CLASSES OF BASES

There are two distinct classes of bases: One is the Rockwood base which depends mainly upon motor weight to supply tension to the belt, acting through an adjustable lever arm. The other is the American Econ-o-matic base with the pivot point close to the motor shaft. This base utilizes mainly the twisting or rotating action of the stator which is equal and opposite to the rotor action. The motor weight is offset enough to give initial tension, and the load determines the tension in the belt automatically. The underlying principles of these drives are the same, but motor weight is more important in one type than the other. These two bases, properly applied and designed, control belt tension accurately, making short center-belt drives practical and economical. These two general classes are suitable for floor, ceiling, wall, and vertical drives. Three special textile bases for spinning and twister frames are available for motors up to 20 hp at 1,800 rpm.

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* The data and tables for this subsection are by the courtesy of J. E. Rhoads and Sons, American Pulley Co., Automatic Motor Base Co., and Rockwood Manufacturing Co.

Figure 14-15 shows the different types of Rockwood bases and their range of application.

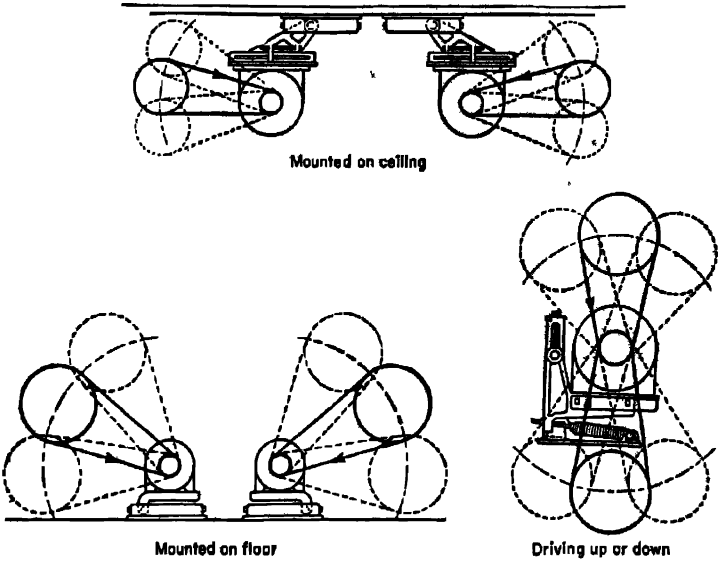

FIG. 14-15. Range of application for Rockwood bases.

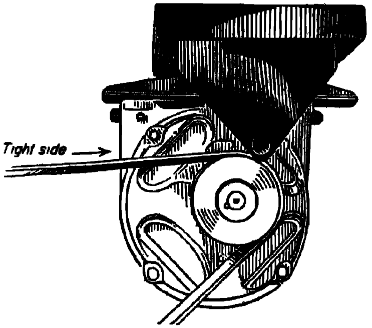

FIG. 14-16. Ceiling-drive type, American base.

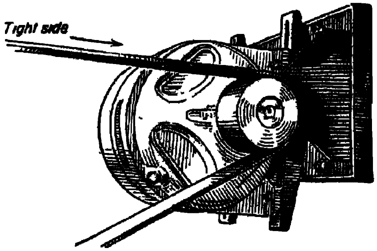

FIG. 14-17 Wall-drive type, American base.

Figures 14-16, 14-17, 14-18, 14-19, and 14-20 show the different types of American bases. On all drives, the tight side of the belt must be near the pivot. This rotation gives increased tension with increased load and decreased tension with decreased load. The opposite rotation gives just the reverse of this, and some drives may be inoperative. In this case, the drive should be changed or a different base used so that the load pull comes near the pivot.

Ceiling-type Bases. Frequently ceiling drives constitute a trouble zone for plant maintenance men. Belts stretch, but because of their inaccessibility, they are not given proper attention, and production is slowed down owing to excess slip. A pivoted motor base on the ceiling will correct this.

The floor-type American base is used on the ceiling with special emphasis on the belt direction. The tight side of the belt must approach the face of the motor pulley closest to the pivot of the base (see ceiling installation, Fig. 14-16).

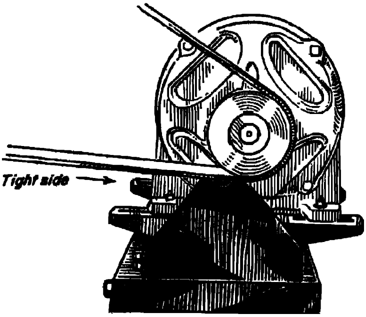

FIG. 14-18. Floor-drive type, American base.

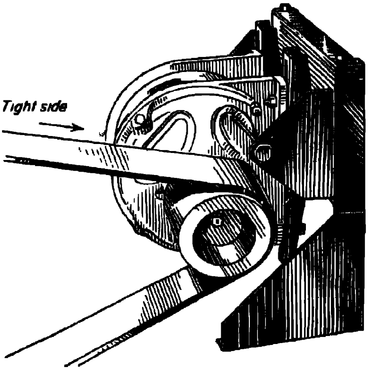

FIG. 14-19. Topside-drive type, American base.

The Rockwood base for use on ceiling drives has special hanger arms and auxiliary adjusting screws for changing tension, if necessary. All other adjustments are the same as in the floor base.

Vertical-type Bases. Fixed-center vertical-motor drives cause trouble sooner or later because of belt stretch resulting in slip and loss in production. The special vertical-type Rockwood base eliminates all this trouble. With the motor above, the springs support the entire motor weight and supply belt tension. With the motor below, the springs take off the motor weight not needed for belt tension.

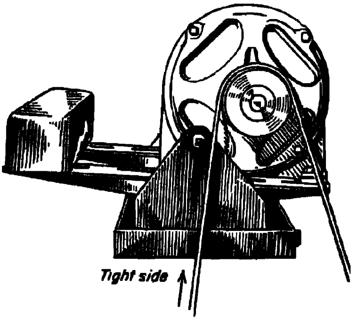

FIG. 14-20. Down-drive type, American base.

The American down-drive base (Fig. 14-20) mounted either on suitable brackets or platform keeps the belt tight with the motor above. The topside base (Fig. 14-19) is used when the center line of the drive is within 45 deg of the vertical. Both types have all the adjustable features found in the regular floor-type bases.

Construction of Rockwood Bases. On a standard base, the motor is bolted rigidly to a pair of steel arms instead of to the usual slide-rail motor base. These arms are pivoted on a steel shaft which is in turn mounted on adjustable screws running the length of the housing. Therefore, the entire pivot shaft can be moved forward or back to accommodate an endless belt length by turning the adjusting screws. The arms are movable parallel to the pivot shaft to accommodate different motor-frame sizes and for alignment. These arms are provided with slots on the sides so that the motor bolted to the arms can be moved toward the pivot for less tension or away from it for more tension, giving a wide range of belt tensions, the final position being determined by the load to be transmitted. The large take-up provided is sufficient in most cases to take care of speed changes by using pulleys of different diameter.

If the base is not quite parallel to the driven shaft after installation, the pivot shaft can in most cases be made parallel by turning one of the long adjusting screws. This can be done while the drive is running. Since the pivot shaft is below the motor feet, the end bells on the motor can be removed without removing the motor from the base. The many adjustments provide interchangeability of motors in case of failure or in many cases make it possible to install a larger motor on the same base without moving the base on its foundation. Figure 14-21*a* shows a typical pivoted motor-base flat-belt drive of the Rockwood floor-type base driving a fan.

Construction of the American Base. The American Econ-o-matic motor base is made of steel throughout—steel because it combines strength with lightness. Steel also has a natural elasticity which relieves and neutralizes any destructive forces that may be set up within the motor because of imperfect balance of the motor, pulleys, or belt. Resilient steel absorbs noise-producing vibrations and motor hum, thus taking the place of an expensive type of noiseproof foundation. The base itself is perfectly quiet in operation, the finished assembly permits no looseness or rattle.

The only moving part is the "cradle" made of cast- or forged-steel cranks, solid-steel cross rods, and pressed-steel motor-base arms. This cradle is pivoted in bronze bearings* of the ball-and-socket self-aligning type. These bearings ensure free movement in any position and eliminate all danger of "freezing" or "binding."

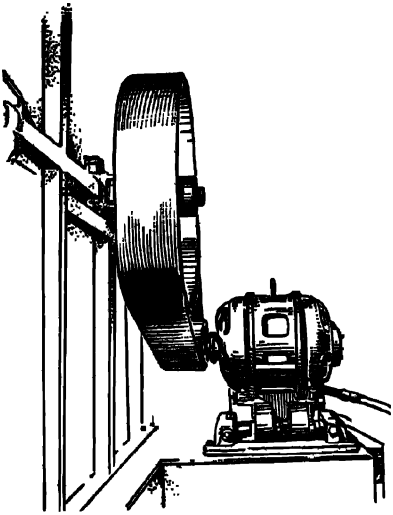

FIG. 14-21*a*. Typical floor-type Rockwood-base installation driving an air-conditioning fan.

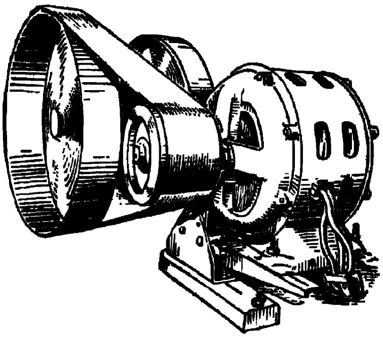

FIG. 14-21*b*. Floor-type American-base installation driving compressor.

Standard bases are available in seven sizes; duplex-type bases in nine sizes; topside bases in five sizes; and there is also a full line of textile bases for spinning and twisting frames. Other bases can be built to special dimensions when required.

It is a simple matter to adapt the base to motors of various dimensions. Liberal adjustments, easily accessible, have been provided in the foundation plates, in the bearing standards, and in the motor-base arms. Any adjustments necessary for the correction of belt alignment or for the take-up of belt stretch can be made while the drive is in operation.

Figure 14-21*b* shows a floor-type installation using a special-tannage leather belt on an American base driving a compressor.

BELT ELONGATION

Belt elongation occurs from centrifugal tension, too much tension, high humidity, heavy loads, too small a belt, and permanent stretch. Stretch allows the motor to tilt on these variable-center drives, but ring-oiling bearings leave a permissible tilt of −7 deg from the horizontal before the arms have to be leveled up by turning the screws provided. Ball-bearing motors can tilt any amount as long as the arms do not bottom on the base. For the sake of appearance, the feet should be kept horizontal.

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* With the exception of A00 and A0.

THE BELT

While any type of belt can be used on the pivoted motor drive, certain belts are better than others. Special-tannage, endless leather belts are very flexible, have the highest grip on the pulleys, can be made endless any desired length, and on the average last longer. Cord belts, V belts, chrome-tanned leather belts, oak belts, special-tannage leather belts, rubber belts, steel-cable flat belts, and others have all been used successfully.

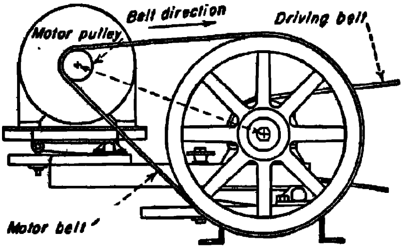


Fig. 14-21c. Double-reduction Rockwood-base drive.

Steel-cable V belts can be operated at 6,000 fpm or higher, but the manufacturer should be consulted on these drives.

The steel-cable flat transmission belts require special engineering, and the manufacturer should be consulted regarding these belts and pulley crowns. High-speed drives up to 10,000 fpm are possible with this steel-cable flat belt.

While any type of pulley can be used, the pulleys presenting the greatest resistance to slip should be used. The motor pulley should be paper or wood, and the driven pulley as well, when the pulley ratio is near unity. Large driven pulleys may be cast iron, pressed steel, or wood. V sheaves should be used with V-belt drives.

All pulleys selected should be large enough for the job to be done. Table 14-66 shows NEMA standard minimum motor-pulley diameters for a given motor size. Motor pulleys smaller than those recommended by NEMA introduce high belt tensions and bearing pressures; therefore, they should be avoided where possible. Belt speeds between 2,500 and 4,500 fpm are ideal. Speeds up to 6,000 fpm and higher are permissible under certain conditions but are not generally recommended. The exception is the steel-cable construction referred to above.

Pulley ratios up to 8:1 are permissible, and ratios up to 60:1 on the double-reduction gravity-base drive are practical, see Fig. 14-21c.

MOTOR FLOAT OR VIBRATION

With the motor free to follow the load pulsations, the motor must move in unison with these pulsations. The amount of vibration or motion depends upon several factors. If the load is steady, the float may hardly be noticeable. Drives on violently fluctuating loads may need snubbers to keep the drive steady. A base too small for the motor may cause vibration, resulting in breakage of some part of the base. Uneven belt thickness, eccentric pulley, and uneven load are all causes of motor vibration. Occasionally, a motor rotor may be out of balance causing undue vibration. With the motor free to follow the load pulsations, the motor, machine, and bearings are freed of these harmful load shocks, and longer machine, bearing, and belt life results.

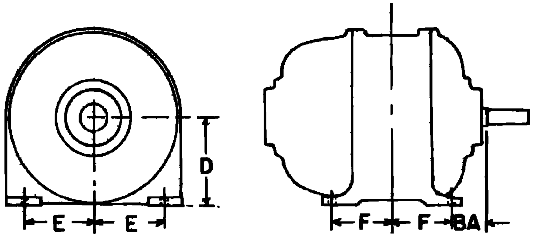
Pulley Centers

Pulley centers may be as short as the diameter of the large pulley, although preferably a little longer. Some drives have very short centers with the pulley faces just missing each other. These ultrashort centers are found mainly in the textile industry on spinning- and twister-frame drives where the space is very limited and the speed ratio is about 2.5:1 or less with a range of 5 to 20 hp. Large horsepower drives may require long centers depending upon circumstances. Most of these drives will come within 12 to 14 ft.

Efficiency

The efficiency of the pivoted drive varies somewhat with the belt, but it reaches a peak around 99 per cent which it closely approaches over a wide range in tensions. Pivoted drives should be designed to handle the maximum capacity of the motor, which for most types of motors is at least 150 to 250 per cent of their rated capacity. However, motors usually operate at less than their rated capacity, so that the drive is operating most of the time at less than one-half its peak capacity. This load characteristic makes important the ability of the drive to maintain its high efficiency under fractional load (see horsepower slip curves and horsepower efficiency, Fig. 14-22).

Table 14-66. Recommended Minimum Pulley Diameters for Pivoted Motor-base Drives



NEMA frame size	Min pulley diam, in.	Floor-base size, Rockwood	*D*	*E*	*F*	*BA*
203	3	0	5	4	2¾	3⅛
204	3	0	5	4	3¼	3⅛
224	4	0	5½	4½	3⅝	3½
225	4	2	5½	4½	4¾	3½
254	4½	2	6¼	5	4⅛	4¼
284	5	4	7	5½	4¾	4¾
324	6	4	8	6¼	5¼	5¼
326	8	6	8	6¼	6	5¼
364	9	6	9	7	5⅝	5⅞
365	9	6	9	7	6⅛	5⅞
404	10	8	10	8	6⅛	6⅝
405	10	8	10	8	6⅞	6⅝
444	12	10	11	9	7¼	7½
445	12	10	11	9	8¼	7½
504	12	12	12½	10	8	8½
505	15	12	12½	12½	9	8½

$F + BA + \frac{1}{2}$ pulley face width $= S$.
S = center of motor and base to center of motor pulley.

GRAPHICAL METHOD FOR DETERMINING PIVOT CENTER DISTANCE

Suggestions for Layout. Select the pulley diameters commensurate with the horsepower and load to be transmitted. Suggested minimum motor pulleys are shown in Table 14-66. Figure the driven pulley diameter D by the following method:

EXAMPLE:

Let
motor hp = 50
Motor rpm N = 860 full-load speed
Motor pulley diameter d = 14 in.
Fan rpm n = 167

Then

$$D = dN/n = (14 \times 860)/167 = 72\text{-in. diameter driven pulley}$$

If the fan speed must be figured more accurately, add the thickness of the belt to each pulley diameter, and deduct 1 per cent from the driven speed for slip and creep losses. Accurate pulley sizes can be figured by the same method; but add 1 per cent of driven speed to the speed wanted, and deduct the belt thickness from the pulley diameters.

Direction of Rotation. It is desirable on floor, wall, and ceiling drives to have the tight strand near the pivot shaft, although a few ceiling drives will operate better with the tight side away from the pivot. This can be controlled by the position of the motor as well as the direction of rotation of the driven shaft.

Center Distance. The center distance may be as short as the large pulley diameter (as long as the pulley faces do not touch) although preferably a little longer.

Slope of Center Line between Pulleys. The most favorable position of the motor is with its center on a level with the driven pulley center, although few drives have horizontal centers. The motor center is usually below the driven shaft, or it can be slightly above.

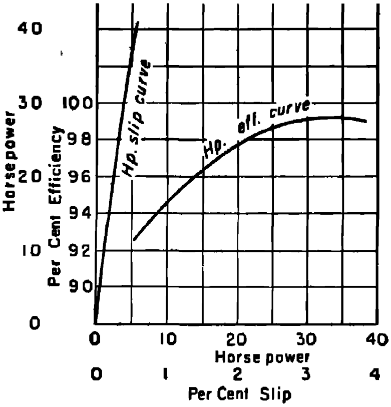


FIG. 14-22. Efficiency and slip chart.

Solution for Pivot Center Distance

Having selected the motor, pulley diameters, center distance, and position of the base to give the load pull near the pivot, the proper setting of the base can be determined. First, calculate the belt tensions at the peak load. The second step is to determine the offset of the motor needed to balance the resultant belt tension.

Nomenclature

In the following calculations, a slide rule can be used:

Initial Data:

D = diameter of driven pulley, in.
d = diameter of motor pulley, in.
C = distance between pulley centers, in.
H = height of center of driven pulley above the bottom of the Rockwood base, in.
P = peak load to be transmitted, hp[*]
N = speed of motor at peak load, rpm
n = speed of driven pulley, rpm
W = weight of motor, lb

Data to Be Determined:

V = belt speed, fpm
T_1 = tension in tight side of belt, lb
T_2 = tension in slack side of belt, lb
G = resultant belt tension, vector sum of T_1 and T_2, lb
$T_e = T_1 - T_2$ or effective tension, lb
e = pivot center distance (horizontal distance between motor center and pivot shaft, in.)
K = see Table 14-69
$\pi = 3.14$

Calculate:

$$V = d\pi N/12 \tag{1}$$

$$T_e = 33{,}000P/V \tag{2}$$

$$(D - d)/C \tag{3}$$

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[*] Safe values of peak load for line-start motors may be taken as 2.5 times the name-plate rating; compensator-started motors as 1.5 times name-plate rating. These values for peak load may be adjusted according to the frequency of occurrence or the inertia and character of the load.

From Table 14-69, take the corresponding value of K for the type of belt and small pulley to be used.

$$(4) \qquad T_1 = T_e K \qquad T_2 = T_1 - T_e$$

Carry through the following operations:

1. From the motor-dimension sheet, select the appropriate base size or from Table 14-66, 14-67, 14-68 or 14-72. For motors not listed, consult your belting supplier.
2. Lay out the side view of the drive to any convenient scale. First, draw the floor, wall, or ceiling line on which the base is mounted. Then include any post, pipes, etc., which limit the location of the drive. Finally, place the pulleys in their positions to assure getting the correct angle of drive. Extend the lines of the belt strands to their point of intersection.*
3. From the point of intersection B, lay off T_1 to any convenient scale on the engineer's rule along the tight strand. At C, lay off T_2 parallel to the slack strand using same scale as above. Draw and scale the resultant belt tension $G = BG$.
4. Draw a vertical line through the center of the motor, and from its intersection A with the resultant belt tension, lay off G on its extension using the same scale as in (3) or to a smaller scale if more convenient. $AX = BG$, only a smaller scale is used.
5. At X, lay off the motor weight W vertically downward using the same scale as in (4). Draw the diagonal from W to A.
6. Draw a horizontal line at the level of the pivot. Its intersection Q with the diagonal AW locates the desired horizontal position of the pivot shaft. The horizontal position of the base with relation to the pivot shaft can be determined from the dimensions shown in Tables 14-71, 14-72, or 14-73. The work may be checked by measuring e and P. If correct, We equals GP. P is perpendicular distance from Q to the resultant BG.

Table 14-67. Rockwood-base Selection Tables

Base No.	Motor horsepower and rpm based on NEMA frame sizes			Heaviest motor for this base, lb	Shipping wt, lb
	1,800	1,200	900		
0	1	3/4	1/2	150	35
	2	1 1/2	1		
2	3	2	1 1/2	250	55
	5	3	2		
4	7 1/2	5	3	450	80
	10	7 1/2	5		
6	15	10	7 1/2	650	115
	20	15	10		
8	25	20	15	900	160
	30	25	20		
10	40	30	25	1,500	240
	50	40	30		
12	60	50	40	2,500	390
	75	60	50		

Use the 14-series bases for motors up to 175 hp. Use the 16-series bases for motors up to 250 hp.

The choice of base for a given motor is made with reference to the maximum bolthole spacing (Tables 14-71 and 14-72) and weight limitation of the base. If your motor weight is more than allowed for a given base, select the base from the maximum-weight column.

7. If it is found that the required overhang of the motor is greater than the standard base permits, the drive is not practical as laid out. By applying one or more of the corrective measures listed on page 1035, almost all drives can be accommodated.

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* When $(D - d)/C$ is less than 0.200, the intersection of the strands is likely to fall off the paper. [illegible] pproxi- [illegible] he belt [illegible] pulley in proportion to T_1 and T_2 with the point of division nearer T_1. Draw [illegible] through the points of division.

Table 14-68. American-base Selection Tables*

Frame No.	Horsepower and rpm of motor based on NEMA frame sizes†			General-purpose motors		Standard bases		Vertical bases		Top-side-top bases		Approx shipping wt floor-type bases lb	Max motor wt recommended, lb	
	1,800	1,200	900	Open	Totally enclosed								Floor	Wall, ceiling and tight-side-top
203	1	¾-1	.....	0-K = 12	0-K = 12	00	K = 8⅛	V-00	K = 8⅛	....	....	10	50	50
204	1½	....	½	2-K = 14	2-K = 14	0	K = 12	V-0	K = 12	....	......	30	200	150
224	2	1½	¾	2-K = 14	2-K = 18									
225	3	2	1	2-K = 18	2-K = 18	2	K = 14, 18	V-2	K = 14 K = 18	T-2	K = 14	65	300	250
254	5	3	1½-2	2-K = 18	4-K = 20	4	K = 18, 20	V-4	K = 18 K = 20	T-4	K = 18	85	500	400
284	7½	5	3	4-K = 20	6-K = 22	6	K = 22, 24	V-6	K = 22 K = 24	T-6	K = 22	95	700	600
324	10	7½	5	6-K = 22	6-K = 24	8	K = 24, 28	V-8	K = 24 K = 28	T-8	K = 24	190	1,000	700
326	15	10	7½	6-K = 24	8-K = 28	10	K = 28, 32	V-10	K = 28 K = 32	T-10	K = 28	210	1,500	1,000
364	20	....	....	6-K = 24	8-K = 28	12	K = 36	...	....	....	....	290	2,500	
365	25	15	10	6-K = 24	8-K = 28	14	K = 40	....	....	....	......	320	3,000	
404	....	20	15	8-K = 28	10-K = 32	15	K = 40	....	....	...	....	500	3,500	
405	30	25	20	8-K = 28	10-K = 32	16	K = 44	..	....	...	...	875	4,000	
444	40	30	25	10-K = 28	12-K = 36	18	K = 48	...	....	...	...	910	4,500	
445	50	40	30	10-K = 32	12-K = 36									
504	60	50	40	12-K = 36	12-K = 36									
505	75	60	50	12-K = 36	14-K = 40									

* Courtesy of American Pulley Co.

† For selection of motor bases when using NEMA frame, general-purpose squirrel-cage induction motors only. Check your motor frame dimensions with dimensions K and V when using motors other than this type to make sure the motor will fit into the base selected. For motor frames larger than shown specify K and V dimension on order.

Table 14-69. Values of K Corresponding to Values of $(D - d)/C$ for Different Types of Belt and Pulley, $T_1 = (T_1 - T_2)K$

$(D - d)/C$	Special tannage leather			Oak leather			Rubber			Rubber V belt
	Paper	Wood	Cast iron	Paper	Wood	Cast iron	Paper	Wood	Cast iron	
0.00	1.180	1.262	1.400	1.262	1.364	1.548	1.400	1.548	1.740	1.33
0.05	1.186	1.270	1.409	1.270	1.374	1.563	1.409	1.563	1.760	1.34
0.10	1.192	1.279	1.420	1.279	1.384	1.578	1.420	1.578	1.781	1.365
0.15	1.199	1.288	1.432	1.288	1.395	1.594	1.432	1.594	1.803	1.378
0.20	1.206	1.297	1.445	1.297	1.407	1.610	1.445	1.610	1.825	1.39
0.25	1.213	1.306	1.459	1.303	1.420	1.627	1.459	1.627	1.848	1.402
0.30	1.221	1.316	1.473	1.316	1.434	1.645	1.473	1.645	1.871	1.415
0.35	1.230	1.327	1.488	1.327	1.449	1.663	1.488	1.663	1.896	1.428
0.40	1.239	1.338	1.504	1.338	1.464	1.681	1.504	1.681	1.920	1.44
0.45	1.248	1.350	1.520	1.350	1.480	1.700	1.520	1.700	1.945	1.455
0.50	1.258	1.363	1.537	1.363	1.496	1.720	1.537	1.720	1.971	1.47
0.55	1.268	1.377	1.554	1.377	1.513	1.742	1.554	1.742	1.998	1.485
0.60	1.279	1.392	1.572	1.392	1.530	1.765	1.572	1.765	2.025	1.503
0.65	1.290	1.407	1.590	1.407	1.548	1.790	1.590	1.790	2.053	1.52
0.70	1.302	1.423	1.610	1.425	1.567	1.815	1.610	1.815	2.081	1.54
0.75	1.315	1.439	1.632	1.439	1.587	1.842	1.632	1.842	2.113	1.56
0.80	1.329	1.456	1.655	1.456	1.608	1.870	1.655	1.870	2.147	1.58
0.85	1.344	1.475	1.680	1.475	1.630	1.900	1.680	1.900	2.182	1.604
0.90	1.360	1.495	1.707	1.495	1.654	1.932	1.707	1.932	2.218	1.625
0.95	1.377	1.517	1.735	1.517	1.679	1.965	1.735	1.965	2.256	1.65
1.00	1.396	1.540	1.765	1.540	1.705	2.000	1.765	2.000	2.295	1.708

Table 14-70. Values of M for Different Belt Speeds and Types of Belt, $T_m = MT_1$

Belt speed, fpm	Special tannage leather	Oak leather	Rubber	Belt speed, fpm	Special tannage leather	Oak leather	Rubber
0–600	1.00	1.00	1.00	3,600	1.16	1.18	1.20
800	1.00	1.01	1.01	3,800	1.18	1.20	1.23
1,000	1.01	1.01	1.01	4,000	1.20	1.23	1.27
1,200	1.01	1.02	1.02	4,200	1.22	1.27	1.31
1,400	1.02	1.02	1.03	4,400	1.25	1.30	1.35
1,600	1.02	1.03	1.04	4,600	1.28	1.33	1.39
1,800	1.03	1.04	1.05	4,800	1.32	1.37	1.44
2,000	1.04	1.05	1.06	5,000	1.35	1.42	1.49
2,200	1.05	1.06	1.07	5,200	1.39	1.47	1.55
2,400	1.06	1.07	1.08	5,400	1.44	1.53	1.63
2,600	1.08	1.09	1.10	5,600	1.48	1.59	1.71
2,800	1.09	1.10	1.12	5,800	1.54	1.66	1.79
3,000	1.10	1.12	1.14	6,000	1.60	1.74	1.90
3,200	1.12	1.14	1.16	6,200	1.67	1.83	2.04
3,400	1.14	1.16	1.18	6,400	1.75	1.94	2.20

a. Arrange the drive with the tight side of the belt running toward the pivot unless already done. (With some ceiling drives, more tension is obtained with the tight side away from the pivot.)
b. Increase pulley diameters and belt speed up to 6,000 fpm if possible.
c. Use a special-tannage belt and paper pulley, if not already done.
d. Raise the motor center up to 10 deg above horizontal.
e. Use a heavier motor, or weight the motor.
f. Use special long arms or the next larger size base.

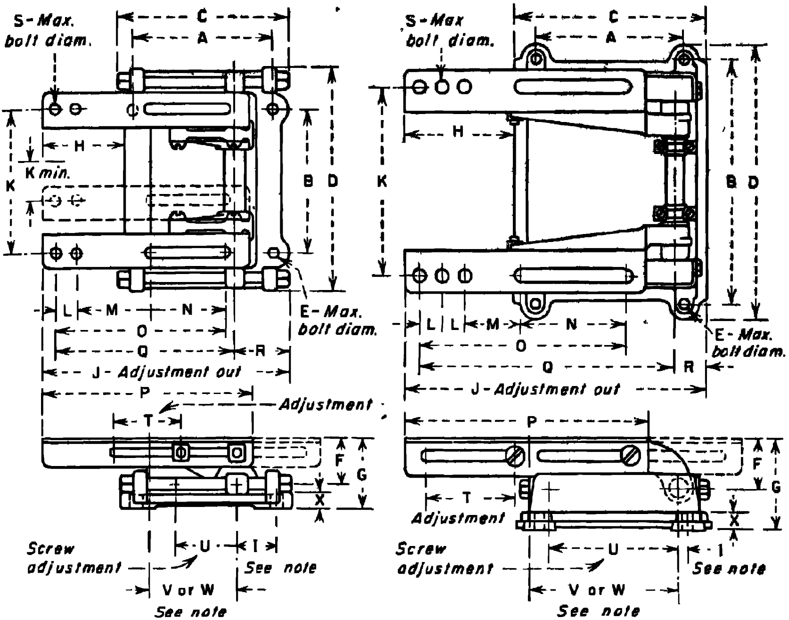
Table 14-71. Rockwood Floor-type Bases*

(See Fig. 14-23)

Base No.	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*J*	*K*		*L*	*M*	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	Shipping wt, lb	Max rotor wt received	
											Min	Max											See Note				Floor maintaining	Wall maintaining
000	5 5/8	5 1/4	7	8 3/8	3/8	1 1/2	2 3/8	2 5/8	2 1/4	9 5/8	1	7 3/8	1	2	4	7	8 1/4	7	1 3/4	3/8	2 1/2	3 3/8	1 3/4	2 5/8	9/16	8	65	40
00	6 1/2	7 1/8	8 3/8	10 5/8	3/8	1 15/16	2 15/16	3 7/8	2 5/8	12	1 3/4	9 3/8	1	4	4	9	10 1/4	9 1/4	2 3/16	3/8	3 1/4	3 3/4	2 3/4	3 3/4	5/8	13	100	60
0	7 5/8	12 3/8	10	14	1/2	2 3/8	4 1/2	6 7/16	2 3/4	16 7/16	5 3/8	8 1/2	1 1/4	3 1/2	5	11	13	13 5/8	1 13/16	1/2	5	6 3/8	4 7/8	6 1/2	1 3/16	35	150	125
2	9 3/8	15 1/4	12	17	5/8	3 1/16	5 1/2	7 1/16	3 1/2	19 1/16	6 1/4	10 1/2	1 5/8	3 7/8	5 7/8	13	15 1/8	16	2	5/8	5 7/8	8	5 1/4	7 1/2	1 1/16	55	250	200
4	11	17 1/4	14	19	5/8	3 1/2	6 1/4	8 1/8	3 7/8	22 1/8	7 1/2	13	1 5/8	4 7/8	6 7/8	15	17 3/4	18 1/2	2 1/4	5/8	6 7/8	9 1/2	6 1/2	8 3/4	1 1/4	80	450	350
6	12 3/4	19	16	21 1/8	3/4	3 13/16	7	9 1/8	4 5/8	25 1/8	8	14 1/4	2	5	8	17	20 1/4	21	2 1/2	3/4	8 1/8	11	7	9 3/4	1 1/2	115	650	500
8	14 1/4	21	18	23 1/4	3/4	4 5/16	8	10 1/16	5 1/8	28 1/16	9	15 1/4	2	6 1/2	8 1/2	19	22 1/2	23 5/16	3	3/4	8 7/8	12	7 3/4	10 3/4	1 3/4	160	900	700
10	16	25	20	27 3/8	7/8	4 9/16	8 1/2	11 15/16	5 7/8	31 15/16	10	18	2 3/8	8 3/8	8 7/8	22	25 3/4	26 11/16	3 3/8	7/8	10 1/2	13 1/4	8 3/4	12 1/4	1 3/4	245	1,500	1,200
12	19 1/8	30	23 1/2	32 3/4	1	5 9/16	10	14 9/16	7	38 1/16	11	20 1/2	2 3/8	12 1/4	10	27	31	32 5/16	3 3/4	7/8	13 1/16	16	10	14 1/2	1 3/4	400	2,500	2,000

* Courtesy of Rockwood Manufacturing Co.

NOTE: Before installing base move pivot shaft to position indicated by dimension *I*. This permits future arm adjustments without affecting shaft centers or belt length. Dimensions *V* and *W* are suggested distances from center-line pivot shaft to center-line motor (assuming its location approximately one-half *O* dimension). Use *V* when tight strand of belt is nearest pivot shaft and *W* when tight strand of belt is away from pivot shaft. Greater transmitting capacity may then be obtained by moving angle rails out on arms, increasing *V* or *W*. Less transmitting capacity is obtained by decreasing *V* or *W*.



Base sizes Nos. 000 and 00.

Base sizes Nos. 0–12, inclusive.

FIG. 14-23. Diagram for Table 14-71.

Solution for Flat Belt Width and Thickness

Belt cross section must be sufficient to accommodate the maximum belt tension Tm which equals T_1 plus the centrifugal tension.

$Tm = MT_1$ where M is a factor taken from Table 14-70 for the type of belt and belt speed.

Belt thickness is limited by the diameter of the small pulley. From Table 14-74, take the belt thickness suited for the small pulley diameter. Belts suited for still smaller pulleys will give increased life, particularly at high speeds.

Use the corresponding value of Tm per inch of width (Table 14-74).

$$\text{Belt width } W = T_1M/Tm \text{ per inch of width}$$

When the width comes to a fraction, use the next larger standard width (see Table 14-75 for maximum width of belt for motor in use). These values may change with different motor manufacturers, and they should be consulted on wider widths.

To find the horsepower of a given belt, solve the following equation.

$$\text{Hp} = WTmV/33{,}000KM$$

Solution for Belt Length

For open belt calculate $(D - d)/C$, and from Table 14-76 find the value of J, and then multiply by C. Calculate $D + d$, and from Table 14-77, find the corresponding value of X. Where $D + d$ is not an even number or is not given in the table, it may be calculated.

Belt length equals $JC + X$. This is tapeline length. Deduct 1 per cent for elasticity on leather belts. On very short center drives where the pulley faces almost touch each other, add the belt thickness to each pulley diameter before figuring the belt length. This corrects the pulleys to the pitch diameter and gives the correct flat measure of the belt.

Table 14-72. Rockwood Semistandard-type Bases[*]
(See Fig. 14-24)

No. 14 base No.	Max bolt-hole spacing	Max motor wt, lb	Y	D	Net base wt, lb	No. 16 base No.[‡]	Max bolt-hole spacing	Max motor wt, lb	Y	D	Net base wt, lb
14A	6	3,000	10	29	529	16A	18	6,000	26	46	1,278
14B	8	3,200	12	31	550	16B	20	6,200	28	48	1,296
14C	10	3,400	14	33	571	16C	22	6,400	30	50	1,315
14D	12	3,600	16	35	592	16D	24	6,600	32	52	1,334
14E	14	3,800	18	37	613	16E	26	6,800	34	54	1,353
14F	16	4,000	20	39	635	16F	28	7,000	36	56	1,371
14G	18	4,200	22	41	656	16G	30	7,200	38	58	1,390
14H	20	4,400	24	43	677	16H	32	7,400	40	60	1,409
14I	22	4,600	26	45	698	16I	34	7,600	42	62	1,428
14J	24	4,800	28	47	698[†]	16J	36	7,800	44	64	1,446
14K	26	5,000	30	49	717	16K	38	8,000	46	66	1,465
14L	28	5,200	32	51	736	16L	40	8,200	48	68	1,484
14M	30	5,400	34	53	755	16M	42	8,400	50	70	1,503

[*] Courtesy of Rockwood Manufacturing Co.

[†] This size and larger, also bases with 30-in. rails, have frame-constructed bottom plates. Add 8 lb to net weight for each 2-in. increase in length over 24 in.

[‡] Add 10¼ lb to net weight for each 2-in. increase in length over 40 in.

NOTE: Bases of all sizes can be supplied with bottom plates longer than standard at nominal extra cost (see price sheet). These usually are for bases that must fit over present foundation bolts, and bottom plate will be drilled to correspond to spacing of bolts in foundation. Raised foundations must be as wide and long as bottom plate.

NOTE: No. 14 bases only. Bases requiring angle-iron arm 40 in. or more long will be supplied with the 30-in. rail and bottom plate. Bases requiring shorter arms will be supplied with 24-in. rail and bottom plate.

When ordering specify horsepower, speed, make, type, frame number of motor. The latter very important. If not given, specify bolt size and spacing in motor feet. Give motor pulley diameter, kind and size of belt, driven speed or pulley diameter, shaft centers, and height of driven shaft above foundation top for Rockwood base.

Solution for Bearing Pressure

When the motor-pulley diameter is smaller than recommended by the motor manufacturer or wider than the standard width, the bearing pressure in the near-motor bearing may limit the belt tensions and should be checked as follows to see that it comes within the safe limits:

Nomenclature

G = the resultant of the tensions T_1 and T_2 under peak loads, lb
Z = the resultant of the tensions T_1 and T_2 at no load, lb
e = horizontal distance between the center of the motor pulley and pivot shaft, in.
Y = the perpendicular distance from pivot shaft to the line joining the pulley centers
W = motor weight
R = distance from center line of motor pulley to center of the far bearing
S = distance between centers of the two bearings
$Z = eW/Y$

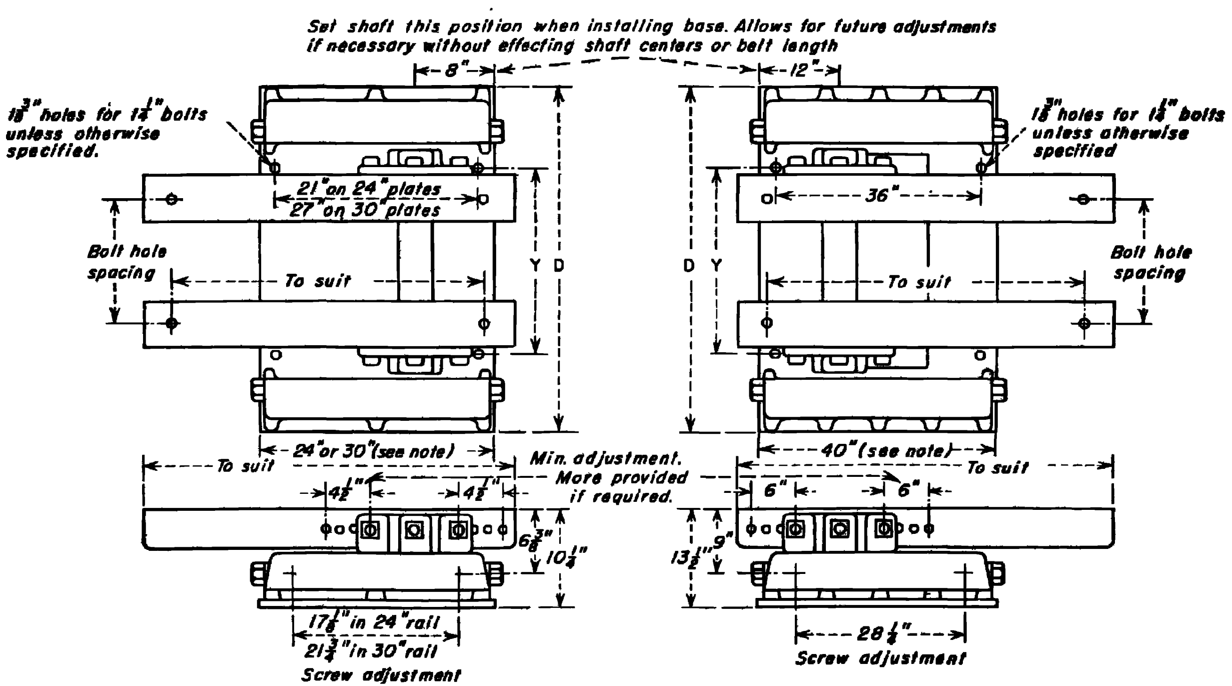
Using either Z or G, whichever is the larger, the maximum bearing pull = Z (or G) $\times$ (R/S).

Where R and S are not known, it is usually safe to assume that $R/S = \frac{4}{3}$.

The projected area for most motor bearings equals (shaft diameter inches)^2 $\times$ 2.5.

$$\text{Bearing pressure from belt} \quad \frac{\text{maximum bearing pull}}{\text{projected area}}$$

The maximum allowable bearing pressure varies with the speed, make of motor, and other conditions. Table 14-78a gives values that are generally safe.


Set shaft this position when installing base. Allows for future adjustments if necessary without effecting shaft centers or belt length
8"
12"
1 3/8" holes for 1 1/4" bolts unless otherwise specified.
1 3/8" holes for 1 1/4" bolts unless otherwise specified
21" on 24" plates
27" on 30" plates
36"
Bolt hole spacing
To suit
Y
D
Bolt hole spacing
24" or 30" (see note)
40" (see note)
To suit
Min. adjustment. More provided if required.
4 1/2"
4 1/2"
6"
6"
6 3/8"
10 1/4"
9"
13 1/2"
17 1/8" in 24" rail
21 3/4" in 30" rail
Screw adjustment
28 1/4"
Screw adjustment

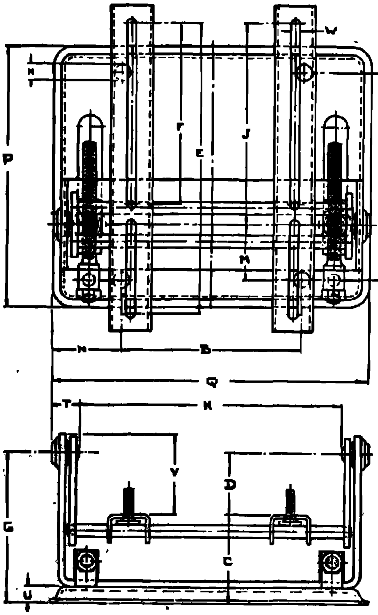
No. 14 semistandard bases

No. 16 semistandard bases

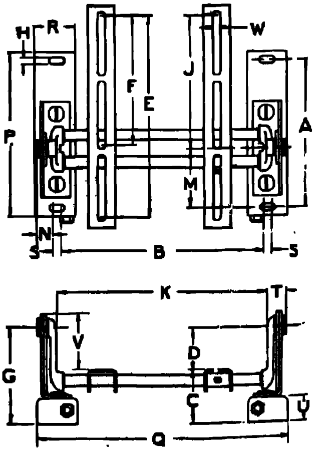
Fig. 14-24. Diagram for Table 14-72.

Table 14-73. American-

	Symbol No.											
	00	0	2		4		6		8		10	
	K-8½	*K*-12	*K*-14	*K*-18	*K*-18	*K*-20	*K*-22	*K*-24	*K*-24	*K*-28	*K*-28	*K*-32
A	6⅜	9¼	13½	13½	15	15	15	15	15	15	19½	19½
B	5½	7	13⅛	17⅛	17⅝	19⅝	21⅝	23⅝	24⅛	28⅛	27¾	31¾
C	2¾	4½	4⅞	4⅞	5¼	5¼	5¼	5¼	6 3/16	6 3/16	6¾	6¾
D	1⅞	2¼	2⅞	2⅞	3 11/16	3 11/16	3 11/16	3 11/16	4 1/16	4 1/16	5¼	5¼
E	9	16¾	17	17	20½	20½	20½	20½	25	25	28	28
F	5½	8	6	6	13¼	13¼	13¼	13¼	9¾	9¾	9¾	9¾
G	4⅝	6¾	7¾	7¾	8 15/16	8 15/16	8 15/16	8 15/16	10¼	10¼	12	12
H	½	9/16	21/32	21/32	21/32	21/32	21/32	21/32	21/32	21/32	25/32	25/32
J	6¼	12⅜	11¼	11¼	13½	13½	13½	13½	17	17	18	18
K	......	......	......	......	......	......	......	......	......	......	......	......
M†	1⅝–4⅝	3–6	4¾–8¾	4¾–8¾	5½–9½	5½–9½	5½–9½	5½–9½	5½–9½	5½–9½	7¼–12¼	7¼–12¼
N	2⅛	3	1⅝	1⅝	1 11/16	1 11/16	1 11/16	1 11/16	1 11/16	1 11/16	2⅜	2⅜
O	......	......	......	......	......	......	......	......	......	......	......	......
P	8	12 5/16	15	15	17	17	17	17	17	17	22	22
Q	9¾	13⅛	17⅞	21⅞	22½	24½	26½	28½	29	33	34	38
R	......	......	4	4	4⅛	4⅛	4⅛	4⅛	4⅛	4⅛	5½	5½
S	......	......	¾	¾	¾	¾	¾	¾	¾	¾	¾	¾
T	⅞	1⅛	1⅞	1⅞	2⅛	2⅛	2⅛	2⅛	2 9/16	2 9/16	2 31/32	2 31/32
U	½	¾	1 15/16	1 15/16	2 5/16	2 5/16	2 5/16	2 5/16	2 5/16	2 5/16	2 13/16	2 13/16
V	2½	3⅛	3 13/16	3 13/16	4 15/16	4 15/16	4 15/16	4 15/16	5 9/16	5 9/16	6¾	6¾
W	11/32	13/32	17/32	17/32	21/32	21/32	21/32	21/32	25/32	25/32	25/32	25/32
Z	......	......	......	......	......	......	......	......	......	......	......	......



(Nos. 00–0)
Floor, wall, and ceiling types



(Nos. 2,4,6,8,10,12,14)
Floor, wall, and ceiling types

base Dimensions*

Symbol No.									
12	14	15	16	18	*T*-2	*T*-4	*T*-6	*T*-8	*T*-10
K-30	*K*-40	—							
26	26	26¼	33	33	13½	15	15	15	19½
35	38¾	46	51½	55½	13¼	17¾	21¾	24⅛	27¾
7¾	8½	6½	9 7/16	7 15/16	4 15/16	5 7/16	5 7/16	6 7/16	6⅞
6¼	7	8	9½	11	2⅞	3 11/16	3 11/16	4 1/16	5¼
38¼	38¼	43¼	50¼	50¼	17	20½	20½	25	28
15¼	15¼	12	18¾	18¾	6	13¼	13¼	9¾	9¾
14	15½	14½	18 15/16	18 15/16	7 13/16	9⅛	9⅛	10½	12
29/32	29/32	1 5/16	1 3/16	1 3/16	21/32	21/32	21/32	21/32	25/32
23⅝	23⅝	24⅝	29¼	29¼	11¼	13½	13½	17	18
		40	44	48	14	18	22	24	28
9½–16½	9½–16½	14⅛–20⅛	14½–23	14½–23	2¼–6¼	2½–6½	2½–6½	4⅝–9⅝	3⅝–8⅝
2½	2½				1⅝	1 11/16	1 11/16	1 11/16	1⅞
					11¾–15¾	14⅞–18⅞	14⅞–18⅞	18½–22½	20–25
28½	28½	29½	38¼	38¼	15	17	17	17	22
43	46¾	55½	64½	68½	18	22⅝	26⅝	29	34
6½	6½	6½	8½	8½	4	4⅛	4⅛	4⅛	5½
1½	1½	3	4½	4½	¾	¾	¾	¼	¾
3½	3⅜	3⅞	4 15/16	4 15/16	1⅞	2⅛	2⅛	2 9/16	2⅞
3⅛	3⅛	2½	2 13/16	2 13/16	1 15/16	2 5/16	2 1/16	2 5/16	2 13/16
8	9	10	12¼	13¾	3 13/16	4 11/16	4 11/16	5 9/16	6¾
25/32	25/32	1 5/16	1 5/16	1 5/16	17/32	21/32	21/32	25/32	25/32
					19 15/16	24 5/16	24 1/16	27 13/16	32½

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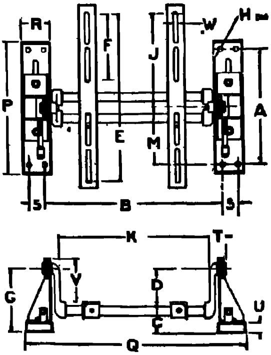
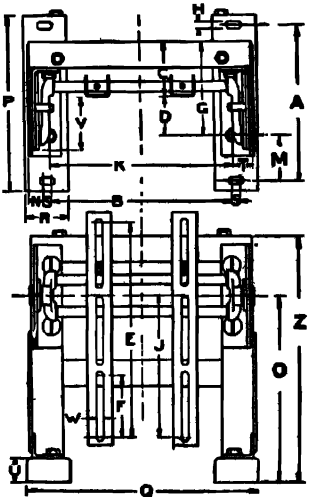
* Courtesy American Pulley Co

† *M* = Maximum and minimum distance from pivot to foundation bolt holes Difference between *M* dimensions adjustment available for belt take-up



(Nos. 15, 16, 18)
Floor type



(Nos T-2, T-4, T-6, T-8, T-10)
Type T

Pivoted Motor Drive
Work Sheet

Motor Specification
HP 50 RPM 860 Make G. E.
Type K Frame 505
Start Compensator
Driven Machine RPM 167
Direction of Pull Bottom
Centrifugal-fan Drive
Height of Driven Shaft Center above Floor 54 in.

Motor Data
$W = 1{,}300$ lb
$D = 12.5$ in.
$E = 10$ in.
$F = 9$ in.

Base Data
No. 12 Base
$G = 10$ in.
$F = 5\frac{5}{16}$ in.

$P = 50 \times 1.5$
$N = 860$
$D = 72$ in.
$d = 14$ in. paper
$C = 75$ in.

$$\frac{D-d}{C} = \frac{58}{75} = 0.77$$

$V = 3{,}150$
$k = 1.32$ (Table 14-69)

$$T_e = \frac{33{,}000 \times 75}{3{,}150} = 785$$

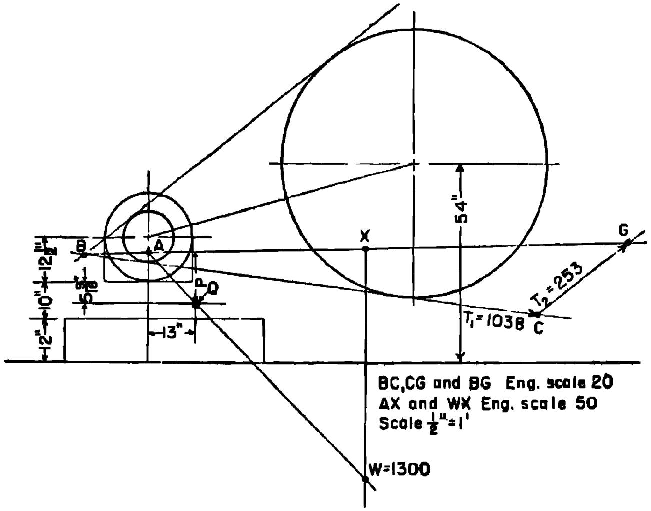
$T_1 = K \times 785 = 1{,}038$
$T_2 = T_1 - T_e = 253$
$G = 1{,}230$ scaled from drawing
$e = 13$ in. by measurement
To check work
$W_e = GP$
$1{,}300 \times 13 = 1{,}230 \times 13.75$
$16{,}900 = 16{,}920$

Belt Thickness
Medium Double Special Tanned
$M =$ (Table 14-70)

$$W = \frac{T_1 M}{T_m \text{ per in. of width}} = \frac{1{,}038 \times 1.11}{94 \text{ (Table 14-74)}}$$

$= 12.25$ use 12 in. medium double

Belt Length $= JC + X$
$J = 2.1505$ (Table 14-76)
$JC = 161.5$
$X = 135.1$ (Table 14-77)
$ 296.6$
$ -2.6$ approx. 1 per cent
$L = 294.0$ in. net length



Layout diagram

If the calculated bearing pressure exceeds these values, the drive is not safe as laid out, and the same corrective measures as in (7), page 1033, should be applied.

The bearing pressure will be highest when the resultant belt tension is highest. When the tight strand is nearer the pivot than the slack strand, the resultant belt tension becomes highest at the peak load; and when the slack belt strand is nearer

Table 14-74

Type of belt	Weight and ply	Min pulley diam, in.		T_m per in., width	Standard widths produced, in.		
		Belts under 8 in.	Belts 8 in. and over		Min	Intermediate	Max
Special tannage leather	Medium single	2	2½	52	1	By quarters to 2 in., by halves to	6
	Heavy single	2½	3	62	1	By quarters to 2 in., by halves to	6
	Light double	4	5	84	1½	By halves to 3 in., by ones to 8 in., by twos to	18
	Medium double	6	7	94	2	By halves to 3 in., by ones to 8 in., by twos to	30
	Heavy double	7	9	108	2	By halves to 3 in., by ones to 8 in., by twos to	36
	Medium 3 ply	15	20	142	4	By ones to 8 in., by twos to	76
	Heavy 3 ply	18	24	164	4	By ones to 8 in., by twos to	76
Oak-tanned leather	Medium single	3	5	52	1	By halves to 4 in., by ones to	6
	Heavy single	5	7	62	1	By halves to 4 in., by ones to	6
	Light double	6	8	84	1½	By halves to 3 in., by ones to 8 in., by twos to	18
	Medium double	8	10	94	2	By halves to 3 in., by ones to 8 in., by twos to	30
	Heavy double	12	14	108	2	By halves to 3 in., by ones to 8 in., by twos to	36
	Medium 3 ply	20	24	142	4	By ones to 8 in., by twos to	76
	Heavy 3 ply	24	30	164	4	By ones to 8 in., by twos to	76
32-oz duck rubber	3 ply	4		45	1	By halves to	4
	4 ply	6		60	2	By halves to 4 in., by ones to 8 in., by twos to	12
	5 ply	8		75	4	By ones to 8 in., by twos to	24
	6 ply	12		90	6	By ones to 8 in., by twos to	42
	7 ply	18		105	26	By twos to	54
	8 ply	24		120	30	By twos to	72
	9 ply	30		135	30	By twos to	72
	10 ply	36		150	30	By twos to	72

Table 14-75. Maximum Belt Widths for Motors, In.

Motor horsepower	Nominal motor speeds, rpm						
	1,800	1,200	900	720	600	514	450
1	2½	2½	4				
1½	4	4	4				
2	4	4	4				
3	4	4	5	6			
5	4	5	6	7			
7½	5	6	7	9			
10	6	7	9	10			
15	7	9	10	10			
20	9	10	10	12			
25	9	10	12	12	12		
30	10	12	12	12	14		
40	12	12	14	14	14		
50	....	14	14	16	16		
60	..	14	14	16	16		
75	....	14	16	18	18	18	20
100	...	14	16	18	18	22	22
125	.	....	18	20	20	22	22
150	.	....	20	20	22	22	24

Table 14-76. Values of J Corresponding to Values of $(D - d)/C$ for Calculating Belt Length on Open Drives

$(D - d)/C$	J	$(D - d)/C$	J	$(D - d)/C$	J	$(D - d)/C$	J
0 00	2 0000	0 26	2 0169	0 51	2 0654		
0 01	2 0000	0 27	2 0183	0 52	2 0680	0 76	2 1466
0 02	2 0001	0 28	2 0197	0 53	2 0707	0 77	2 1505
0 03	2 0002	0 29	2 0212	0 54	2 0735	0 78	2 1545
0.04	2 0004	0 30	2 0227	0 55	2 0763	0 79	2 1585
0 05	2 0006	0 31	2 0243	0 56	2 0791	0 80	2 1625
0 06	2 0009	0 32	2 0259	0 57	2 0820	0 81	2 1666
0 07	2 0013	0 33	2 0275	0 58	2 0849	0 82	2 1708
0 08	2 0017	0 34	2 0291	0 59	2 0878	0 83	2 1751
0 09	2 0021	0.35	2 0308	0 60	2 0908	0 84	2.1794
0 10	2 0026	0 36	2 0325	0 61	2 0939	0 85	2 1838
0.11	2 0031	0 37	2 0343	0 62	2 0970	0 86	2 1882
0 12	2 0037	0 38	2 0362	0 63	2 1002	0 87	2 1927
0 13	2 0043	0 39	2 0382	0 64	2 1034	0 88	2 1972
0.14	2 0050	0 40	2 0402	0 65	2 1067	0 89	2 2018
0 15	2 0057	0 41	2 0423	0 66	2 1100	0 90	2 2065
0 16	2 0065	0 42	2 0444	0 67	2 1134	0 91	2 2112
0 17	2 0074	0 43	2 0466	0 68	2 1169	0 92	2 2159
0 18	2 0083	0 44	2 0488	0 69	2 1205	0 93	2 2207
0.19	2 0092	0 45	2 0510	0 70	2 1241	0 94	2 2255
0 20	2 0101	0 46	2 0533	0 71	2 1277	0 95	2 2304
0 21	2 0111	0 47	2 0556	0 72	2 1314	0 96	2 2353
0 22	2 0122	0 48	2 0580	0 73	2 1351	0 97	2 2403
0 23	2 0133	0 49	2 0604	0 74	2 1389	0 98	2 2454
0 24	2 0144	0 50	2 0629	0 75	2 1427	0 99	2 2506
0 25	2 0156					1 00	2 2558

Table 14-77. Values of X Corresponding to Values of $D + d$ for Calculating Belt Length

(Where $X = (D + d/2)\pi$)

$D + d$	X	$D + d$	X	$D + d$	X	$D + d$	X
1	1 6	31	48 7	61	95 8	91	142 9
2	3 1	32	50 3	62	97 4	92	144 5
3	4 7	33	51 8	63	99 0	93	146 1
4	6 3	34	53 4	64	100 5	94	147 7
5	7 9	35	55 0	65	102 1	95	149 2
6	9 4	36	56 5	66	103 7	96	150 8
7	11 0	37	58 1	67	105 2	97	152 4
8	12 6	38	59 7	68	106 8	98	153 9
9	14 1	39	61 3	69	108 4	99	155 5
10	15 7	40	62 8	70	110 0	100	157 1
11	17 3	41	64 4	71	111 5	101	158 7
12	18 9	42	66 0	72	113 1	102	160 2
13	20 4	43	67 5	73	114 7	103	161 8
14	22 0	44	69 1	74	116 2	104	163 4
15	23 6	45	70 7	75	117 8	105	164 9
16	25 1	46	72 3	76	119 4	106	166 5
17	26 7	47	73 8	77	121 0	107	168 1
18	28 3	48	75 4	78	122 5	108	169 7
19	29 8	49	77 0	79	124 1	109	171 2
20	31 4	50	78 5	80	125 7	110	172 8
21	33 0	51	80 1	81	127 2	111	174 4
22	34 6	52	81 7	82	128 8	112	175 9
23	36 1	53	83 3	83	130 4	113	177 5
24	37 7	54	84 8	84	132 0	114	179 1
25	39 3	55	86 4	85	133 5	115	180 6
26	40 8	56	88 0	86	135 1	116	182 2
27	42 4	57	89 5	87	136 7	117	183 8
28	44 0	58	91 1	88	138 2	118	185 4
29	45 6	59	92 7	89	139 8	119	186 9
30	47 1	60	94 2	90	141 4	120	188 5

Table 14-78a

Motor speed, rpm	*Maximum bearing pressure, psi projected area*
600	110
720	100
900	90
1,200	75
1,800	60

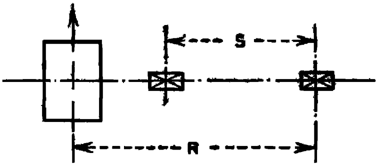


Fig. 14-25. Bearing pressure diagram.

the pivot than the tight strand, the resultant belt tension is highest at zero load. Using the fan-drive work sheet for data:

Example: $Z = eW/Y$
Let e = 13 in.
Y = 20.75
W = 1,300
G = 1,230
R/S = 4/3 (assume, since R/S is unknown)
Motor speed = 870 rpm

$$Z = (13 \times 1{,}300)/20.75 = 81$$

Use G = 1,230 as it is larger than Z = 814.

$$\text{Max bearing pull} = 1{,}230 \times \tfrac{4}{3} = 1{,}640 \text{ lb}$$
$$\text{Projected bearing area} = 3^2 \times 2.5 = 22.5$$

Using G, the bearing pressure from belt = (1,640/22.5) = 72.8 psi of projected area.

The maximum bearing pressure allowable from Table 14-78a above indicates that this figure will be satisfactory. This pressure comes only at extreme peak load so that the running pressure is below this figure.

INSTALLATION

Place the motor base on its foundations with the pivot shaft moved back about one-third the total take-up. With the pivot in this position, future increase in tension, if necessary, can be had without changing the center distance. Change the oil cups on the motor to the machine side of the motor. Put oil in the bearings, and with the belt in place start the drive up. Any small amount of misalignment can be corrected by turning one of the take-up bolts. Some small amount of tension increase or decrease may be necessary depending upon how close your original assumption of the peak load required was to the actual load. Some very heavy static friction loads, as in a heavy fan, may have to be freed before the motor will start the fan. This in no way is a reflection on the belt drive.

POSSIBLE PIVOTED-MOTOR-BASE-DRIVE DIFFICULTIES AND HOW TO OVERCOME THEM

Following the graphical solution for the pivot setting should eliminate most difficulties that are encountered. Vibration can in most cases be reduced to a minimum. Usually the cause suggests the cure. In some cases bracing the base support may eliminate vibration. Slightly more or less tension may be required. The bolts holding the motor to the base may be loose, or the cap bolts holding the arms to the pivot hubs may be loose. A worn bearing may be the cause. Checking the period of vibration of the motor with the belt or driven machine may help in locating the difficulty. Divide the belt length in feet into the belt speed in feet per minute. If this number is the same as the number of vibrations per minute, the belt is the cause.

MOTOR-ROTOR FLOAT

1. Often misalignment causes the belt to hold the motor rotor tight against the retaining colars, and there is no float at all. With correct alignment and with pulleys in the same plane, this should be eliminated.

2. Motor-rotor end play may not be sufficient. This should be increased to $\frac{1}{8}$ in. or reduced to practically nothing by using fiber washers.

The belt runs on the center of the pulley when load is on but runs off center with no load. Misalignment is the cause. If the belt will not run true by turning one of the center distance adjusting screws, the drive is out of level with the driven machine and should be corrected.

Advantages

To sum up, the main advantages of pivoted motor-base drive are

1. Complete dependability
2. Low maintenance
3. Constant correct tension (overcomes centrifugal tension)
4. Maintains its high efficiency over a wide range of loading (see Fig. 14-22)
5. Requires little space
6. Furnishes long life to belt and pulleys
7. Has a definite release point eliminating costly repairs to expensive machinery if a jam occurs
8. Easy drive alignment
9. Inexpensive to install

SPRING AUTOMATIC MOTOR BASES

The Automatic motor base is a device used in place of motor slide rails. It can be used in any position with any direction of rotation of the motor and no engineering is required for proper application. The base consists of a stationary member, motor plate, tensioning springs and adjusting screw.

Selection. To determine the size of base required, obtain the NEMA frame number from the motor name plate. Then refer to Table 14-78*b* of NEMA equivalents where the relation of frame numbers and base sizes is given.

In the absence of the NEMA frame number, ascertain the horsepower and speed of the motor and the distance between centers of the bolt holes. Then refer to Table 14-78c (Rating) and Table 14-78*d* (Dimensions). The slots are arranged to accommodate motors having standard NEMA frames. If the motor-bolt holes are not located to NEMA standards, use one pair of slots and drill two holes.

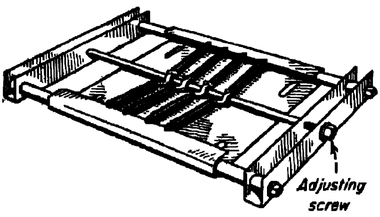


FIG. 14-26. Base 35 inverted to illustrate arrangement of springs.

On sizes 40, 45, 50, and 55 the motor support is articulated and may be separated so the slots will accommodate nonstandard motors.

For vertical down drive, *i.e.*, where the motor is side-wall mounted with the shaft horizontal and located above the driven pulley, specify "vertical" when ordering. "Vertical" bases should also be used where the motor-pulley diameter is smaller than recommended by NEMA standards and for all chain drive applications. "Vertical" bases are equipped with larger adjusting screws and with additional or heavier springs.

Figure 14-26 shows the spring arrangement and the simplicity of the base design.

The recent trend to increase motor horsepower without increasing the size of frame and the fact that the enclosed and explosion-proof motors are usually in larger frames, accounts for more than one size of base being listed under the same rating in Table 14-78c.

As the chain or belt stretches from normal use and wear, the tension is easily readjusted by turning a screw while operating under load.

Motor-mounting position, direction of pulley rotation, the type of belt or chain used or the nature of the load in no way affects its operation. Figure 14-27 shows a typical flat-belt installation.

Table 14-78b.
Nema Equivalents

Nema frame No.	*Base No.*
203	15
204	15
224	20
225	20
254	25
284	30
324	35
326	35
364	40
365	40
404	45
405	45
444	50
445	50
504	55
505	55

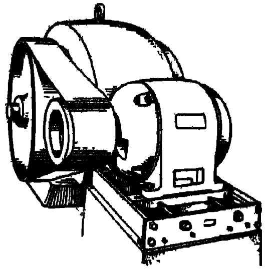


FIG. 14-27. A 50-hp motor mounted on a 55-R Automatic base.

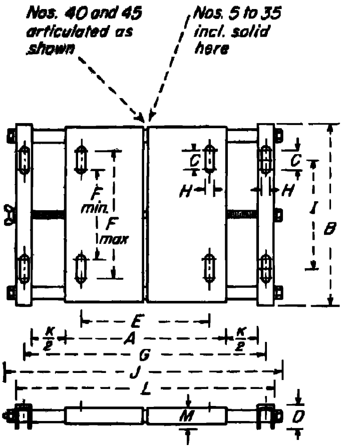
The type R automatic was recently developed for motors up to 100 hp. The "R" stands for roller bearings, eight of them, which ride on cold-drawn steel tracks and support the motor. On the larger sizes, duplex springs and adjusting screws are used. The screws are connected by a roller chain, hence their action is synchronized. Turning either screw adjusts both springs simultaneously. This base is extremely compact and can be mounted in any position.

Installation. The automatic motor base is bolted to the motor feet and together they are bolted down as a unit. With the motor plate all the way forward, place the belt or belts on the drive. Apply tension to the drive by turning the adjusting screw shown in Fig. 14-26. This puts tension in the springs and belt by pulling the motor plate away from the driven machine. With the motor running, final tension adjustment can be made to handle peak loads.

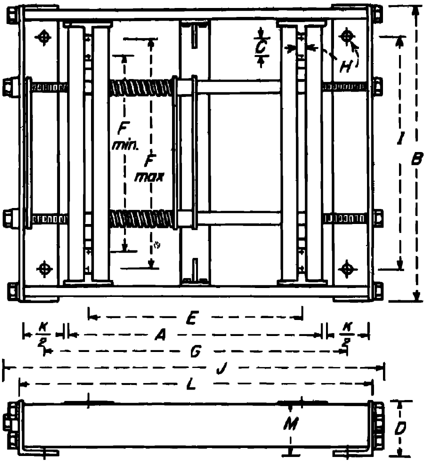
Table 14-78c. Rating Table

Motor hp	Any phase 60-cycle a-c, or d-c motors, any voltage, rpm						25- or 50-cycle a-c motors, rpm	
	1,800	1,200	900	720	600	514	1,500	750
⅙-⅓	5	10	10-10A	15	20			
½	10	10-A	15-20	20	20			
¾	10-A	15-20	20	20	25	30	15	20
1	15-20	15-20	20	25	30	30-35	15-20	25
1½	15-20	20	25	30	30-35	35	20	30
2	20	20-25	25-30	30-35	35	35-40	20-25	30-35
3	20-25	25-30	30-35	35	35-40	40-45	25	35
5	25-30	30	35	35-40	40-45	45	30	35-40
7½	30	35	35	40-45	40-45	45-50	30-35	40-45
10	35	35	40	40-45	45-50	50	35	40-45
15	35	40	40-45	45-50	45-50	50-55	40	45-50
20	40	40-45	45	45-50	50-55	55	40-45	45-50
25	40	45	45-50	50-55	50-55	55	45	50-55
30	40-45	45-50	50-55	50-55	55	.....	45-50	50-55
40	45-50	50-55	50-55	55	.....	...	50	55
50	50-55	50-55	55	.....	.....	.....	50-55	
60	50-55	55	.		....	....	50-55	
75	50-55	.....	..	.	....	.....	55	
100	55							

Table 14-78*d*. The "Automatic" Motor Base


Nos. 40 and 45 articulated as shown
Nos. 5 to 35 incl. solid here
C
H
F min.
F max
I
B
E
K/2
A
G
J
L
M
D

Sliding type
sizes 5 to 45 incl



Type-R
sizes 50 and 55

Base No.	Dimensions															Shipping wt, lb
	A	*B*	*C*	*D*	*E*	*F* min	*F* max	*G*	*H*	*I*	*J*	*K*-travel	*L*	*M*	*N*	
5	6⅞	7⅛	1 1/16	1⅜	4⅞	2½	4⅜	9¾	11/32	3¾	11 13/16	2	10 11/16	1 5/16	15/16	6
10	8	7⅛	1 1/16	1⅜	4⅞	2½	4⅝	11	11/32	3¾	13 1/16	2⅛	11 15/16	1 5/16	15/16	7
10-A	9½	8½	1¼	1 9/16	5⅞	3⅛	5⅝	12⅞	13/32	4⅜	15 5/16	2 5/16	13 15/16	1 5/16	1 1/16	9
15	10¼	10	1¼	1 13/16	8	4⅛	6⅝	14 5/16	13/32	5½	16¾	3	15⅜	1 9/16	1 1/16	13
20	11¼	11½	1¼	2	9	5⅜	7⅞	15½	13/32	6⅝	18⅜	3	16¾	1¾	1¼	16
25	13⅛	13	1⅝	2¼	10	5⅝	8⅞	18 13/16	17/32	7⅜	21⅞	4⅜	20⅛	1 15/16	1 5/16	25
30	14½	15	1⅝	2¼	11	7¼	10½	20⅞	17/32	9⅜	23 15/16	5 1/16	22 3/16	2	1 5/16	31
35	16⅛	18	2¼	2⅜	12½	9⅛	13⅝	23⅝	21/32	11⅛	26⅞	6	25⅛	2 1/16	1½	43
40	20*	20	2¼	3⅛	14*	10	14½	28¼	21/32	12¼	32½	5¾	30¾	2 11/16	2½	100
45	20*	21½	2¼	3 5/16	16*	10½	15	28½	13/16	11¾	33¼	5⅞	31⅛	2¾	2⅝	140
50-R	23*	26⅛	1⅞	5	18*	13½	17¼	28½	13/16	20¾	34⅞	7	32	4⅛	3½	200
55-R	26*	28¼	2	5	20*	15	19	32⅜	15/16	22	38⅝	8	36⅜	4⅞	3½	285

* Variable.

CHAIN DRIVING

BY CHARLES R. WEISS

Chief Engineer, Chain Division, Link-Belt Company

INTRODUCTION

Mechanical transmission of power and speed conversion through chains is extensively employed because of the following factors:

1. Efficiency comparable to any means or method of positive driving. No slippage.
2. Flexibility and ease of application to meet widely varying industrial requirements.
3. Absorption of shock with less vibration than with gears.
4. Reasonable installation costs.

A chain drive consists of a chain connecting a *driver* sprocket to one or more *driven* sprockets. Pitch, either chain pitch or sprocket pitch, is the distance from center to center of two adjacent chain joints. The pitch circle of a sprocket would be the circle drawn through the centers of the chain joints of an endless chain, all the links of which are in mesh with all the teeth of the sprocket. The diameter of the pitch circle is called the "pitch diameter" of the sprocket. The number of teeth in the sprocket times the pitch of the chain in inches, times the rpm of the sprocket equals its pitch-line speed in terms of inches per minute, or dividing by 12 gives it in feet per minute (fpm).

The types of chain in most general use are treated in this subsection. However, many various other types of chain are produced, together with devices for adapting the various types of chains for use in conveyance.

Table 14-79. Industrial Standards for Open-type Detachable Chains*

(Chains listed in the order of their strengths)

Chain No.	Pitch, in.	Avg ultimate strength, lb (mall. iron)	Avg wt per ft, lb
25	0.902	700	0.24
32	1.154	1,100	0.32
33	1.394	1,200	0.32
34	1.398	1,300	0.40
42	1.375	1,600	0.55
45	1.630	1,700	0.52
51	1.155	1,900	0.70
55	1.631	2,300	0.72
52	1.506	2,300	0.80
62	1.654	3,200	1.04
77	2.297	3,600	1.45
78	2.609	5,500	1.86
88	2.609	6,400	2.30
103	3.075	10,000	4.00
114	3.250	12,000	5 25
124	4.063	17,000	6.40

* Courtesy Link-Belt Co.

OPEN-TYPE MALLEABLE-IRON AND PRESSED-STEEL CHAIN

In 1874, the first chain, the Ewart-type detachable shown by Fig. 14-28, was developed and is still produced for speeds up to 400-fpm for low-load non-abrasive installations for power transmission and, with suitable carriers, for conveyance. It consists of cast malleable-iron links designed to allow ease of

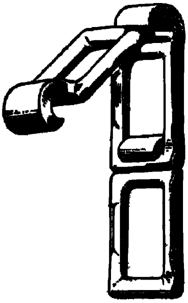


FIG. 14-28. Ewart-type detachable chain.

assembly and disassembly; usually operating on cast-tooth cast-iron sprockets. A number of common sizes are given in Table 14-79. In recent years this chain has also been produced in pearlitic malleable iron, increasing ultimate strength approximately 25 per cent with higher wear-resistant qualities, in bronze for corrosion resistance, and in chrome-nickel alloys, in the larger pitch sizes, for corrosion resistance.

In the smaller sizes the Ewart-type chain is made in heat-treated steel stampings with substantially identical performance characteristics at a slightly lower cost and is used to the greatest extent on agricultural machinery with special sprockets.

CLOSED-TYPE MALLEABLE-IRON CHAIN

The closed-type malleable-iron chain, as shown by Fig. 14-29, is used for chain installations of speeds up to 400 fpm where loads are greater than could be carried with open-type chain and where increased abrasive-resistant qualities are required. The sprockets used are identical with those for open-type chains. A number of common sizes are given in Table 14-80. Other stock sizes and special types are manufactured.

Table 14-80. Closed-type Malleable-iron Chains*

(Chains listed in the order of their strengths)

Chain No.	Pitch, in.	Avg ultimate strength, lb (mall. iron)	Avg wt per ft, lb
942 or 442	1.375	6,000	1.4
945 or 445	1 630	6,000	1.5
952 or 452	1.506	7,000	2.0
955 or 455	1 630	7,300	1 9
962 or 462	1.634	9,000	2 5
977 or 477	2.308	9,600	2.0
988 or 488	2.609	11,000	2.9
9103 or 4103	3.075	22,000	5 7
4124	4.063	33,000	8.5

* Courtesy Link-Belt Co.

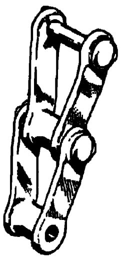


FIG. 14-29. Closed-type malleable-iron chain.

The closed type is also manufactured in pearlitic malleable iron for increased ultimate strength, in bronze for corrosive resistance, in stainless steel for corrosive resistance; and in chrome-nickel alloy for corrosive and heat resistance.

STEEL ROLLER CHAIN

Steel roller chain is constructed from steel side bars and hardened steel pins, bushings, and rollers. It will operate at speeds up to 700 fpm under heavier loads and with improved wear resistance as compared with corresponding sizes of malleable-iron chains. Parts may consist of stainless or alloy steel for special applications. This chain will operate on the same sprockets as malleable-iron chains, although for heavier loads a sprocket of more rugged design is required. Cast-tooth sprockets are generally used because of lower initial cost. However, operation at the higher speeds necessitates cut-tooth sprockets to increase chain life with smoother operation. In selecting sprockets, it is generally recommended, for increased chain life with more satisfactory service, that the smaller sprocket have not less than 10 teeth, which will usually limit ratios to not more than 5:1. A limited list of popular sizes is given in Table 14-81. Numerous other stock sizes are produced by various manufacturers, together with many special designs.

Finished-steel Roller Chain (ASA Standard). *Manufacture.* The links, rollers, pins, and bushings of this type, as shown by Fig. 14-30, are accurately made of high-grade steel, hardened, with the latter two being finish-ground. Owing to the precision manufacture, the pitches are held to close tolerances. This results in smooth operating action and increased chain and sprocket life. These chains are made of other materials, including bronze and stainless steel, to meet special operating requirements.

Selection of Chain Size. The revolutions per minute of the faster running shaft

Table 14-81. Steel Roller Chain, Offset-sidebar Type*
(Industrial standards)

Chain No.	Pitch, in.	Ultimate strength, lb.
IS-620	1 654	8,000
IS-622	1 654	20,000
IS-2560	2.500	60,000
IS-881	2.609	20,000
IS-882	2.609	28,000
IS-1032	3.075	60,000
IS-3514	3 500	140,000
IS-1242	4 063	140,000
IS-1245	4.073	170,000

* Courtesy Link-Belt Co.

Table 14-82. Speeds and Pitches of Finished-steel Roller Chain*

RPM of small sprocket	Teeth, if used as a driver	Teeth, if driven	Trial pitch, in.
2,200–3,200	21	23	3/8
1,700–2,200	21	23	1/2
1,400–1,700	21	23	5/8
1,000–1,400	21	23	3/4
800–1,000	21	23	1.0
500– 800	21	23	1 1/4
400– 500	21	23	1 1/2
340– 400	21	23	1 3/4
250– 340	21	23	2.0
Below 250	21	23	2 1/2

* Courtesy Link-Belt Co.

limits the maximum pitch of chain that should be employed. A trial chain pitch should be selected from Table 14-82. If the single-width trial chain does not meet working-load-capacity requirements, proper multiple width of chain is selected.

Working Load Capacities. The average ultimate tensile strength for the various pitches of single-strand finished-steel roller chain are shown in Table 14-83.

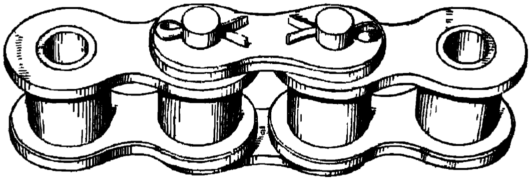


FIG. 14-30. Finished-steel roller chain.

The working load capacity based upon durability and ultimate strength is governed by horsepower, chain speed, and operating conditions. Permissible working loads for single-strand chain for various pitches and speeds are given in Table 14-84. The horsepower values listed in this table are based on 10-hr-per-day service, uniform loads, and good lubrication, where there is little shock or sharp variation in speed throughout a single revolution of the driven wheel. This rating can be adjusted to other periods of service and load characteristics by multiplying the known horsepower to be transmitted by the factors given in Table 14-85.

Table 14-83. Manufacturers' (ASA) Standard Sizes*

Chain No.	Pitch, in.	Average ultimate strength, lb	Wt per foot, lb
35	3/8	2,100	0.20
40	1/2	3,700	0.41
50	5/8	6,100	0.64
60	3/4	8,500	1 00
80	1 0	14,500	1 68
100	1 1/4	24,000	2 56
120	1 1/2	34,000	3.73
140	1 3/4	46,000	4 65
160	2 0	58,000	6 32
200	2 1/2	95,000	11 00

* Courtesy Link-Belt Co.

Chain Speeds. Finished-steel roller chains operate from 1,400 to 2,400 fpm depending upon number of teeth in faster running sprocket and chain pitch, as given in Table 14-84. Under certain operating conditions, roller chains can be operated at speeds up to 4,000 fpm, but the chain manufacturer should be consulted in such cases. Proper lubrication is essential for satisfactory chain life, and the importance of sufficient lubrication increases with the higher chain speeds.

Sprockets. Sprockets are all cut tooth, accurately machined from grey iron, cast steel, and forged or bar-stock steel, with pinions hardened and the larger sprockets

Table 14-84. Horsepower of Single-strand Finished-steel Roller Chain, Based on Pitch and Speed*

Pitch	No. of teeth	Revolutions per minute											
		100	200	400	600	800	1,000	1,200	1,600	2,000	2,400	3,000	3,600
⅜, No. 35	17	0.26	0.49	0.90	1.25	1.56	1.83	2.08	2.48	2.79	3.03	3.23	3.30
	19	0.30	0.56	1.01	1.41	1.76	2.07	2.36	2.82	3.19	3.47	3.74	3.83
	21	0.33	0.61	1.12	1.56	1.95	2.30	2.61	3.14	3.53	3.86	4.16	4.28
	23	0.36	0.67	1.23	1.71	2.14	2.51	2.85	3.42	3.89	4.22	4.56	4.68
	30	0.47	0.87	1.58	2.19	2.74	3.22	3.64	4.33	4.90	5.28	5.64	5.73
	35	0.54	1.01	1.82	2 52	3.13	3 67	4.13	4.90	5.47	5.87	6.17	
	40	0.61	1.14	2.05	2.83	3.50	4.08	4.59	5.39	5.96	6.32	6.55	
		100	200	300	400	600	800	1,000	1,200	1,400	1,600	2,000	2,400
½, No. 40	17	0.61	1.13	1.59	2.02	2.77	3.41	3.95	4.41	4.80	5.10	5.57	5.81
	19	0.69	1.27	1.80	2.28	3.14	3.86	4.49	5.02	5.48	5.85	6.41	6.75
	21	0.76	1.41	1.99	2.52	3.47	4.27	4.97	5.57	6.07	6.50	7.13	7.50
	23	0.83	1.54	2.18	2.76	3.79	4.68	5.42	6.09	6.64	7.11	7.80	8.20
	30	1.08	1.99	2.81	3.56	4.86	5.95	6.93	7.76	8.40	8.90	9.72	10.1
	35	1.25	2.30	3.24	4.09	5.56	6.81	7.86	8.71	9.42	9.99	10.7	11.0
	40	1.41	2.60	3.65	4.59	6.22	7.57	8.07	9.60	10.3	10.9	11.5	
		100	200	300	400	600	800	1,000	1,200	1,400	1,600	1,800	2,000
⅝, No. 50	17	1.16	2.14	2.99	3.77	5.09	6.17	7.05	7.75	8.31	8.70	8.98	9.13
	19	1.31	2.41	3.39	4.25	5.76	7.00	8 04	8.87	9.52	10.0	10.4	10.6
	21	1.45	2.66	3.75	4.70	6.38	7.77	8.90	9.84	10.6	11.1	11.6	11.8
	23	1.58	2.91	4.09	5.16	6.98	8.50	9.74	10.8	11.6	12.2	12 6	12 9
	30	2.06	3.77	5.28	6.63	8.93	10.8	12.4	13.6	14.5	15.2	15 6	15.9
	35	2.38	4.35	6.07	7.59	10.2	12.3	13.9	15.2	16.1	16.7	17.1	
	40	2.70	4.91	6.82	8.51	11.3	13.6	15.3	16.6	17.5	18.0		
		50	100	200	300	400	500	600	800	1,000	1,200	1,400	1,600
¾, No. 60	17	1.05	1.97	3.59	4 98	6.22	7.32	8 29	9 91	11.1	12.1	12.7	
	19	1.18	2.23	4.05	5.62	7.03	8.29	9 42	11.3	12 8	13.9	14 7	15 2
	21	1.31	2.46	4.49	6.24	7.80	9.19	10.4	12.5	14 1	15.4	16.3	16.9
	23	1.44	2.69	4.90	6.83	8.53	10.1	11.4	13.7	15.5	16.9	17.8	18.5
	30	1.86	3.48	6.32	8.78	10.9	12.8	14.6	17.4	19.5	21.1	22.2	
	35	2.16	4.03	7.29	10.1	12.5	14 7	16 5	19.6	21 9	23.4		
	40	2.45	4.55	8.20	11.3	14.0	16.3	18 4	21.6	23 9			
		25	50	100	150	200	300	400	500	600	700	800	1,000
1.0, No. 80	17	1.29	2.44	4.52	6.39	8.08	11.1	13.6	15.8	17.6	19.2	20.5	22.3
	19	1.44	2.74	5.09	7.20	9.12	12.5	15.4	17.9	20.1	21.9	23.4	25 7
	21	1.59	3.03	5.62	7.96	10.1	13 9	17.1	19.9	22.3	24.3	26.0	28 5
	23	1.75	3.32	6.16	8.72	11.1	15.2	18.7	21.8	24.4	26.6	28.4	31.2
	30	2.27	4.30	7.96	11.2	14 2	19.5	23.9	27.7	30.9	33.6	35.8	38.8
	35	2.63	4.98	9.19	13.0	16.4	22.3	27.2	31.4	34.9	37.7	40.0	
	40	2.99	5.65	10.4	14.6	18.4	24 9	30.3	34.7	38.4	41.2	43.5	
		10	25	50	100	150	200	300	400	500	575	700	800
1¼, No. 100	17	1.05	2.49	4.67	8.56	12.0	15.1	20.3	25.0	28.2	30.3	33.2	
	19	1.17	2.79	5.24	9.64	13.6	17.0	23.0	28.3	32.2	34.7	38.1	40.0
	21	1.29	3.08	5.80	10.7	15.0	18.9	25.5	31.4	35.7	38.5	42.3	44.6
	23	1.42	3.38	6.36	11.7	16.4	20.6	28.0	34.0	39.0	42.2	46.2	48.8
	30	1.84	4.38	8.22	15.1	21.1	26.5	35.7	43.2	49.3	53.1	58.0	
	35	2.15	5.09	9.53	17.4	24.3	30.4	40 8	49.0	55.6	59.5		
	40	2.44	5.77	10.8	19.6	27.3	34.0	44.3	54.2	61.0	65.0		

Table 14-84. Horsepower of Single-strand Finished-steel Roller Chain, Based on Pitch and Speed* (*Cont.*)

Pitch	No. of teeth	Revolutions per minute											
		5	10	25	50	100	150	200	250	300	400	500	600
1½, No. 120	17	0.93	1.79	4.23	7.91	14.4	19.9	24.9	29.3	33.2	39.7	44.6	
	19	1.03	2.01	4.74	8.90	16.2	22.6	28.2	33.2	37.6	45.2	51.1	
	21	1.14	2.23	5.24	9.85	17.9	24.9	31.2	36.8	41.7	50.0	56.7	61.7
	23	1.25	2.43	5.74	10.8	19.6	27.3	34.2	40.2	45.7	54.8	62.1	67.5
	30	1.63	3.17	7.44	13.9	25.3	35.2	43.8	51.4	58.3	69.5	78.2	
	35	1.89	3.68	8.64	16.1	29.1	40.3	50.1	58.6	66.2	78.3		
	40	2.16	4.18	9.80	18.2	32.8	45.2	56.0	65.3	73.4	86.1		
		5	10	20	50	100	150	200	250	300	350	400	450
1¾, No. 140	17	1.45	2.83	5.40	12.3	22.2	30.6	38.0	44.3	49.9	54.7	58.8	
	19	1.61	3.17	6.06	13.9	25.1	34.7	43.0	50.3	56.7	62.3	67.0	
	21	1.78	3.49	6.70	15.3	27.7	38.4	47.6	55.7	62.8	69.0	74.5	79.0
	23	1.97	3.83	7.33	16.8	30.3	42.0	52.1	61.0	68.7	75.5	81.2	86.5
	30	2.57	4.98	9.51	21.7	39.1	53.9	66.7	77.8	87.3	95.7	103.	
	35	2.99	5.78	11.0	25.0	44.9	61.8	76.1	88.4	98.9	108.		
	40	3.40	6.58	12.5	28.3	50.6	69.1	84.9	98.2	109.			
		5	10	20	60	100	140	180	220	260	300	340	380
2.0, No. 160	17	2.14	4.18	7.97	21.1	32.4	42.0	50.6	58.1	64.8	70.5	75.5	
	19	2.40	4.69	8.95	23.9	36.4	47.6	57.2	65.8	73.6	80.3	86.2	
	21	2.65	5.19	9.88	26.4	40.4	52.6	63.4	73.1	81.5	89.1	95.6	102.
	23	2.93	5.68	10.8	28.9	44.2	57.7	69.4	80.0	89.3	95.7	104.	111.
	30	3.81	7.38	14.0	37.3	56.9	73.9	88.9	102.	114.	124.	132.	
	35	4.40	8.57	16.3	42.9	65.4	84.6	101.	116.	129.	139.		
	40	5.00	9.74	18.5	48.5	73.5	94.8	113.	129.	142.			
		5	10	20	40	60	80	100	120	140	180	220	260
2½, No. 200	17	4.18	8.06	15.3	28.3	39.9	50.5	60.2	69.1	77.6	92.3	105.	
	19	4.70	9.05	17.1	31.8	45.0	57.0	68.2	78.2	87.7	105	119.	
	21	5.18	9.99	18.9	35.1	49.7	63.1	75.3	86.6	97.2	116.	132.	146
	23	5.68	10.9	27.0	38.5	54.5	69.0	82.4	94.9	106	127.	144.	160.
	30	7.39	14.2	26.9	49.8	70.3	88.8	106.	122	136.	162.	184.	
	35	8.59	16.4	31.2	57.5	80.9	102.	122.	139.	155.	184.	208.	
	40	9.76	18.7	35.3	64.9	91.0	115.	136.	156.	173.	204.		

* Courtesy Link-Belt Co.

Table 14-85. Service Factors*

Machine load characteristics	10-hr per day service		24-hr per day service	
	Type of drive			
	Motor or turbine	Steam or gas engine (4 cycles or more)	Motor or turbine	Steam or gas engine (4 cycles or more)
Even load	1.00	1.20	1.25	1.50
Moderate uneven load	1.30	1.60	1.65	2.00
Severe uneven load	1.40	1.70	1.80	2.20
Shock load	1.60	1.90	2.00	2.50

* Courtesy Link-Belt Co.

Table 14-86. Maximum Bores with Standard Key Seat When Using Maximum Hub Diameters*†

No. of teeth	Pitch of roller chain, in.									
	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$
9	$\frac{11}{32}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{7}{8}$	1	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{5}{8}$
10	$\frac{7}{16}$	$\frac{19}{32}$	$\frac{3}{4}$	$\frac{27}{32}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{7}{8}$	$2\frac{1}{8}$	$2\frac{1}{2}$	$3\frac{1}{8}$
11	$\frac{1}{2}$	$\frac{23}{32}$	$\frac{31}{32}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{5}{8}$	$2\frac{1}{4}$	$2\frac{5}{8}$	3	$3\frac{3}{4}$
12	$\frac{9}{16}$	$\frac{27}{32}$	$1\frac{1}{8}$	$1\frac{5}{16}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$2\frac{11}{16}$	$3\frac{1}{8}$	$3\frac{3}{8}$	$4\frac{5}{8}$
13	$\frac{11}{16}$	1	$1\frac{9}{32}$	$1\frac{1}{2}$	2	$2\frac{7}{16}$	$3\frac{3}{16}$	$3\frac{5}{8}$	4	$5\frac{1}{4}$
14	$\frac{3}{4}$	$1\frac{3}{16}$	$1\frac{7}{16}$	$1\frac{11}{16}$	$2\frac{1}{4}$	$2\frac{13}{16}$	$3\frac{5}{8}$	$4\frac{1}{16}$	$4\frac{9}{16}$	$5\frac{7}{8}$
15	$\frac{13}{16}$	$1\frac{1}{4}$	$1\frac{5}{8}$	$1\frac{13}{16}$	$2\frac{9}{16}$	$3\frac{1}{4}$	4	$4\frac{11}{16}$	5	$6\frac{11}{16}$
16	$\frac{15}{16}$	$1\frac{3}{8}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$2\frac{7}{8}$	$3\frac{1}{2}$	$4\frac{7}{16}$	$5\frac{1}{16}$	$5\frac{5}{8}$	
17	1	$1\frac{17}{32}$	$1\frac{7}{8}$	$2\frac{5}{16}$	$3\frac{1}{8}$	$3\frac{13}{16}$	$4\frac{11}{16}$	$5\frac{5}{8}$	$6\frac{5}{16}$	
18	$1\frac{1}{16}$	$1\frac{5}{8}$	$2\frac{1}{16}$	$2\frac{1}{2}$	$3\frac{5}{8}$	$4\frac{3}{16}$	$5\frac{3}{16}$	$5\frac{15}{16}$	7	
19	$1\frac{3}{16}$	$1\frac{3}{4}$	$2\frac{1}{4}$	$2\frac{11}{16}$	$3\frac{11}{16}$	$4\frac{9}{16}$	$5\frac{9}{16}$	$6\frac{9}{16}$		
20	$1\frac{5}{16}$	$1\frac{15}{16}$	$2\frac{3}{8}$	$2\frac{13}{16}$	$3\frac{15}{16}$	$4\frac{7}{8}$	$5\frac{15}{16}$	$7\frac{1}{8}$		
21	$1\frac{7}{16}$	$2\frac{1}{16}$	$2\frac{9}{16}$	$3\frac{1}{16}$	$4\frac{1}{8}$	$5\frac{1}{4}$	$6\frac{3}{8}$			
22	$1\frac{9}{16}$	$2\frac{3}{16}$	$2\frac{11}{16}$	$3\frac{1}{4}$	$4\frac{7}{16}$	$5\frac{5}{8}$	$6\frac{13}{16}$			
23	$1\frac{11}{16}$	$2\frac{9}{32}$	$2\frac{15}{16}$	$3\frac{7}{16}$	$4\frac{5}{8}$	$5\frac{13}{16}$	$7\frac{5}{16}$			
	Maximum value of S (center line of shaft to top of key seat)									
9	$\frac{15}{64}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{13}{32}$	$\frac{17}{32}$	$\frac{5}{8}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{1}{4}$	$1\frac{5}{8}$
10	$\frac{9}{32}$	$\frac{23}{64}$	$\frac{15}{32}$	$\frac{35}{64}$	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{3}{16}$	$1\frac{5}{16}$	$1\frac{9}{16}$	$2\frac{5}{16}$
11	$\frac{5}{16}$	$\frac{15}{32}$	$\frac{37}{64}$	$\frac{11}{16}$	$\frac{7}{8}$	1	$1\frac{25}{64}$	$1\frac{41}{64}$	$1\frac{7}{8}$	$2\frac{9}{32}$
12	$\frac{11}{32}$	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{27}{32}$	$1\frac{5}{64}$	$1\frac{5}{16}$	$1\frac{21}{32}$	$1\frac{61}{64}$	$2\frac{1}{8}$	$2\frac{13}{16}$
13	$\frac{7}{16}$	$\frac{21}{32}$	$\frac{13}{16}$	$\frac{31}{32}$	$1\frac{17}{64}$	$1\frac{17}{32}$	$1\frac{31}{32}$	$2\frac{17}{64}$	$2\frac{31}{64}$	$3\frac{1}{4}$
14	$\frac{15}{32}$	$\frac{47}{64}$	$\frac{59}{64}$	$1\frac{3}{32}$	$1\frac{29}{64}$	$1\frac{49}{64}$	$2\frac{1}{4}$	$2\frac{17}{32}$	$2\frac{25}{32}$	$3\frac{11}{16}$
15	$\frac{1}{2}$	$\frac{27}{32}$	$1\frac{1}{32}$	$1\frac{7}{32}$	$1\frac{41}{64}$	2	$2\frac{1}{2}$	$2\frac{55}{64}$	$3\frac{1}{8}$	$4\frac{3}{32}$
16	$\frac{19}{32}$	$\frac{29}{32}$	$1\frac{1}{8}$	$1\frac{23}{64}$	$1\frac{7}{8}$	$2\frac{7}{32}$	$2\frac{3}{4}$	$3\frac{3}{16}$	$3\frac{9}{16}$	
17	$\frac{5}{8}$	$\frac{63}{64}$	$1\frac{15}{64}$	$1\frac{15}{32}$	$2\frac{3}{32}$	$2\frac{7}{16}$	3	$3\frac{15}{32}$	$3\frac{7}{8}$	
18	$\frac{21}{32}$	$1\frac{5}{64}$	$1\frac{21}{64}$	$1\frac{19}{32}$	$2\frac{11}{64}$	$2\frac{5}{8}$	$3\frac{1}{4}$	$3\frac{3}{4}$	$4\frac{1}{4}$	
19	$\frac{23}{32}$	$1\frac{5}{32}$	$1\frac{7}{16}$	$1\frac{23}{32}$	$2\frac{11}{32}$	$2\frac{7}{8}$	$3\frac{1}{2}$	$4\frac{1}{32}$		
20	$\frac{27}{32}$	$1\frac{1}{4}$	$1\frac{9}{16}$	$1\frac{27}{32}$	$2\frac{31}{64}$	$3\frac{3}{32}$	$3\frac{3}{4}$	$4\frac{5}{16}$		
21	$\frac{29}{32}$	$1\frac{21}{64}$	$1\frac{21}{32}$	$1\frac{63}{64}$	$2\frac{21}{32}$	$3\frac{5}{16}$	$3\frac{31}{32}$			
22	$\frac{31}{32}$	$1\frac{13}{32}$	$1\frac{3}{4}$	$2\frac{7}{64}$	$2\frac{53}{64}$	$3\frac{1}{2}$	$4\frac{7}{32}$			
23	$1\frac{1}{32}$	$1\frac{1}{2}$	$1\frac{27}{32}$	$2\frac{15}{64}$	$2\frac{63}{64}$	$3\frac{47}{64}$	$4\frac{15}{32}$			

* Courtesy Link-Belt Co.

† Standard tolerances for bores are as shown below. Unless otherwise specified, the tolerance limits will be observed as standard

Bores, in.	Tolerance limits, in.	
Less than 2	+0.002	−0.000
2 and less than 3	+0.003	−0.000
3 and less than 4	+0.004	−0.000
4 and over.	+0.005	−0.000

usually hardened for durability. It is recommended for most satisfactory service that sprockets have a minimum of 21 teeth, but pinions with as few as 15 teeth are used to meet special conditions. Sprockets are available without hubs, with hubs on one or both sides, split type, demountable-rim type, and in the shear-pin type. To accommodate bore for the shaft, it may at times be necessary to increase the number of teeth in the smaller sprocket (see Table 14-86).

Velocity Ratios. A wide range of ratios is permissible. It is economy not to exceed 6:1, but larger ratios are possible. Where a larger ratio is required, a two-step drive is recommended.

Chain Care. Lubrication of chains should be sufficient effectively to oil the contacting surfaces of the chain joints and sprockets. Experience indicates that the desirable method is influenced by chain speeds and power transmitted (see Table 14-87). Consideration should be given to the use of chain casings, properly designed. Open running drives should be demounted and cleaned regularly with solvents, care being taken to reoil bearing surfaces copiously before reinstalling. Proper chain tension and alignment of sprockets are necessary for satisfactory chain life.

Table 14-87*

Chain speed	Lubrication method	
	Automatic	Manual
Under 600 fpm	Bath or splash	Oil cup or brush
600–1,500 fpm	Bath or splash	Oil cup (rapid drip)
Over 1,500 fpm	Pump or disk	Not recommended

* Courtesy Link-Belt Co.

SILENT CHAIN

Manufacture. A silent chain consists of a number of flat leaves contoured to provide teeth at both ends and connected by various types of pins to form a flexible, continuous rack in a wide variation of widths. Precision tools are used in the manufacture to maintain close control over size of hole, contour, pins, and bushings, where used, ensuring accurate pitch. Links are made of heat-treated high-carbon or alloy

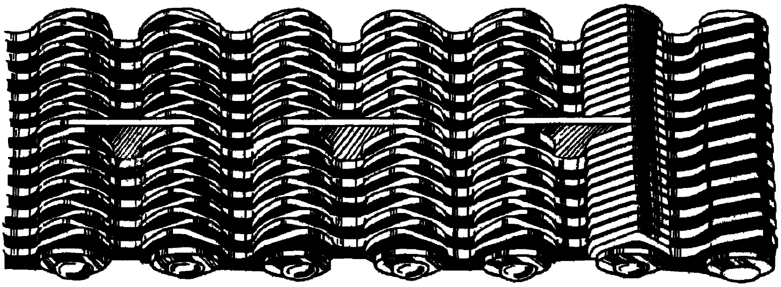


FIG. 14-31. Middle-guide silent chain.

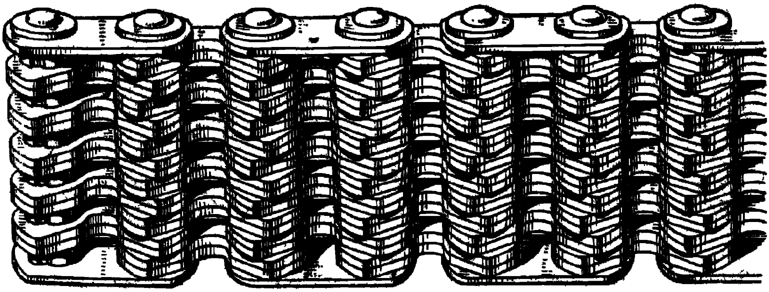


FIG. 14-32. Side-guide silent chain.

steel. Pins may consist of a single pin, bushed or unbushed, or a pair of pins with rocker faces. All joint parts are hardened for wear resistance.

Chain Guiding on Sprockets. Two classes of arrangements are used:

1. Flange links as a component part of the chain, known as the middle-guide type (Fig. 14-31) and (2) the side-guide type (Fig. 14-32)

Selection of Chain Size. The revolutions per minute of the faster running shaft limits the maximum pitch of chain that should be employed. Select a trial pitch size from Table 14-88.

Table 14-87*a*. Oil Viscosity for Silent Chain

Ambient temp., deg F	*SAE viscosity No.*
20–40	20
40–100	30
100–140	50

Where quiet operation is essential, use should be made of smaller pitches and wider chains, conservative ratings, and not less than 25 teeth in the faster running sprocket. The approximate maximum width of chain that may be used is 16 times the pitch.

Working Load Capacities. The working load capacity, based on durability and ultimate strength, is governed by horsepower, chain speed, and operating conditions. After selection of trial pitch size, reference should be made to Table 14-90, which gives horsepower ratings for various pitches of chains per inch of width. This rating is governed by a combination of rpm and number of teeth in small sprocket. The horsepower values listed in this table are based on 10-hr-per-day service, uniform loads, and good lubrication, where there is little shock or sharp vibration in speed throughout a single revolution of the driven wheel. This rating can be adjusted to other periods of service and load characteristics by multiplying the known horsepower to be transmitted by the factors in Table 14-85. Small sprockets must be checked to determine that maximum bore size will accommodate shaft necessary to transmit required horsepower.

Table 14-88. Silent Chain, Speed and Pitch Sizes*

Rpm of small wheel	*Trial pitch*
4,000–6,000	3/8
3,500–4,000	1/2
2,500–3,500	5/8
2,000–2,500	3/4
1,500–2,000	1
1,200–1,500	1 1/4
900–1,200	1 1/2
Below 900	2

* Courtesy Link-Belt Co.

Chain Speeds. For long life and low maintenance silent chains should be operated at maximum speeds from 1,400 to 3,500 fpm, depending on pitch and number of teeth in small sprocket. Silent chain drives under certain operating conditions are functioning successfully at chain speeds in excess of 5,000 fpm.

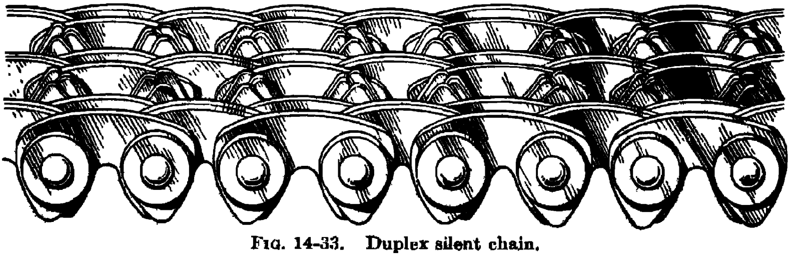


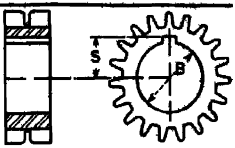
Fig. 14-33. Duplex silent chain.

Sprockets. All sprockets are of the cut-tooth type, accurately machined from gray iron, cast steel, and forged or bar-stock steel with pinions hardened and the larger sprockets either hardened or unhardened. It is recommended for most satisfactory service that sprockets have a minimum of 21 teeth (25 teeth where quiet operation is essential), but pinions with as low as 17 teeth are used. Sprockets are available without hubs, with hubs on one or both sides, in the split type, and in the shear-pin

Table 14-89. Maximum Bores for Silent Chain Wheels*

(Unless otherwise specified, the following tolerance limits will be considered as standard.)

Bores, in.	Limits, in.	Bores, in.	Limits, in.
Less than 1	±0.001	3-4	+0.004, -0.000
1-2	+0.002, -0.000	4 and over	+0.005, -0.000
2-3	+0.003, -0.000		



(Dimensions in inches)

No. of teeth	Pitch of pinion, in.									
	3/16	3/8	1/2	5/8	3/4	1	1 1/4	1 1/2	2	2 1/2
	Maximum advisable value of *S*, in. (approx)									
15	3/16	1/2	3/4	7/8	1	1 1/2	1 3/4	2 1/16	2 7/8	3 1/2
16	7/32	9/16	13/16	31/32	1 1/8	1 5/8	1 15/16	2 5/16	3 3/16	3 7/8
17	1/4	5/8	7/8	1 1/16	1 1/4	1 3/4	2 1/8	2 9/16	3 1/2	4 1/4
18	9/32	11/16	15/16	1 5/32	1 3/8	1 7/8	2 5/16	2 13/16	3 13/16	4 5/8
19	5/16	3/4	1	1 1/4	1 1/2	2	2 1/2	3 1/16	4 1/8	5
20	11/32	13/16	1 3/32	1 11/32	1 5/8	2 3/16	2 11/16	3 9/32	4 7/16	5 3/8
21	3/8	7/8	1 3/16	1 7/16	1 3/4	2 3/8	2 7/8	3 1/2	4 3/4	5 3/4
22	13/32	15/16	1 1/4	1 17/32	1 7/8	2 17/32	3 1/8	3 3/4	5 1/16	6 1/8
23	7/16	1	1 5/16	1 5/8	2	2 11/16	3 3/8	4	5 3/8	6 1/2
24	15/32	1 1/16	1 13/32	1 3/4	2 1/8	2 27/32	3 9/16	4 1/4	5 11/16	7
25	1/2	1 1/8	1 1/2	1 7/8	2 1/4	3	3 3/4	4 1/2	6	7 1/2
26	17/32	1 3/16	1 9/16	1 31/32	2 11/32	3 5/32	3 15/16	4 23/32	6 5/16	7 7/8
27	9/16	1 1/4	1 5/8	2 1/16	2 7/16	3 5/16	4 1/8	4 15/16	6 5/8	8 1/4
	Maximum advisable bore *B*									
15	1/4	7/8	1 1/8	1 1/2	1 5/8	2 1/4	2 3/4	3 3/8	4 1/4	5 3/4
16	5/16	15/16	1 1/4	1 5/8	1 13/16	2 1/2	3 1/16	3 3/4	4 7/8	6 3/8
17	3/8	1 1/16	1 3/8	1 3/4	2	2 3/4	3 3/8	4 1/8	5 1/2	7
18	7/16	1 1/8	1 1/2	1 7/8	2 3/16	3	3 11/16	4 1/2	6 1/8	7 5/8
19	1/2	1 1/4	1 5/8	2	2 3/8	3 1/4	4	4 7/8	6 3/4	8 1/4
20	9/16	1 5/16	1 3/4	2 1/8	2 9/16	3 1/2	4 1/4	5 5/16	7 1/8	8 7/8
21	5/8	1 3/8	1 7/8	2 1/4	2 3/4	3 3/4	4 1/2	5 1/2	7 3/4	9 3/4
22	11/16	1 1/2	2	2 7/16	3	4 1/16	5	6	8 3/8	10 3/8
23	3/4	1 5/8	2 1/8	2 5/8	3 1/4	4 3/8	5 1/2	6 1/2	9	11
24	13/16	1 11/16	2 1/4	2 13/16	3 7/16	4 9/16	5 3/4	6 7/8	9 1/2	12
25	7/8	1 3/4	2 3/8	3	3 5/8	4 3/4	6	7 1/4	10	13
26	15/16	1 7/8	2 1/2	3 1/8	3 3/4	5 1/16	6 3/8	7 11/16	10 5/8	13 1/2
27	1	2	2 5/8	3 1/4	3 7/8	5 3/8	6 3/4	8 1/8	11 1/4	14

* Courtesy Link-Belt Co.

type. To accommodate bore for the shaft, it may at times be necessary to increase the number of teeth in the smaller sprocket (see Table 14-89).

Velocity Ratios. The most desirable ratios for this chain range from 1.1 to 6:1. Large-ratio drives are less economical in both first cost and maintenance than comparable double-reduction drives would be.

Duplex or Double-back Type. The duplex or double-back type unit (Fig. 14-33) is constructed of silent chain parts so assembled that there are driving teeth on both sides of the chain. Sprockets can be driven from both its surfaces. It is particularly

Table 14-90. Horsepower per Inch of Width*

(Maximum width of chain, 4 in.; pitch of chain, ⅜ in.)

No. of teeth, small sprocket	Revolutions per minute, small sprocket												
	100	500	1,000	1,200	1,500	1,800	2,000	2,500	3,000	3,500	4,000	5,000	6,000
17†	0.37	1.7	3.7	3.9	4.2	5.2	5.5	6.3	6.8	7.0	7.0		
19†	0.42	2.0	3.8	4.3	5.2	5.9	6.3	7.3	7.9	8.3	8.4	7.8	
21	0.46	2.2	4.1	4.8	5.8	6.6	7.2	8.3	9.1	9.6	9.9	9.5	8.2
23	0.50	2.4	4.5	5.3	6.4	7.4	8.0	9.2	10	11	11	11	9.5
25	0.55	2.6	4.9	5.8	7.0	8.1	8.8	10	11	12	12	12	11
27	0.59	2.8	5.4	6.3	7.6	8.8	9.5	11	12	13	14	14	13
29	0.64	3.0	5.8	6.8	8.2	9.5	10.3	12	13	14	15	15	14
31	0.68	3.3	6.2	7.3	8.8	10	11	13	14	15	16	16	15
33	0.72	3.5	6.6	7.8	9.4	11	12	14	15	17	17	17	16
35	0.77	3.7	7.0	8.3	10	12	13	15	16	18	18	18	17
37	0.82	3.9	7.3	8.7	11	12	13	16	17	19	19	19	
40	0.9	4.2	8.1	9.5	12	13	14	17	19	20	21	21	
45	1.0	4.8	9.1	10	13	15	16	19	21	22	23		
50	1.1	5.3	10	12	14	16	18	21	23	24			
	Bath or splash, oil cup, or brush			Disk or circulating pump				Consult manufacturer's engineering department for proper method of lubrication					

(Maximum width of chain, 7 in.; pitch of chain, ½ in.)

No. of teeth, small sprocket	Revolutions per minute, small sprocket										
	100	500	700	1,000	1,200	1,800	2,000	2,500	3,000	3,500	4,000
17†	0.66	3	4	5	6	8	9	9	9	9	
19†	0.74	3	4	6	7	9	10	11	11	11	
21	0.81	4	5	7	8	11	11	12	13	13	
23	0.89	4	6	8	9	12	13	14	15	15	14
25	0.97	4	6	8	10	13	14	16	17	17	16
27	1.0	5	7	9	10	14	15	17	19	19	18
29	1.1	5	7	10	11	15	17	19	20	20	20
31	1.2	6	8	10	12	17	18	20	22	22	22
33	1.3	6	8	11	13	18	19	22	23	24	23
35	1.4	6	9	12	14	19	20	23	25	26	24
37	1.5	7	9	13	15	20	21	24	26	26	
40	1.6	7	10	14	16	22	23	26	28	28	
45	2	8	11	15	18	24	24	29	31		
50	2	9	12	17	20	27	29	32			
	Bath or splash, oil cup, or brush		Disk or circulating pump				Consult manufacturer's engineering department for proper method of lubrication				

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* Courtesy Link-Belt Co.

† For best results, smaller sprocket should have at least 21 teeth.

adaptable where several shafts are required to operate in different directions of rotation or where it is desired to employ a chain take-up device on the back of the chain.

CHAIN-TYPE SELECTION

Chains of the different types to fit varying needs are grouped in Table 14-91 to facilitate a satisfactory selection based on a comparison of the design characteristics.

Table 14-20. Horsepower per Inch of Width* (*Cont.*)
(Maximum width of chain, 8 in.; pitch of chain, ⅝ in.)

No. of teeth, small sprocket	Revolutions per minute, small sprocket									
	100	500	700	1,000	1,200	1,800	2,000	2,500	3,000	3,500
17†	1	5	6	8	9	11	12	11		
19†	1.1	5	7	10	11	13	14	14		
21	1.3	6	8	10	12	15	16	16	16	
23	1.4	6	9	12	13	17	18	19	18	
25	1.5	7	9	13	15	19	20	21	21	19
27	1.6	8	10	14	16	21	22	23	23	21
29	1.7	8	11	15	17	22	24	25	25	23
31	1.9	9	12	16	18	24	25	27	27	25
33	2	9	13	17	20	26	27	29	29	27
35	2.1	10	13	18	21	27	29	31	31	28
37	2.2	10	14	19	22	29	31	34	33	
40	2.4	11	15	21	24	31	33	35		
45	2.7	13	17	23	27	35	37			
50	3	14	19	26	30	38	40			
	Bath or splash, oil cup, or brush		Disk or circulating pump		Consult manufacturer's engineering department for proper method of lubrication					

(Maximum width of chain, 10 in.; pitch of chain, ¾ in.)

No. of teeth, small sprocket	Revolutions per minute, small sprocket								
	100	500	700	1,000	1,200	1,500	1,800	2,000	2,500
17†	1.5	6.5	8.6	11	12	13	14	14	
19†	1.6	7.4	10	12	14	16	17	17	
21	1.8	8	11	14	16	18	19	20	19
23	2.0	9	12	16	18	20	22	22	22
25	2.2	10	13	17	20	23	25	25	24
27	2.3	11	14	19	22	25	27	28	28
29	2.5	12	16	21	24	27	29	30	30
31	2.7	12	17	22	25	29	32	33	33
33	2.9	13	18	24	27	31	34	35	35
35	3.0	14	19	25	29	33	36	37	37
37	3.2	15	20	27	31	35	38	39	39
40	3.5	16	22	29	33	38	41	42	42
45	3.9	18	24	32	37	42	45	46	
50	4.3	20	27	36	41	46	49		
	Bath or splash, oil cup, or brush		Disk or circulating pump		Consult manufacturer's engineering department for proper method of lubrication				

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* Courtesy Link-Belt Co.

† For best results, smaller sprocket should have at least 21 teeth.

The numerals indicate order of preference, *i.e.*, 1 is most desirable, etc., and the x's indicate recommendations.

Chain Centers and Length

In the following computations of chain length, applicable to all types of chains, an even number of pitches is desirable. However, offset coupling links may be furnished if necessary. The center distance for any drive must be more

Table 14-90. Horsepower per Inch of Width* (*Cont.*)

(Maximum width of chain, 14 in.; pitch of chain, 1 in.)

No. of teeth, small sprocket	Revolutions per minute, small sprocket										
	100	200	300	400	500	700	1,000	1,200	1,500	1,800	2,000
17†	3	5	7	9	11	14	17	18			
19†	3	6	8	10	12	16	20	21	22		
21	3	6	9	12	14	18	23	25	26	26	
23	3	7	10	13	15	20	25	28	30	30	
25	4	7	11	14	17	22	28	31	33	33	33
27	4	8	12	15	19	24	31	34	37	37	36
29	4	9	13	16	20	26	33	37	40	41	40
31	5	9	13	18	22	28	36	40	43	44	43
33	5	10	14	19	23	30	39	43	47	47	46
35	5	10	15	20	24	32	41	45	49	50	49
37	5.4	11	16	21	26	34	43	48	52	53	
40	6	12	18	23	28	36	47	52	56		
45	7	13	20	25	31	41	52	57	61		
50	8	15	22	28	34	45	57	62			
	Bath or splash, oil cup, or brush			Disk or circulating pump				Consult manufacturer's engineering department for proper method of lubrication			

(Maximum width of chain, 20 in.; pitch of chain, $1\frac{1}{4}$ in.)

No. of teeth, small sprocket	Revolutions per minute, small sprocket										
	100	200	300	400	500	600	700	800	1,000	1,200	1,500
19†	4.5	8	12	16	19	21	23	25	27	28	
21	5	9	14	18	21	24	26	29	32	33	
23	5.5	10	15	19	23	27	29	32	36	37	37
25	6	11	16	21	25	29	32	35	40	42	42
27	6.4	12	18	23	28	32	35	39	43	46	46
29	6.9	13	19	25	30	34	38	42	47	50	51
31	7.4	14	21	27	32	37	41	45	51	54	55
33	7.9	15	22	28	34	39	44	48	55	58	59
35	8.4	16	23	30	36	42	47	51	58	62	62
37	9	17	24	32	38	44	50	54	61	65	
40	9.6	19	27	35	42	48	54	59	66	70	
45	10.7	21	30	39	47	54	60	65	73		
50	12	23	34	43	52	59	66	72	80		
	Bath or splash, oil cup, or brush			Disk or circulating pump			Consult manufacturer's engineering department for proper method of lubrication				

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* Courtesy Link-Belt Co.

† For best results, smaller sprocket should have at least 21 teeth.

than half the sum of the diameters of the two sprockets in order that the teeth may clear. The pitch diameters for the sprockets may be computed from the following formula, or from Table 14-92:

$$\text{Pitch diameter} = \text{pitch} \times \operatorname{cosec} \frac{180}{\text{No. of teeth in sprocket}}$$

Table 14-90. Horsepower per Inch of Width* (*Cont.*)

(Maximum width of chain, 24 in.; pitch of chain, 1½ in.)

No. of teeth, small sprocket	Revolutions per minute, small sprocket										
	100	200	300	400	500	600	700	800	900	1,000	1,200
19†	6.4	12	17	22	25	28	31	32	33	34	
21	7	13	19	24	29	32	35	37	39	39	
23	8	15	21	27	32	36	39	42	44	45	44
25	8	16	23	30	35	40	44	47	49	52	51
27	9	18	25	32	38	43	48	51	54	56	56
29	10	19	27	35	41	47	52	56	59	60	61
31	11	20	29	37	44	51	56	60	63	65	66
33	11	22	31	40	47	54	60	64	68	70	71
35	12	23	33	42	50	57	63	68	72	74	75
37	13	24	35	47	53	61	67	72	77	79	
40	14	26	38	53	58	66	72	78	84		
45	15	30	43	54	65	74	81	86	90		
50	17	33	47	60	71	81	89	94			
Bath or splash, oil cup, or brush			Disk or circulating pump		Consult manufacturer's engineering department for proper method of lubrication						

(Maximum width of chain, 30 in.; pitch of chain, 2 in.)

No. of teeth, small sprocket	Revolutions per minute, small sprocket								
	100	200	300	400	500	600	700	800	900
19†	11	21	29	35	40	43	45		
21	12.5	23	32	40	42	50	52		
23	13.5	26	36	44	51	56	59	60	
25	14	28	39	49	56	62	66	68	68
27	16	30	43	53	62	68	73	75	75
29	17	33	46	58	67	74	79	82	82
31	18	35	50	62	72	80	85	88	88
33	20	37	53	66	77	85	91	94	94
35	21	40	57	70	82	91	97	100	100
37	22	42	60	74	88	99	102	105	
40	24	46	65	81	94	103	110	113	
45	27	51	72	90	105	115	121		
50	30	57	80	100	115	125			
Bath or splash, oil cup, or brush		Disk or circulating pump		Consult manufacturer's engineering department for proper method of lubrication					

* Courtesy Link-Belt Co.

† For best results, smaller sprocket should have at least 21 teeth.

In order to determine the exact center distance required, it is first necessary to compute the correct number of pitches for the approximate desired center distance. The following three steps are necessary to arrive at the solution of the number of pitches required:[1]

1. Subtract the pitch diameter of the smaller sprocket from the pitch diameter of the larger sprocket.

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[1] This method takes into consideration the fact that the length of chain around a sprocket is less than the circumference of the pitch circle because the chain links describe a series of chords within the pitch circle.

Table 14-91*

Characteristics	Types of chain			
	Silent	Finished steel roller	Steel roller	Malleable or pressed steel
For use in low speeds, such as elevator and conveyor chain, where lubrication and cleanliness are impractical†	..	2	1	1
Light and heavy power-transmission primary drive	x	x		
Power-transmission secondary drive	..	x	x	x
Light conveyor and elevator work as conveyor or elevator chain under clean conditions not necessarily lubricated†	..	x	x	x
Heavy conveyor and elevator chains	..	..	x	x
Speeds up to 400 fpm	x	x	x	x
Speeds up to 700 fpm	x	x	x	
Speeds over 700 fpm	x	x		
Ratios up to 5:1	x	x	x	x
Ratios up to 10:1	x	x		
Ranking as to quietest, smoothest, and most efficient drive at equal speeds	1	2	3	4
Sprockets				
Cut tooth	x	x		
Cast tooth	..	..	x	x

* Courtesy Link-Belt Co.

† Favors malleable and rough-finished chain because of larger bearing area in joint.

Table 14-92. Pitch Diameter for a Pitch of 1 in.*

(Pitch diameters for other pitches are in proportion to the pitches)

No. of teeth	Pitch diam	No. of teeth	Pitch diam	No. of teeth	Pitch diam	No. of teeth	Pitch diam	No. of teeth	Pitch diam	No. of teeth	Pitch diam
17	5.442	41	13.064	66	21.016	91	28.972	116	36.929	141	44.885
18	5.759	42	13.382	67	21.335	92	29.290	117	37.247	142	45.203
19	6.076	43	13.700	68	21.653	93	29.608	118	37.565	143	45.521
20	6.393	44	14.018	69	21.971	94	29.926	119	37.883	144	45.840
		45	14.336	70	22.289	95	30.245	120	38.201	145	46.158
21	6.710										
22	7.027	46	14.654	71	22.607	96	30.563	121	38.519	146	46.477
23	7.344	47	14.972	72	22.926	97	30.881	122	38.837	147	46.796
24	7.661	48	15.290	73	23.244	98	31.199	123	39.156	148	47.114
25	7.979	49	15.608	74	23.562	99	31.518	124	39.475	149	47.432
		50	15.926	75	23.880	100	31.836	125	39.794	150	47.750
26	8.296										
27	8.614	51	16.244	76	24.198	101	32.154	126	40.112		
28	8.932	52	16.562	77	24.517	102	32.473	127	40.430		
29	9.249	53	16.880	78	24.835	103	32.791	128	40.748		
30	9.567	54	17.198	79	25.153	104	33.109	129	41.066		
		55	17.517	80	25.471	105	33.427	130	41.384		
31	9.885										
32	10.202	56	17.835	81	25.790	106	33.746	131	41.702		
33	10.520	57	18.153	82	26.108	107	34.064	132	42.020		
34	10.838	58	18.471	83	26.426	108	34.382	133	42.338		
35	11.156	59	18.789	84	26.744	109	34.701	134	42.656		
		60	19.107	85	27.063	110	35.019	135	42.975		
36	11.474										
37	11.792	61	19.426	86	27.381	111	35.337	136	43.293		
38	12.110	62	19.744	87	27.699	112	35.655	137	43.611		
39	12.428	63	20.062	88	28.017	113	35.974	138	43.930		
40	12.746	64	20.380	89	28.335	114	36.292	139	44.249		
		65	20.698	90	28.654	115	36.610	140	44.567		

* Courtesy Link-Belt Co.

2. Divide this difference by twice the desired chain centers.
3. Find in Col. *A* in Table 14-93 the value next larger than the result obtained by step 2, and use the corresponding values (in the same line) from Cols. *B*, *C*, and *D* to solve the following equations:

Centers

$B \times$ pitch = no. of pitches between sprockets
$C \times T_1$ = no. of pitches on smaller sprocket
$D \times T_2$ = no. of pitches on larger sprocket

where T_1 = no. of teeth in smaller sprocket
T_2 = no. of teeth in larger sprocket

The sum of these three calculations in step 3 is the number of pitches of chain required for the center distance used. However, if this value works out to a fractional number of pitches, it is necessary to perform additional calculation to obtain the proper center distance, except when the fractional number of pitches is slightly less than the nearest whole number, which difference will take care of desired coupling slack. So select the nearest whole number to the calculated number and proceed as follows:

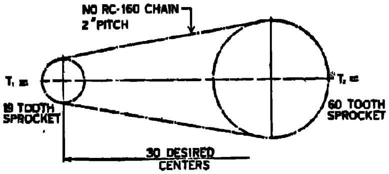
Table 14-93

A	*B*	*C*	*D*	*A*	*B*	*C*	*D*	*A*	*B*	*C*	*D*	*A*	*B*	*C*	*D*
0.00000	2.0000	0.5000	0.5000	0.19937	1.9599	0.4361	0.5639	0.39073	1.8410	0.3722	0.6278	0.56641	1.6483	0.3083	0.6917
0.00436	2.0000	0.4986	0.5014	0.20364	1.9581	0.4347	0.5653	0.39474	1.8376	0.3708	0.6292	0.57000	1.6433	0.3069	0.6931
0.00873	1.9999	0.4972	0.5028	0.20791	1.9563	0.4333	0.5667	0.39875	1.8341	0.3694	0.6306	0.57358	1.6383	0.3055	0.6945
0.01309	1.9998	0.4958	0.5042	0.21218	1.9545	0.4319	0.5681	0.40275	1.8306	0.3681	0.6319	0.57715	1.6333	0.3042	0.6958
0.01745	1.9997	0.4944	0.5056	0.21644	1.9526	0.4306	0.5694	0.40674	1.8271	0.3667	0.6333	0.58070	1.6282	0.3028	0.6972
0.02181	1.9995	0.4931	0.5069	0.22070	1.9507	0.4292	0.5708	0.41072	1.8235	0.3653	0.6347	0.58425	1.6231	0.3014	0.6986
0.02618	1.9993	0.4917	0.5083	0.22495	1.9487	0.4278	0.5722	0.41469	1.8199	0.3639	0.6361	0.58779	1.6180	0.3000	0.7000
0.03054	1.9991	0.4903	0.5097	0.22920	1.9468	0.4264	0.5736	0.41866	1.8163	0.3625	0.6375	0.59131	1.6129	0.2986	0.7014
0.03490	1.9988	0.4889	0.5111	0.23345	1.9447	0.4250	0.5750	0.42262	1.8126	0.3611	0.6389	0.59482	1.6077	0.2972	0.7028
0.03926	1.9986	0.4875	0.5125	0.23769	1.9427	0.4236	0.5764	0.42657	1.8089	0.3597	0.6403	0.59832	1.6025	0.2958	0.7042
0.04362	1.9981	0.4861	0.5139	0.24192	1.9406	0.4222	0.5778	0.43051	1.8052	0.3583	0.6417	0.60182	1.5973	0.2945	0.7055
0.04798	1.9977	0.4847	0.5153	0.24615	1.9385	0.4208	0.5792	0.43445	1.8014	0.3569	0.6431	0.60529	1.5920	0.2932	0.7068
0.05234	1.9972	0.4833	0.5167	0.25039	1.9363	0.4194	0.5806	0.43837	1.7976	0.3556	0.6444	0.60876	1.5867	0.2917	0.7083
0.05669	1.9968	0.4819	0.5181	0.25460	1.9341	0.4181	0.5819	0.44229	1.7937	0.3542	0.6458	0.61222	1.5814	0.2903	0.7097
0.06105	1.9963	0.4806	0.5194	0.25882	1.9319	0.4167	0.5833	0.44620	1.7899	0.3528	0.6472	0.61566	1.5760	0.2889	0.7111
0.06540	1.9957	0.4792	0.5208	0.26303	1.9296	0.4153	0.5847	0.45010	1.7860	0.3514	0.6486	0.61909	1.5706	0.2875	0.7125
0.06976	1.9951	0.4778	0.5222	0.26724	1.9273	0.4139	0.5861	0.45399	1.7820	0.3500	0.6500	0.62251	1.5652	0.2861	0.7139
0.07411	1.9945	0.4763	0.5237	0.27144	1.9249	0.4125	0.5875	0.45787	1.7780	0.3486	0.6514	0.62592	1.5598	0.2847	0.7153
0.07846	1.9938	0.4750	0.5250	0.27564	1.9225	0.4111	0.5889	0.46175	1.7740	0.3472	0.6528	0.62932	1.5543	0.2833	0.7167
0.08281	1.9931	0.4736	0.5264	0.27983	1.9201	0.4097	0.5903	0.46561	1.7700	0.3458	0.6542	0.63271	1.5488	0.2819	0.7181
0.08716	1.9924	0.4722	0.5278	0.28402	1.9176	0.4083	0.5917	0.46947	1.7659	0.3445	0.6555	0.63608	1.5432	0.2805	0.7195
0.09150	1.9916	0.4708	0.5292	0.28820	1.9151	0.4070	0.5930	0.47332	1.7618	0.3431	0.6569	0.63944	1.5377	0.2792	0.7208
0.09585	1.9908	0.4694	0.5306	0.29237	1.9126	0.4056	0.5944	0.47716	1.7576	0.3417	0.6583	0.64279	1.5321	0.2778	0.7222
0.10019	1.9899	0.4681	0.5319	0.29654	1.9100	0.4042	0.5958	0.48099	1.7535	0.3403	0.6597	0.64612	1.5265	0.2764	0.7236
0.10453	1.9890	0.4667	0.5333	0.30071	1.9074	0.4028	0.5972	0.48481	1.7492	0.3389	0.6611	0.64945	1.5208	0.2750	0.7250
0.10887	1.9881	0.4653	0.5347	0.30486	1.9048	0.4014	0.5986	0.48862	1.7450	0.3375	0.6625	0.65276	1.5151	0.2736	0.7264
0.11320	1.9871	0.4639	0.5361	0.30902	1.9021	0.4000	0.6000	0.49242	1.7407	0.3361	0.6639	0.65606	1.5094	0.2722	0.7278
0.11754	1.9861	0.4625	0.5375	0.31316	1.8994	0.3986	0.6014	0.49622	1.7364	0.3347	0.6653	0.65935	1.5037	0.2708	0.7292
0.12187	1.9851	0.4611	0.5389	0.31730	1.8966	0.3972	0.6028	0.50000	1.7321	0.3333	0.6667	0.66262	1.4979	0.2694	0.7306
0.12620	1.9840	0.4597	0.5403	0.32144	1.8939	0.3958	0.6042	0.50377	1.7277	0.3320	0.6680	0.66588	1.4921	0.2680	0.7320
0.13053	1.9829	0.4583	0.5417	0.32557	1.8910	0.3944	0.6056	0.50754	1.7233	0.3305	0.6695	0.66913	1.4863	0.2667	0.7333
0.13485	1.9817	0.4569	0.5431	0.32969	1.8882	0.3931	0.6069	0.51129	1.7188	0.3292	0.6708	0.67237	1.4804	0.2653	0.7347
0.13917	1.9805	0.4555	0.5445	0.33381	1.8853	0.3917	0.6083	0.51504	1.7143	0.3278	0.6722	0.67559	1.4746	0.2639	0.7361
0.14349	1.9793	0.4542	0.5458	0.33792	1.8824	0.3903	0.6097	0.51877	1.7098	0.3264	0.6736	0.67880	1.4686	0.2625	0.7375
0.14781	1.9780	0.4528	0.5472	0.34202	1.8794	0.3889	0.6111	0.52250	1.7053	0.3250	0.6750	0.68200	1.4627	0.2611	0.7389
0.15212	1.9767	0.4514	0.5486	0.34612	1.8764	0.3875	0.6125	0.52621	1.7007	0.3236	0.6764	0.68518	1.4567	0.2603	0.7397
0.15643	1.9754	0.4500	0.5500	0.35021	1.8733	0.3861	0.6139	0.52992	1.6961	0.3222	0.6778	0.68835	1.4507	0.2583	0.7417
0.16074	1.9740	0.4486	0.5514	0.35429	1.8703	0.3847	0.6153	0.53361	1.6915	0.3208	0.6792	0.69151	1.4447	0.2569	0.7431
0.16505	1.9726	0.4472	0.5528	0.35837	1.8672	0.3833	0.6167	0.53730	1.6868	0.3194	0.6806	0.69466	1.4387	0.2556	0.7444
0.16935	1.9711	0.4458	0.5542	0.36244	1.8640	0.3819	0.6181	0.54097	1.6821	0.3180	0.6820	0.69779	1.4326	0.2542	0.7458
0.17365	1.9696	0.4444	0.5556	0.36650	1.8608	0.3806	0.6194	0.54464	1.6773	0.3166	0.6834	0.70091	1.4265	0.2528	0.7472
0.17794	1.9681	0.4430	0.5570	0.37056	1.8576	0.3792	0.6208	0.54829	1.6726	0.3153	0.6847	0.70401	1.4204	0.2514	0.7486
0.18224	1.9665	0.4416	0.5583	0.37461	1.8544	0.3778	0.6222	0.55194	1.6678	0.3139	0.6861	0.70711	1.4142	0.2500	0.7500
0.18652	1.9649	0.4403	0.5597	0.37865	1.8511	0.3764	0.6236	0.55557	1.6629	0.3125	0.6875				
0.19081	1.9633	0.4389	0.5611	0.38268	1.8478	0.3750	0.6250	0.55919	1.6581	0.3111	0.6889				
0.19509	1.9616	0.4375	0.5625	0.38671	1.8444	0.3736	0.6264	0.56280	1.6532	0.3097	0.6903				

For All Types of Chains

If the Number of Pitches Selected Is Less than the Calculated Length. Subtract the number of pitches selected from the fractional number of pitches. The result is then multiplied by the pitch, and to this is added the amount of coupling slack desired. Divide this sum by value B, and subtract the result from the center distance used in the original calculations (refer to Example 1). Use an even number of pitches where

Example 1. When Centers Must Be Decreased to Suit Chain Length.



T_2 = 60-tooth sprocket, pitch diameter = 38.215 in.
T_1 = 19-tooth sprocket, pitch diameter = 12.151 in.
26.064 in.

$$\frac{26.064 \text{ in.}}{2 \times 30} = 0.43440$$

Then from Table 14-93,

Value A = 0.43445
Value B = 1.8014 × (30/2) = 27.021
Value C = 0.3569 × 19 = 6.781
Value D = 0.6431 × 60 = 38.586/72.388 pitches required for 30-in. centers

Specify 72 pitches, assuming that $\frac{1}{16}$-in. coupling slack is desired. Decrease center distance by

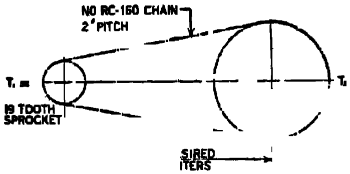
$$\begin{array}{l} 72.388 \\ \underline{72.000} \\ 0.388 \times 2 = 0.776 \text{ in.} \\ \qquad\qquad + \underline{0.062 \text{ in.}} \text{ slack for coupling} \\ \qquad\qquad\quad 0.838 \text{ in.} \\ 0.838 \text{ in.} \div 1.8014 = 0.465 \text{ in.} \end{array}$$

Then the correct center distance for this drive is

$$30.000 \text{ in.} - 0.465 \text{ in.} = 29.535 \text{ in.—say } 29\tfrac{17}{32} \text{ in.}$$

If the Number of Pitches Selected Is More than the Calculated Length. Subtract the fractional number of pitches calculated from the number of pitches selected, and multiply by the pitch. From this subtract the amount of coupling slack desired, and divide by value B. Add the result to the center distance used in the original calculations (refer to Example 2).

Example 2. When Centers Must Be Increased to Suit Chain Length.



T_2 = 60-tooth sprocket, pitch diameter = 38.215 in.
T_1 = 19-tooth sprocket, pitch diameter = 12.151 in.
26.064 in.

$$\frac{26.064 \text{ in.}}{2 \times 29} = 0.44938$$

Then from Table 14-93

Value A = 0.45010
Value B = 1.7860 × (29/2) = 25.897
Value C = 0.3514 × 19 = 6.677
Value D = 0.6486 × 60 = 39.916/71.490 pitches required for 29-in. centers

Specify 72 pitches, assuming that $\frac{1}{16}$-in. coupling slack is desired. Increase center distance by

72.000
71.490
0.510 × 2 in. = 1.020 in.
−0.062 in. slack for coupling
0.958 in.
0.958 in. ÷ 1.7860 = 0.536 in.

Then the correct center distance for this drive is

29.000 in. + 0.536 in. = 29.536 in.—say $29\frac{17}{32}$ in.

Relative Position of Sprockets

Careful attention should be given to the relative positions of shafts. Satisfactory operation can be secured with the drive at any angle to the horizontal if proper consideration is given to tension adjustment (see Fig. 14-34). Certain positions require less care and attention and are therefore less apt to cause trouble.

Tension-adjusting Devices

Means for adjusting tension will prolong chain life particularly when the position of the drive approaches the vertical. On positions approaching the horizontal, the chain itself acts as a tensioner. Regardless of position, a drive subject to pulsations and shocks requires some form of adjustment. The methods employed are either the standard slide rails of the motor or an idler sprocket mounted for automatic or manual adjustment. It is essential to locate idler sprockets on the slack or non-load-carrying side of the chain and preferably as close as possible to the driving sprocket. All idler arrangements should provide for no less than three teeth of the idler being in contact with the chain. Installations of both manual and automatic idlers are shown by Fig. 14-35.

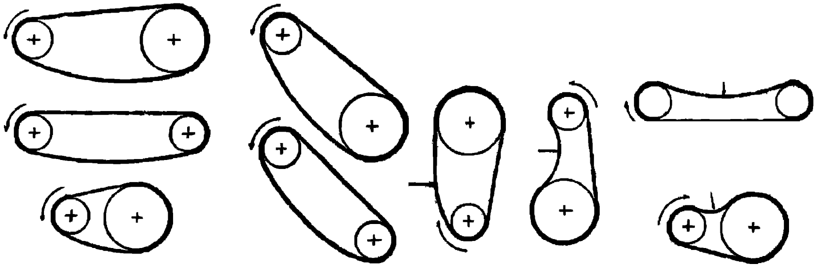


Fig. 14-34. Shaft positions for chain driving.

INSTALLATION

Alignment. Care should be exercised in the alignment of chain drives to obtain maximum service life with satisfactory performance. Proper alignment is particularly important with high-speed installations where silent and roller chains are used. The following checks should be made of the drive installation:

1. By checking the distance between the ends of the shafts on both sides of the sprockets it can be seen whether the shafts are parallel. These distances must be equal (Fig. 14-36).

2. A spirit level should be used on the sprocket shafts to see that they are level (Fig. 14-36).

3. Alignment of the wheels should be checked by the use of a straightedge along the face of the two wheels. On center-guide silent chain drives, this test should be made on center groves of sprockets (Fig. 14-36)

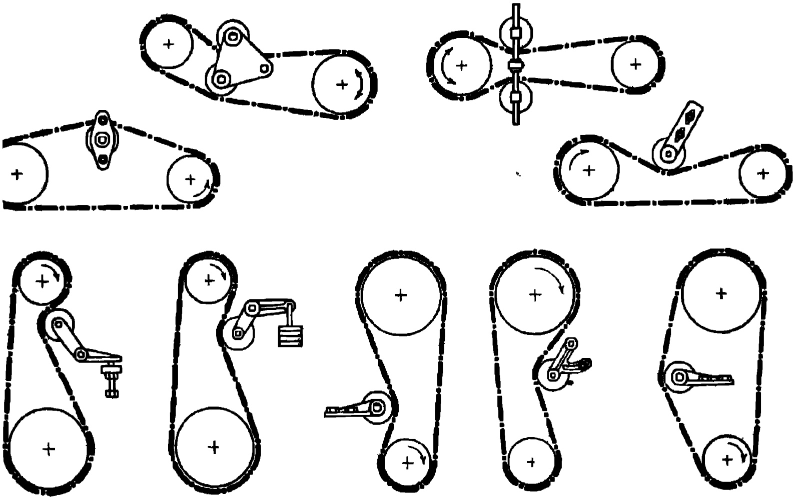


FIG. 14-35. Installation of manual and automatic idlers.

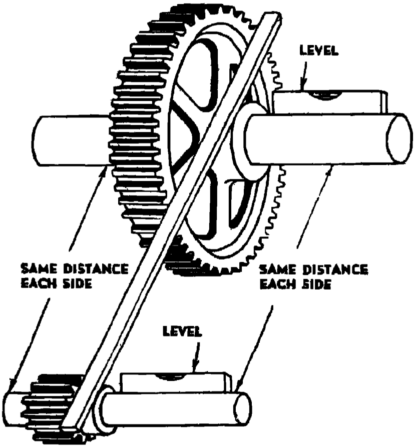


FIG. 14-36. Method for aligning sprockets.

4. Bearings should be sufficiently rigid to maintain alignment as originally mounted.

5. Sprockets should be located as near as possible to shaft bearings.

After drive is started, the alignment can be checked by inspecting the sides of the sprocket faces. This is particularly necessary if one sprocket is on the shaft of a motor that has shaft float. Wear on faces indicates wheel misalignment, which should be immediately corrected.

Coupling of Chain. Chains may be most easily coupled by lapping the ends over and engaging the teeth of the larger sprocket, which will act to hold the ends together while the coupling is made. In some cases it may be expedient to use the special coupling tools produced by the chain manufacturer. Some silent chains are designed to operate in one direction only, and care must be taken with this type of chain to see that the arrow on the side of the chain points in the direction of chain movement. Eward and closed-end pin-type chains will operate most satisfactorily if the hook or barrel end is run ahead of the bar end.

Tension. Owing to varying conditions, the amount of tension on a chain is difficult to prescribe. As a general rule, however, this tension should be less than that of a leather belt because a chain does not depend on tension for its tractive power. It is well to arrange for adjustable centers, if possible, in order to maintain a uniform amount of slack. In the case of vertical drives and in cases where there is likely to be considerable fluctuation in speed, irregularity of load, and considerable stopping and starting, chain should be run comparatively taut.

CASING

Chain-drive casings are used to exclude foreign materials which would cause excessive chain-joint wear; to provide means for adequate chain lubrication; to promote safety; and to improve the appearance and maintenance of cleanliness of the installation. Generally there are two types of casings produced: the oil-retaining casing and the oil- and dusttight casing. The first is most commonly used because of its low initial cost, but the latter will be found to be less expensive over a period of time where the chain must operate in an atmosphere laden with abrasive materials. Where absolute cleanliness is required, the latter type will prove the most satisfactory. For methods of lubrication see Figs. 14-37, 14-38, and 14-39.

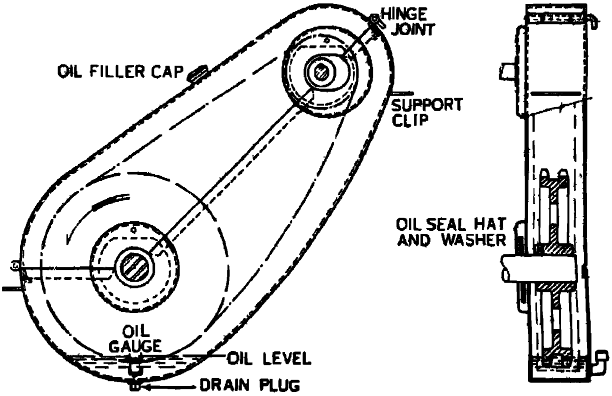


Fig. 14-37. Casing for splash lubrication.

CORROSION- AND HEAT-RESISTANT CHAINS

Corrosion Resistance. The rate of corrosion of a metal is dependent on many variables such as temperature, concentration, and degree of aeration of the corrosive agent which must be taken into consideration in the selection of the material used in the chain. Care must be taken to avoid electrolytic action, or chains will show excessive corrosion. There are two general types of chain for corrosion resistance:

1. Standard chains, which are galvanized, cadmium plated, or tinned.
2. Chains made of special materials, such as manganese, phosphor, aluminum, or silicon bronze, stainless steel (18 per cent chrome and 8 per cent nickel), or straight-chrome stainless steel (12 to 14 per cent chrome).

Chains of the first category will resist mild corrosive conditions, but the presence of more severe conditions will require that a chain of special materials be used. Manufacturers should be consulted for recommendations as to suitable material best to resist conditions to which chains will be subjected. It is highly desirable wherever possible to subject a small length of chain to the actual operating conditions before making a decision because of the fact that there are so many variations which materially affect results which on the surface seem of small importance.

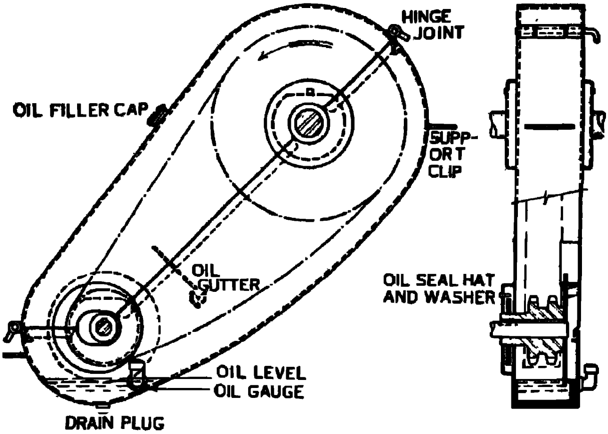


FIG. 14-38. Casing for oil-disk lubrication.

Heat Resistance. Metal loses strength rapidly with increase in temperature and is subject to oxidation and scaling. The factors that affect the life of a chain at elevated temperatures are the maximum temperature, the length of time at that temperature, and the atmospheric conditions. Standard chains will generally operate satisfactorily at temperatures up to 500°F, while some pearlitic malleable chains are

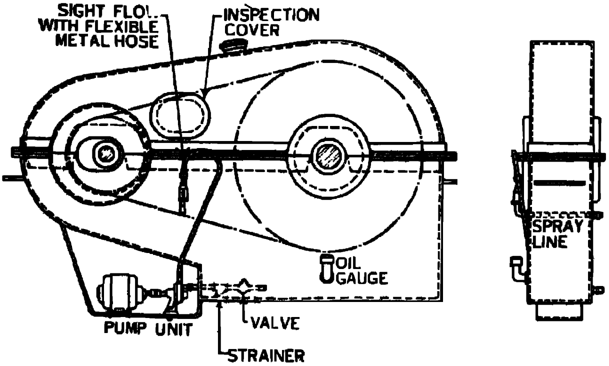


FIG. 14-39. Casing for force-feed lubrication.

operating satisfactorily at temperatures up to 1000°F. However, operating loads must be reduced with increase in temperature. Cast or unit-type construction is preferred for all high-temperature application to chains made up of a number of parts. For higher applications special alloys are available, and the chain manufacturer should be consulted. Many of these special alloy chains are operating successfully at temperatures up to 1800°F.

FLEXIBLE COUPLINGS

BY ERNEST HAGENLOCHER

Vice-President, Thomas Flexible Coupling Company

CAUSES OF SHAFT MISALIGNMENT

The most efficient means for transmitting power from a prime mover to a reduction gear or machine or from one machine to another is by flexible couplings of the proper selection and design.

Flexible couplings are designed to be used between two connected shafts to eliminate the stresses that would occur if the shafts were out of line and a solid coupling or through shaft were used. They are principally used to take care of misalignments that occur during operations. They also assist in the setting up of machines outside the erection shop since it is sometimes very difficult to get exact alignment in the field. Some types of flexible couplings also enable machines to be disconnected easily by the removal of the center ring assembly.

The misalignments that occur in operation result from

1. Settling of foundations
2. Wear in bearings
3. Temperature distortions
4. Vibrational disturbances
5. Belt tension, etc.

Machines should be aligned as accurately as possible when first set up. The full capacity of the coupling is then available to take care of subsequent misalignments.

The manufacturer's catalogue should be referred to for information as to the permissible amount of misalignment.

In parallel misalignment the two shafts are not in line but are displaced so they do not intersect (see Fig. 14-40*a*). In angular misalignment the center lines of the two shafts intersect at an angle (see Fig. 14-40*b*).

In practice there usually exists a combination of angular and parallel misalignments (see Fig. 14-40*c*). For this reason, it is a good practice to install a flexible coupling between the two connected shafts so that each machine can assume its normal position in its bearings without affecting the other machine and to eliminate undue strain on the shaft, cross pull, and end thrust on the bearings.

Consideration of what would happen in a diesel-engine and ammonia-compressor combination will indicate how serious it would be if a misalignment, either angular or parallel, were present and the two shafts were connected by a solid coupling. Since the four or more bearings cannot be held absolutely in line, reverse bending effects would occur in the shafts with each revolution. This would be disastrous both to the crankshaft of the diesel engine and to the crankshaft of the compressor.

The solid connection of a two-bearing motor and a two-bearing generator would subject the shafts to the same reversed stresses, and they would eventually break.

When a reduction gear is used, flexible couplings are required on both the high- and low-speed sides to take the strain off the pinion and its bearings as well as the low-speed gear and its bearings.

Similarly when three units are coupled together such as the motor-driven multiple-stage pump shown by Fig. 14-41, flexible couplings should be used to allow the shaft in each unit to run in its own bearings without any cross pull or end stresses from the other units.

TYPES OF FLEXIBLE COUPLINGS

There are many types of flexible couplings available, and new ones or improvements of older types are constantly being developed. However, of all the couplings on the market, there are only a few types that meet the requirements of present-day industrial operations. These are the ones that will be treated here.

Couplings can be roughly divided into two general types—single and double groups of elements. Single types are used for angular misalignment only, while double types are used for parallel and angular misalignment. Parallel misalignment is resolved into double angular by the action of the center member.

Dependability, long service, freedom from injury to connected machines, and low maintenance are the prime requirements for flexible couplings.

A flexible coupling except in a few special cases should drive like a solid coupling (especially important when the operating speed approaches critical); require a minimum of lubrication; have a minimum of backlash; provide for angular or parallel misalignment or both; have free end float under load and misalignment; have a minimum of wearing parts; and be noiseless.

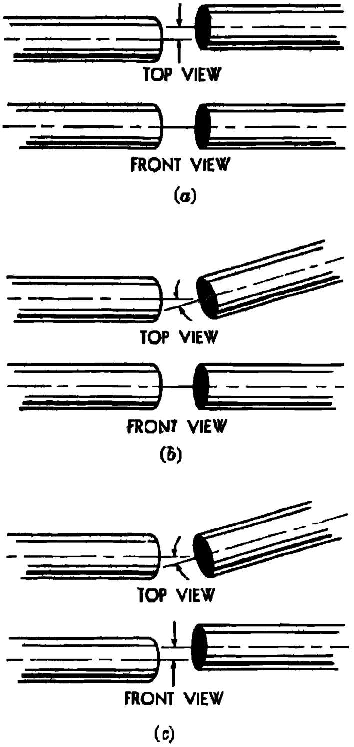


FIG. 14-40. The three ways of misalignment. (a) Parallel misalignment. (b) Angular misalignment. (c) Combination of angular with parallel misalignment in one plane.

Gear-tooth Type. Gear-tooth couplings consist primarily of two hubs with flanges, on the outer rim of which are gear teeth surrounded with an outer shell-type center member having internal teeth which mesh with the toothed flanges. The drive is from one toothed flange to the center member to the other toothed flange (see Fig. 14-42).

The method of supporting the floating center member and the shape of the gear teeth, as well as the method of retaining the lubricating oil, are the principal differences between the various makes of gear-tooth couplings.

Gear-tooth couplings are inherently of the double-engagement type and can be used for care of angular as well as parallel misalignment. A certain amount of clearance between teeth is necessary to permit the movement of the center member which acts as the flexible element in conjunction with the teeth on the flanges. The gear teeth actually act as a spline drive (see Fig. 14-43).

Grid-spring Type. The grid-spring type of coupling has a grid-formed spring which is placed in grooves which have been milled in the outer edges of two flanged hubs. The drive is through the sections of the spring grid that are carrying the load (see Fig. 14-44). Because of the constant working of the springs in the grooves, the coupling must be lubricated. This is accomplished by a bolted split cover over the flanges. This coupling is primarily a single-type coupling.

Pin and Bushing Type. Another form of coupling, the pin and bushing type, usually consists of a number of pins engaging with rubber bushings which are imbedded in holes in the opposite flange (see Fig. 14-45). This coupling is a single type. In operation this coupling depends upon the resiliency of the rubber for its flexibility which is subject to considerable stress in transmitting driving torque and accommodat-

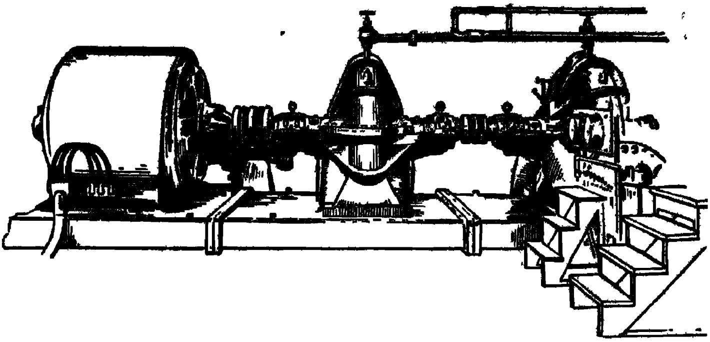


FIG 14-41. Motor-driven multiple-stage pump (coupled).

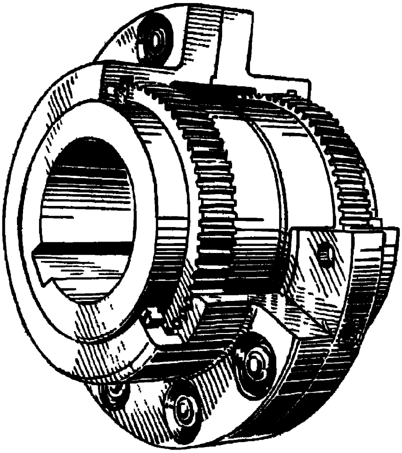


FIG. 14-42. Gear-tooth coupling.

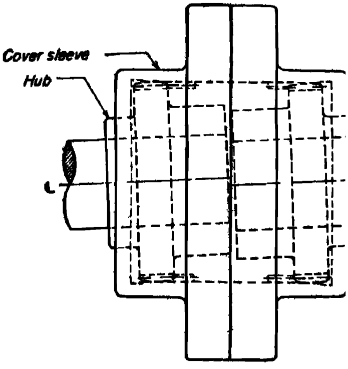


FIG. 14-43. Gear-tooth coupling in misalignment.

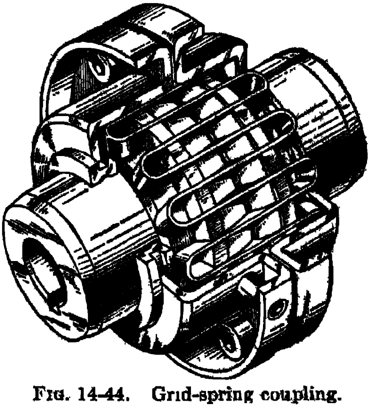


FIG. 14-44. Grid-spring coupling.

ing misalignment. In order to prevent the rubber bushings from being chafed by the action of the pins, they are usually provided with metal bushing inserts.

Chain Type. Chain-type couplings usually consist of two sprockets with hubs, one driving the other through a chain wrapped around the sprockets. The chain may be of the finished-steel roller or silent type (see Fig. 14-46).

Chain couplings are primarily of the single type and find application on smooth torque jobs. To obtain flexibility, there must be play and clearance between the chain and sprockets which result in a small amount of backlash.

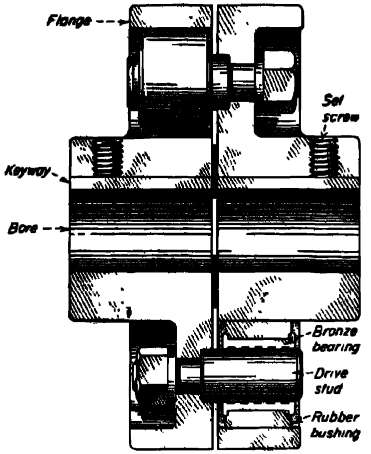


Fig. 14-45. Pin and bushing coupling.

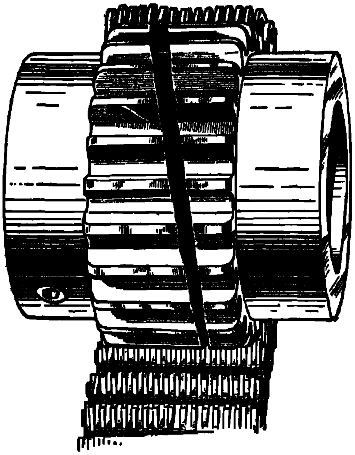


Fig. 14-46. Chain coupling.

To accommodate misalignment, the chain must be loose, which makes the chain carry the load in an offset manner. Some chain couplings have multiple rollers which reduce the friction on the teeth but place the strain on the pins.

Chain couplings require an enclosure to retain a lubricant and as a protection against foreign matter.

Block and Jaw Type. Block and jaw couplings are built on what is known as the Oldham principle, in which two hubs are provided with jaws which surround a square block, which acts as a center driving member (see Fig. 14-47).

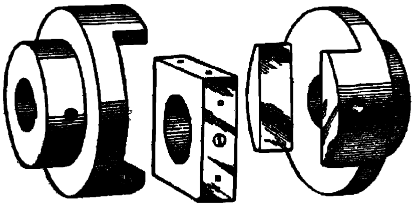


Fig. 14-47. Block and jaw coupling.

When such a coupling is operated under parallel misalignment, the block is free to slide provided there is no load on it.

If there is a load on the coupling and no misalignment, the jaw drives through a line contact on two corners of the block. As the coupling rotates under parallel misalignment, these corners rub back and forth. The greater the load, the more pressure there is on this line of contact. If operating under angular misalignment, line contact changes to point contact.

Clearance must be provided if there is to be any angular misalignment.

Metal-disk Type. The metal-disk coupling consists of two hubs with flanges to which are bolted flexible laminated-disk rings. These laminated disk rings in turn are bolted to a center member which may be in the shape of a round plate, a cylindrical spool, or a long shaft with flanges on the ends (see Figs. 14-48 and 14-49).

The principle on which the flexible metal-disk coupling operates is that any misalignment causes a slight displacement of the center member permitted by the bending of the disk rings. This is limited to 1 deg, which protects the metal fr[illegible]

The load is carried by the metal of the disks in direct tension between the bolts. There is no driving through spring action. The metal used in the disks is a specially designed alloy which has extremely high fatigue resistance.

This type of coupling drives like a solid coupling but has flexibility to take care of both kinds of misalignment either singly or in combination as well as providing free end float under load. There are no wearing parts, and no lubrication is required.

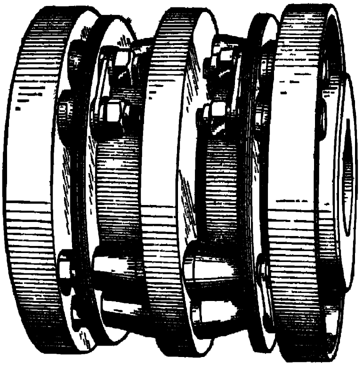


FIG. 14-48. Metal-disk coupling heavy duty, high speed.

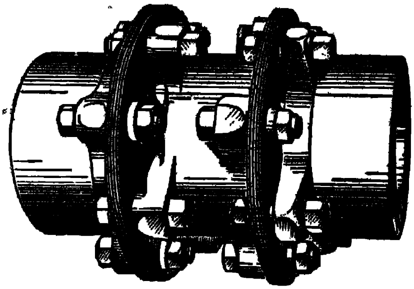


FIG. 14-49. Metal-disk coupling—heavy duty, medium speed.

All parts are in balance under any conditions of misalignment, which is especially important in high-speed couplings. The torsional characteristics remain constant regardless of length of operation since all parts are solidly bolted together.

In the double type of couplings, angular misalignment flexes the laminated disk rings with the total divided between the two rings. Parallel misalignment would also cause flexing of the disk rings, the total being divided between the two rings the same as if each disk ring were taking care of angular misalignment.

The flexible metal-disk ring coupling in the single type (see Fig. 14-50) is used for many purposes. The center ring is omitted, leaving one flexible laminated disk ring bolted to one flange and at alternate points to the other flange. In this form, since the parts are solidly bolted together, the concentricity is assured at all times, and the disk ring can be designed to carry any amount of weight. It can be used to support generator armatures where the generator has only one outboard bearing. The outboard bearing carries approximately half the weight of the armature, and the other half is carried through the coupling to the nearest bearing of the driver. Similarly, in motor-driven compressors the motor need have only one bearing—resulting in economies in material, floor space, and maintenance.

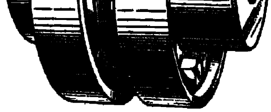


FIG. 14-50. Metal-disk single-type coupling.

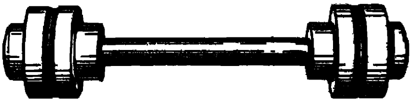


FIG. 14-51. Floating-shaft metal-disk coupling.

The Floating Shaft. The floating shaft is in reality an elongated flexible coupling and is generally used to span shaft-connected distances where the use of bearings is prohibitive. Many of the standard makes of flexible couplings can be used in conjunction with it. The metal-disk-type flexible coupling, when used in conjunction with the floating shaft, results in a very reliable and efficient drive since the farther apart the disk rings are the more misalignment can be accommodated. All parts of the metal-disk coupling type of floating shaft remain in perfect balance regardless of the period of operation since there are no wearing parts, and there is no change in the torsional characteristics. A floating-shaft metal-disk coupling is shown by Fig. 14-51. No pilot bearings are required.

SELECTION

Flexible couplings should be selected with care so that they will be large enough to accommodate the shaft sizes and suitable to withstand the stresses that occur owing to the torque load and to misalignment. The coupling should preferably have a minimum of backlash. With pulsating or uneven loads, backlash results in a hammer-like action which tends to shorten equipment life.

An important requirement in a flexible coupling is free end float *under load* so that no side thrust or axial strain will be transmitted to the connected machine. Generators and motors require a certain amount of free float in order to run in their magnetic center. Most machines are not provided with thrust bearings.

There is another important requirement when selecting couplings for machines that are placed some distance apart. Coupling hubs should be placed as closely as possible to the bearings of the connected machines, thereby shortening the overhang. The floating-shaft center member should be able to maintain its central position without the use of pilot bearings. The floating-shaft type of coupling is a very desirable method of connecting two machines since it permits greater misalignment, and this condition usually exists when machines are on separate foundations. Couplings of the flexible metal-disk type are held in exact centers because of their solidly bolted construction, and since there are no wearing parts, they always stay in balance.

Where there is a possibility of torsional vibration from medium or high speeds, serious thought should be given to flexible-coupling selection. If a coupling has wearing parts or is built upon the principle of springs or cushions, it is apparent that its torsional characteristics will vary with the degree of wear, the misalignment, and the load. Couplings of the flexible metal-disk type drive like a solid coupling, and their torsional characteristics are constant. This is of great importance since torsional-vibration stresses can be very harmful to the connected machines.

The factors that usually enter into the selection of the size of couplings are

1. The coupling must be able to transmit the torque.
2. The coupling must be able to withstand the type of load, *i.e.*, whether steady, fluctuating, or reverse.
3. The coupling selected on the basis of the above factors should have hubs that are large enough to accommodate the shafts of the units being connected.
4. The coupling should be of the type to withstand the stresses due to the speed at which it will run.
5. The misalignment to be encountered should be within the rating of the coupling.
6. Torsional stiffness.
7. The coupling must always remain in balance.

Most manufacturers supply complete information for standard applications. If the application is of a special nature, additional data should be obtained from the manufacturer before making the final selection. It is good practice to obtain the manufacturer's recommendation on applications where unusual conditions exist.

Referring to factor 1, calculate the maximum horsepower per 100 rpm on the coupling. Take the overload horsepower of the driver, divide by the rpm, and multiply by 100.

For factor 2, consult manufacturer's catalogue for type of load, such as compressor, pump, crusher, etc., and select a service factor that is dependent upon the characteristics of the driving unit.

Factor 3 is definitely a matter of dimensions.

In reference to factor 4, high-speed couplings are usually made of steel forgings machined all over and balanced. Over 1,000 rpm is usually considered high speed. Low-speed couplings are usually made of castings of steel, cast iron, or other metal.

If this misalignment (factor 5) is exceeded, some couplings will completely wear out, whereas in others only parts will wear.

Considering factor 6, if the installation is such that torsional vibration may be a factor, it is important to select a coupling possessing permanent torsional characteristics. If torsional vibration is present, it is almost impossible to absorb in the small space occupied by any make of coupling the energy necessary to change the period of vibration unless it be the floating-shaft type where the diameter of the intermediate shaft can be altered. This is most effective only if the floating shaft is at the center of the node of vibration.

Balance (factor 7) is especially important in couplings running at high speed, but it is also of great importance in slow-speed couplings where the center member is large, especially in applications such as ship drives, etc.

There are many applications where, by the use of a single-element coupling, the unit may be designed to save both space and cost. Single-bearing motors and single-bearing generators, when attached to their driven or driving units, introduce angular misalignment only. Such applications are found in diesel-driven generators on submarines, locomotives etc., and in motor-driven compressors etc. The cost of one bearing can be eliminated, and the amount of metal in the bedplate can be reduced in two ways—first, by shortening the bedplate owing to less space required; second, by lightening because less rigidity is required. Couplings for these applications must be capable of supporting the weight of the armature and still maintain exact concentricity between the shafts. Angular misalignment of 1 deg is usually permissible in such a design.

The floating-shaft type of coupling, *i.e.*, two single-element couplings connected by an intermediate tube or shaft, is being used more and more where greater misalignment is found. It also solves the problem of having the driving and driven units in different compartments or separated by fireproof walls, etc. Such cases are found in some pumping stations and aboard some ships. Also in marine work it is the ideal installation between the propeller shaft and the diesel engine. Couplings exceeding 20 ft in length have operated successfully. In such cases, however, the coupling must be capable of carrying the weight of the intermediate shaft and maintain its concentricity especially at medium and high speeds.

If thrust is present in marine applications, the application becomes special, and should be referred to the manufacturer for recommendations.

APPLICATIONS

In the ordinary installation of machinery, each unit in a coupled combination has its own bearings. This requires a "double type" of coupling to take care of the stresses that are set up if they were connected solidly to each other. This type of coupling has a center ring with connections on each side to the flanges which transforms the coupling into the equivalent of two "single types," each taking care of an angular condition resulting from the misalignment. This is easily recognized in some of the types and is illustrated here with the flexible-disk type of coupling (see Figs. 14-52, 14-53, and 14-54). If a double type of coupling is not used for this condition, the result of a single type is destructive cross pull on the bearings.

A single type of coupling is one that consists of two flanges with rubber bushings, chains, spring grid, or a single flexible metal disk ring connecting the two flanges. These are fundamentally able to take care of angular misalignment only, which is a condition which exists when the two shafts intersect at a point near the center of the coupling. This condition exists when a single outboard bearing is used to support a shaft or in a combination of two machines, one of which has only one bearing. Two single-type couplings connected with an intermediate shaft are the equivalent of a double-type coupling.

Low first cost is one reason for the popularity of some types of couplings. Manufacturers find it convenient to install these on units that are very competitive in price, and since their guarantee does not extend over a year, it is the customer's responsibility to take care of the replacements or maintenance after this period. Meanwhile, the coupling, if not of the right type, affects adversely the bearings and other parts of the unit. A little more thought given to the selection of a coupling would doubtless justify some additional expenditure initially and result in an installation that will require a minimum of maintenance or replacement.

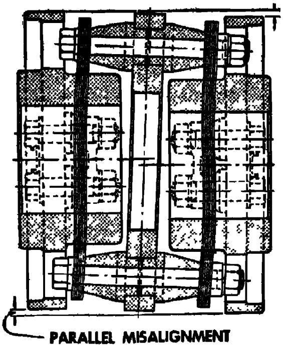


Fig. 14-52. Metal-disk coupling in parallel misalignment.

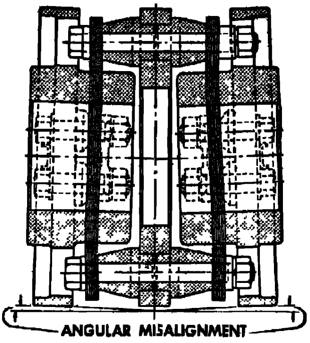


Fig. 14-53. Metal-disk coupling in angular misalignment.

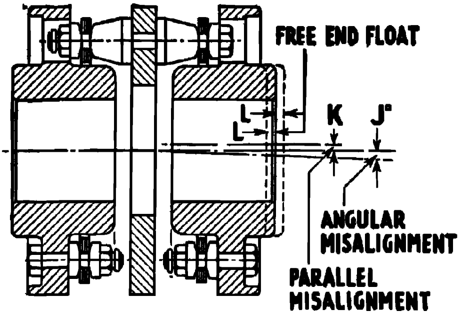


Fig. 14-54. Metal-disk coupling showing free end float.

MAINTENANCE

To avoid costly shutdowns of machinery, the couplings should be inspected on pumps, compressors, and other essential units periodically. Guard covers over couplings should be hinged to facilitate inspection.

Certain types of couplings require lubrication. If this requirement is neglected, wear will take place and will therefore do considerable damage to the coupling. Some couplings depend on lubrication to give free end float. If this is not present, side pressure is likely to be transmitted to the connected machines. This will damage the bearings or other parts. Where couplings are made with self-contained oil reservoirs, it is sometimes difficult to retain the oil at high speeds. This should be considered when making a selection.

If couplings of the single type are being used on installations that require the double type, wear and other troubles are multiplied.

For best performance couplings should always be placed close to bearings.

If trouble is experienced with a coupling drive, the initial check should be on the alignment of the shafts. This should be corrected if necessary. Then the coupling should be studied to see if it is the right type for the purpose. If not, it should be replaced with one that is more suitable and correct for the requirements.

When a coupling shows signs of strain or wear, it is economy to replace it at once before it damages the connected machinery.

VARIABLE-SPEED MECHANISMS

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INFINITE OR STEPLESS SPEED VARIATION

Infinite or stepless speed variation has extensive industrial application within two general categories:

1. Synchronizing the speeds of parts of the same machine or operations carried out in series to provide for uniform flow
2. Providing rapid, manual or automatic, flexibility in the operations of a machine for increased capacity, lower production costs, and higher precision standards

This subsection will treat the mechanical methods of stepless speed variation (see Sec. 17 for hydraulic systems and Sec. 16 for electric speed control). Mechanical systems have the advantages of being low in first cost, simple in construction and maintenance, and comparatively efficient in operation. Stepless speed variation, under load, may be obtained through the use of one or more of the following five mechanical principles:

A. Flat-belt type
B. Friction-disk or cone type
C. Ratchet or interrupted-motion type
D. V drives with adjustable conical pulley or pulleys
  1. Vari-speed-pulley type
  2. Interlocking dual pulleys, countershaft type
  3. Adjustable cone pulleys, parallel-shaft type
    a. Belt drive
    b. Chain drive
    c. Steel-ring drive
  4. Compound adjustable pulleys and interlocking dual pulleys
E. Controlled differential-gear system

In the selection of the type of speed-variation mechanism to employ, consideration should be given to some or all of the following factors:

1. Ratio of speed variation required
2. Torque or horsepower, in relation to speed requirements, throughout the range of speed variation
3. Operating conditions
    a. Source of power
    b. Uniform loading
    c. Moderate or heavy shock loading
    d. Necessity for frequent starting or reversal
    e. Braking effect required
4. Type of unit (open, semi-enclosed, enclosed) dust, fumes, weather, etc.
5. Space and mounting requirements
6. Accuracy of unit in respect to ability, under constant torque load, to duplicate ratio settings after previous adjustment
7. Per cent change of ratio accompanying change in torque

TYPES OF VARIABLE-SPEED MECHANISMS

Flat-belt Type. The flat-belt type of mechanism consists essentially of a flat belt operating with two conical (nonadjustable) pulleys with suitable belt guides and shifting means as shown by Fig. 14-55.

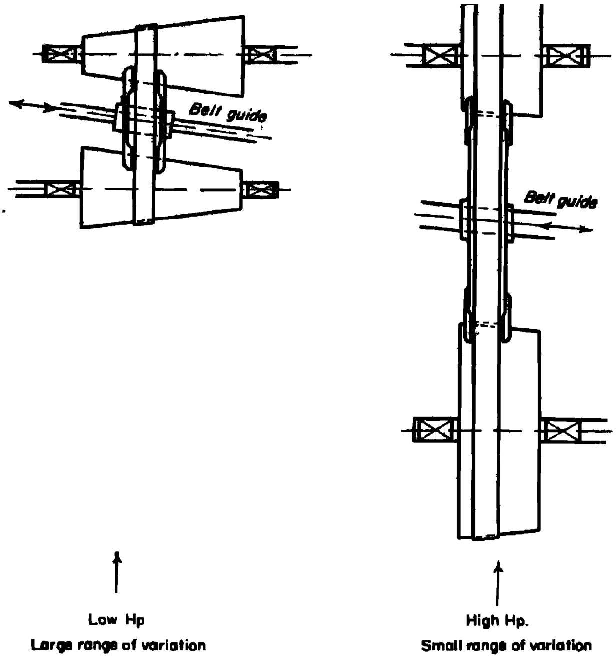

FIG. 14-55. Flat-belt type.

Two general types are in common use, although not available as standard transmission units. Flat-belt mechanisms are usually designed to suit specific applications.

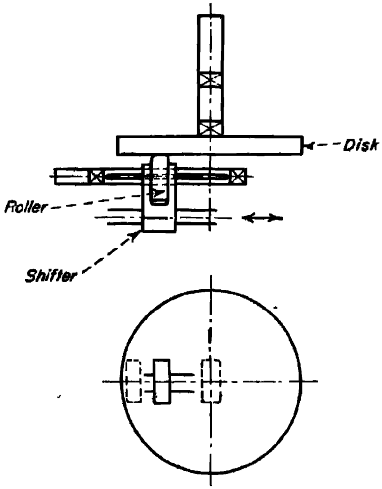

FIG. 14-56. Friction-disk type.

Friction-disk or Cone Type. One of the earliest forms of a mechanical speed mechanism consisted of a driver and a driven disk mounted at right angles to each other, and having a friction contact (see Fig. 14-56). Control of the relative speed of rotation of the two disks was maintained through varying the point of contact of the disks.

A friction-cone type of variable-speed drive is manufactured[1]* which utilizes the planetary system similar to that used in the differential drives. Three tapered rollers are used, supported in a carrier rotated by the input shaft. The rollers are free to revolve about their own centers and are mounted so that their tapered outer edge is parallel to the central axis of the machine (see Figs. 14-57 and 14-58). A steel nonrotating ring contacts the outer edge of the rollers.

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* List of manufacturers of various units, designated by superior numbers, is given at the end of this subsection.

This ring is movable longitudinally so that the point of contact on the rollers may be changed to any position between either end of the rollers, thus varying the velocity at which they revolve. Pinions on one end of the rollers engage a ring gear connected to the output shaft.

The carrier supports each roller in double bearings located between the pinions and the cone. Bearings are shim-adjusted in assembly to assure proper contact between rollers and ring. Centrifugal force acting upon the roller during rotation presses the roller firmly against the ring. The small end of the tapered roller (the end giving the slowest, or reversed, speed when contacted by the ring at that point) is at the end of the roller opposite the sliding block Thus the lever arm of the roller in relation to the lever arm of the ring is the greatest during the slow speed, or reversed, operation of the output shaft, giving the greatest traction of the roller on the ring at that time.

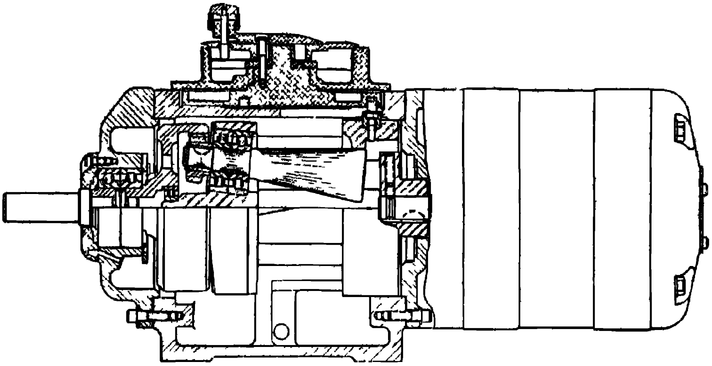


Fig. 14-57. Friction-cone type.

This unit is made in seven motor sizes from $\frac{1}{15}$ to 3 hp, and these units have speed ranges from approximately one-fifth motor speed down to zero. The same units are available for speed ranges which are equal on both sides of zero, the range being approximately one-sixth motor speed forward and reverse through zero. Motors and reduction-gear sets are available for integral mounting.

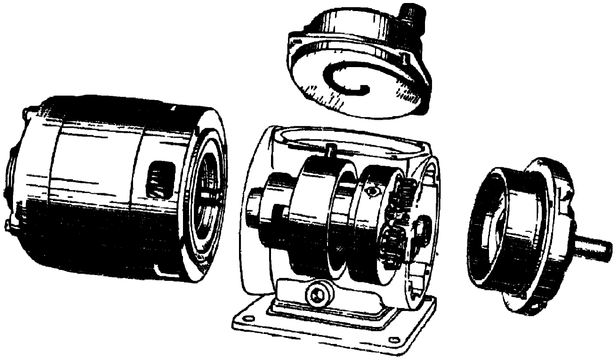


Fig. 14-58. Friction-cone type (exploded view).

Ratchet or Interrupted-motion Type.[2] The principle of the ratchet or interrupted-motion device is shown by Fig. 14-59. The amount of throw to the crank (thus the rotation of the output shaft) may be adjusted by varying the amount of eccentricity obtained on the input shaft through rotation of the double eccentrics. Another method is to provide a center pivot about which the crank arm rotates, and by changing the point of pivot of the crank arm, the amount of throw at the output end of the crank may be varied. Through the use of a ratchet or roller clutch, movement of the crank produces unidirectional rotation of the output shaft. Increasing the number of eccentrics, cranks, and clutch units will reduce the time lag between pulsations in the output shaft.

A unit of this type (shown by Fig. 14-60) is manufactured particularly for use with feeders and conveyors. This unit is rated at 3 hp, 250-rpm maximum input speed. The output-speed range is adjustable from 22 to 0.83 per cent of the input speed.

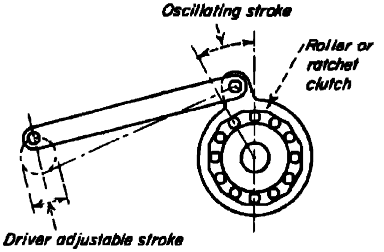


Fig. 14-59. Ratchet type (principle).

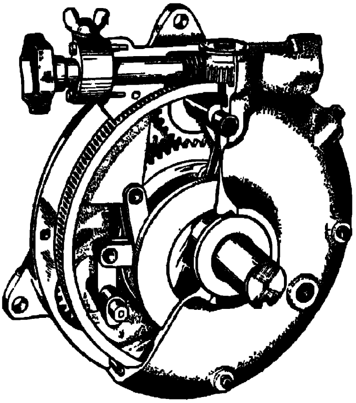


Fig. 14-60. Ratchet type.

V DRIVES WITH ADJUSTABLE CONICAL PULLEYS

The majority, in number of different types produced, of stepless speed-variation devices are modifications of V drives with adjustable conical pulleys. In general, they can be grouped under one of four subclassifications, though with some specific devices there will be an overlapping of the methods involved:

1. Vari-speed pulley
2. Interlocking dual pulleys, countershaft type
3. Adjustable cone pulleys, parallel-shaft type
4. Compound adjustable pulleys and interlocking dual pulleys

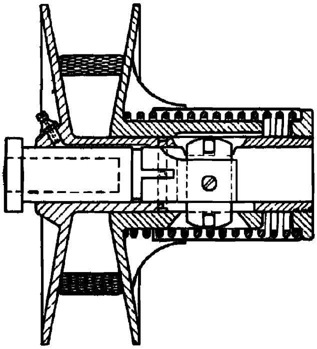
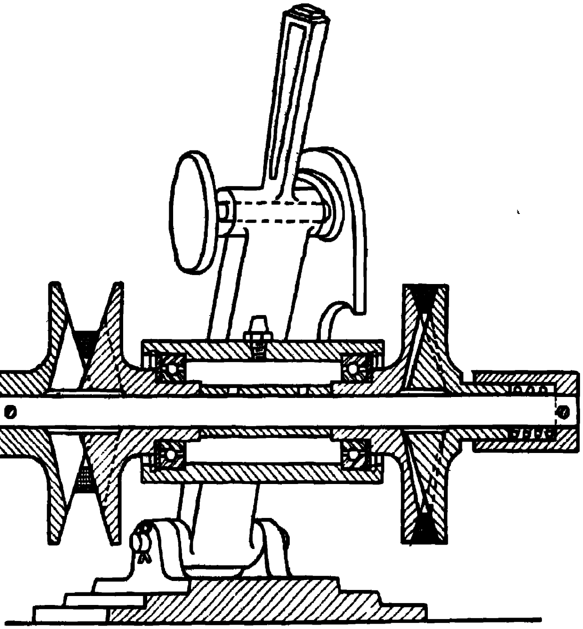


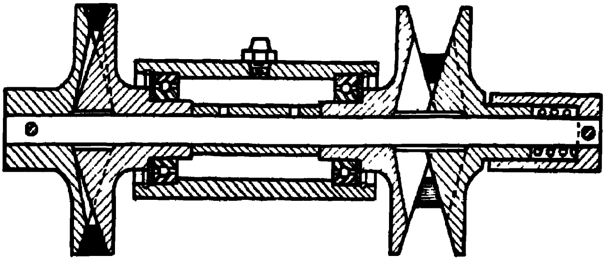
Fig. 14-61. Vari-speed pulley.

Vari-speed Pulley.[3] The simplest form of speed control, the vari-speed pulley, consists of two conical, or half pulleys, mounted on the motor shaft, one half being fixed and the other sliding laterally along the shaft. The sliding half generally has a loading spring pressing against its outer face (see Fig. 14-61). Provision is made for adjustable shaft centers, usually by means of a sliding motor base with screw-adjustable handwheel. When the motor is nearest the driven shaft, the V belt assumes the largest arc of contact on the pulley faces, with the operating effect of large pulley diameter. By turning the handwheel, the motor is moved away from the driven shaft, and the tightening of the V belt forces the pulley faces apart by compressing the spring, thus reducing the arc of contact of the pulley with the effect of smaller pulley diameter. The total variation in the shaft center distance varies from approximately 2 in. on the smaller sizes to 8 in. on the larger sizes.

Vari-speed pulleys are manufactured in a number of sizes for use with standard V belts, with capacities up to 7½ hp, 1,740 rpm, and ratios up to 4:1. They are also made in six sizes for use with multiple block-type V belts for motors up to 15 hp, 870 rpm, and ratios of 2¼:1. The maximum ratio obtainable with this type is 3:1 on some of the smaller motor sizes. A special belt may be used which combines the features of a V belt for contact between the V disks and a flat belt for contacting the driven pulley, with operating characteristics substantially the same as for standard V belts.



Slow speed position



High speed position

Fig. 14-62. Interlocking dual pulleys.

Interlocking Dual Pulleys, Countershaft Type.[4] The countershaft type of interlocking dual pulleys (Fig. 14-62) consists of two interlocking split pulleys mounted on a countershaft between the fixed driving and driven wheels. The two variable pulleys are connected on the shaft in such a manner that movement of one of the faces of one pulley, increasing the distance between the faces and thus reducing the effective pulley diameter, results in an equal amount of closure in the space between the faces of the other pulley, with increased effective pulley diameter on the second pulley. The countershaft is pivoted on its base (or in some cases on a sliding base) to allow lateral movement toward either the driver or driven shaft, which movement will result in a tightening of one belt and a slackening in the other. The effect of tightening one belt and slackening the other causes an automatic shift in the faces of the two pulleys equalizing the tension in both belts and resulting in a new ratio between their running diameters. Provision is made for locking the countershaft in the new lateral position.

Like the vari-speed-pulley type of drive, this device is economical in both first cost and maintenance but permits a greater ratio variation through the use of two adjustable wheels rather than one.

These units are made for use with standard V belts or combination belts to operate on both V and flat wheels. Sizes range up to 7½ hp at 2½:1 ratio, and ratios up to 10:1 may be obtained with some of the smaller horsepower sizes.

Adjustable Cone Pulleys, Parallel-shaft Type. The parallel-shaft type of adjustable cone pulley consists of two parallel shafts on which are mounted a pair of adjustable cone pulleys, with a belt, chain, or ring as the connecting medium. Provision is made for adjusting the spacing between the faces of the cones by a movement of one or both halves of the pulley in such a manner that, when the spacing of the pulley opening on one shaft is being increased, the spacing of the pulley opening on the opposite shaft is being decreased. Thus when the effective diameter of the pulley on the first shaft is being decreased, that of the opposing pulley is increased, and the necessary length of belt, chain, or ring to encompass the two pulleys is substantially the same.* A multiple of pulleys and belts may be mounted within a single unit.

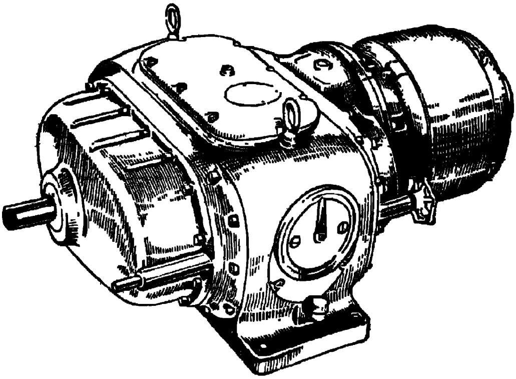


Fig. 14-63. Adjustable cone pulleys, parallel-shaft type, motorized with integral gear reduction.

Units of this general construction are invariably used where larger horsepower requirements are necessary. Most units are available with integral motors and gear reduction units (Fig. 14-63) with floor, ceiling, or wall mounting in horizontal or vertical styles.

Adjustable cone pulleys may be equipped with various accessories such as tachometers (mechanical or electrical type), electric control motors, or hydraulic control elements, with manual, semiautomatic, or full automatic control, cam or lever controls, etc. In fact, practically any known measuring or controlling instrument or element may be used to control variable-speed mechanisms provided a slight time lag can be allowed in order to allow the control to function. Beyond this much depends upon the ability to coordinate the signal of the control instrument or element with the variable-speed-control actuating means with sufficient accuracy to provide desired results. Generally speaking, most automatic controls are of the follower type; *i.e.*, a temperature, tension, speed, etc., is the signal element which sets in motion the variable-speed-control actuating means. In such a case, the variable speed control is attempting to respond to, and correct, an existent condition which is approaching the limit of temperature, tension, speed, etc.

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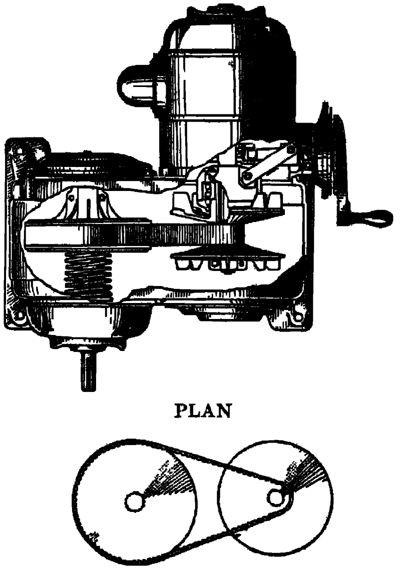
* Actually there is a difference. The correct length of belt to operate over 1:1 ratio pulleys with fixed shaft centers will not be long enough to operate on other than 1:1 ratio pulleys on the same shafts even though the sum of the pulley diameters in the second case be equal to the sum of the pulley diameters at the 1:1 position. On some drives adjustment is made by mechanical means, such as equalizing linkage or spring loading of one half of a pulley, while on others a convex curving of the V pulley faces tightens the belt as it approaches the 1:1 position.

Manufacturers, when specifying the maximum allowable ratio of speed variation, give the ratio from the highest to lowest output speed, not the amount of variation, higher or lower, that may be made of the output speed in relation to the input speed. Most units of this type are furnished with pulleys of equal size on both shafts, which allows the speed of the output shaft to be increased or decreased an equal amount from the speed of the input shaft. In such a case, if the speed of the input shaft is known, the maximum high or low speed of the output shaft may be calculated by multiplying and dividing, respectively, the speed of the input shaft by the square root of the specified speed variation of the unit. With a unit having a specified maximum speed variation of 9:1 and an input speed of 1,150 rpm, the maximum output speed would be the square root of 9 (or 3) times 1,150 for a total of 3,450 rpm. The minimum output speed would be 1,150 divided by 3, or 383 rpm (3,450 to 383 is a 9:1 ratio). Thus in this unit, the ratio of input speed to output speeds is 3:1 and 1:3.

The following discussion of specific applications of this device is divided on the basis of the type of connecting medium (belt, chain, steel ring).

Belt.[5] Devices are available for use with standard V belts or with special sizes of fabric and steel in rubber belts. These are manufactured in numerous sizes up to 75 hp at 3 :1 ratio, while some smaller units will provide ratios up to 6 :1 (see Fig. 14-64).


PLAN

ELEV.

FIG. 14-64. Adjustable cone pulleys, parallel-shaft type, motorized.

Several manufacturers produce devices for use with a special belt consisting of an endless-cord belt clamped between wood blocks whose ends, tipped with leather pads, are tapered at the same angle as the cone faces (see Figs. 14-65 and 14-66). Various sizes are produced up to 87-hp output at a 2:1 ratio. In the smaller sizes will be found units providing ratios up to 16:1.

Chains.[6] The PIV (positively, infinitely variable) type of transmission differs from the other types of V drives in that it does not depend upon friction for its tractive effort. The chain consists of bushed pin-joined cages, each holding a pack of flat hardened-steel laminations tapered at the ends to the same angle as the wheel faces. These laminations are free to slide from side to side, individually, with relation to each other (Fig. 14-67). This type of transmission utilizes an arrangement of conical-faced wheels having radial teeth cut into their faces (see Fig. 14-68). The wheels are so mounted on the shaft that immediately opposite the projecting tooth on one face of a wheel is a groove on the face of the opposing wheel. As the chain enters the V formed by the two wheel faces, the staggered teeth of the wheels push the requisite number of steel laminations transversely (at right angles to the chain travel) into position in the opposing tooth spaces to form the positive engagement which transmits the load. This engagement is complete before the load or pull on the slats is applied, and there is no further movement while they are under load.

The wheel faces are not straight cones but have a slight convex curve to maintain proper chain tension at all positions of speed variation. A hardened-steel shoe rides lightly on both the upper and lower strands of the chain to assure correct entry and engagement of the chain slats into the sprockets. The use of one on both the top and bottom strands allows the chain to be run in either direction. Splash lubrication is provided in the sealed case.

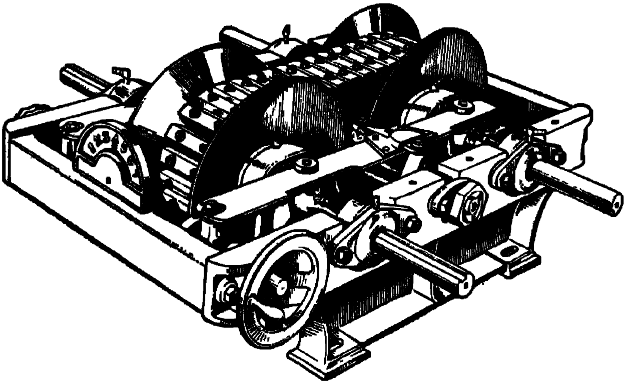


Fig. 14-65. Adjustable cone pulleys, parallel-shaft type, open.

Eight sizes are produced, in capacities from ½ to 25 hp, with maximum ratios of up to 6:1.

The VRD (variable roller drive) is of the parallel-shaft type with a pair of adjustable conical wheels and employs a roller chain of special design for the transmission of power between the wheels on the two shafts (see Fig. 14-69). The endless side-contact chain (Fig. 14-70) has openings in each pitch, or section of links, which provide a longitudinal (parallel to direction of chain travel) pocket for each of two hard-steel rollers which protrude sufficiently to contact the smooth-faced conically opposed disks. At each engagement of chain and wheels, with the opposed disks approaching at an acute angle, each pair of chain rollers is free to roll into driving contact with the wheels. The disengagement of the chain is likewise a rolling action. This device operates in a totally enclosed, oiltight housing, providing splash lubrication.

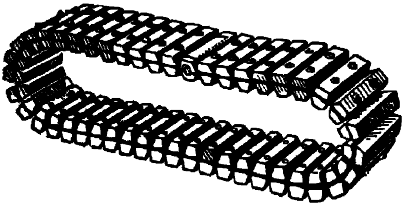


Fig. 14-66. Endless belt with leather-tipped blocks.

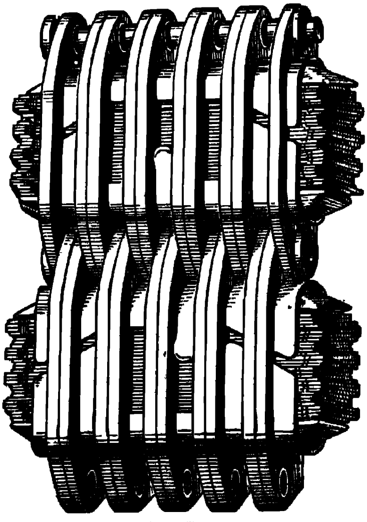


Fig. 14-67. PIV chain.

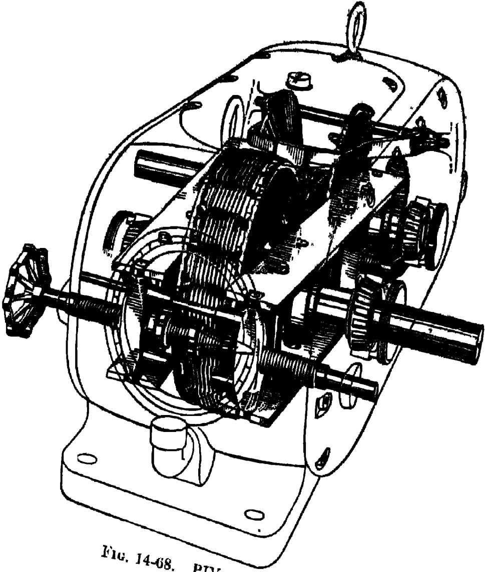


Fig. 14-68. PIV transmission.

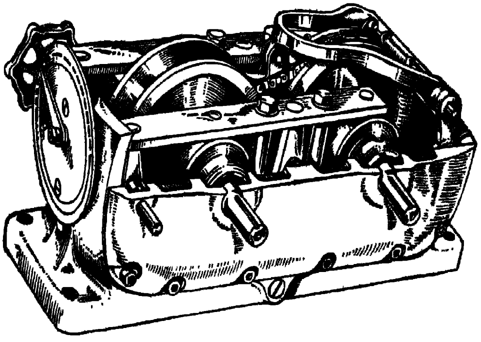


Fig. 14-69. VRD transmission.

The VRD unit is built in one size having a maximum rating of ½ hp and a maximum ratio of speed variation of 10:1.

Steel-ring Drive.[7] Steel-ring drive employs a circular steel ring running on variable conical pulleys to transmit the power between the parallel shafts (see Fig. 14-71). When a load is applied to the output shaft, the rotating ring shifts laterally in a direction at right angles to a line connecting the two shaft centers. This movement of the ring wedges it into the angle of the cones and increases the contact pressure between the cones and the ring in proportion to the load applied. Lubrication is provided by an oil bath in the sealed case.

Various sizes of units are manufactured up to 3 hp at 6:1 ratio, with a maximum ratio of 15:1 available on some of the smaller sizes.

Compound Adjustable Pulleys and Interlocking Dual Pulleys.[8] Compound adjustable pulleys and interlocking dual pulleys are enclosed units. They contain an input and an output shaft mounted in line, each with an adjustable V pulley. A speed-adjustment lever controls the opening in each of these pulleys in such a manner that movement of the lever increasing the opening of one pulley decreases the opening of the second. A countershaft of the interlocking dual-pulley type is mounted parallel to the first two shafts, one of the dual pulleys being belted to the input-shaft pulley and the second belted to the output-shaft pulley (see Fig. 14-72). Thus movement of the speed-variation adjustment lever, causing a tightening of one belt and a loosening of the other, will result in an automatic shift in the openings of the countershaft pulleys to equalize the tension in both belts at the new ratio.

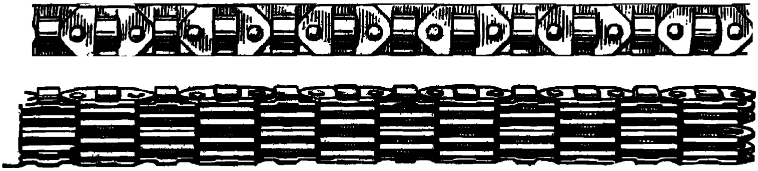


FIG. 14-70. VRD chain.

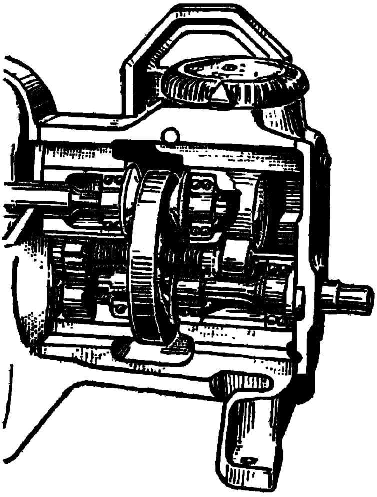


FIG. 14-71. Steel-ring drive.

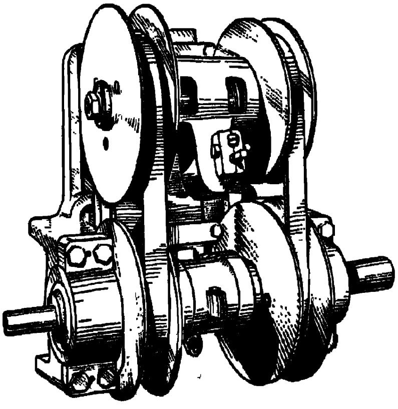


FIG. 14-72. Compound and interlocking dual pulleys, open type shown. Also available with case and cover.

Various sizes of this device are produced, ranging from ¼ hp with 16 :1 ratio, up to 10 hp with 6 :1 ratio.

CONTROLLED DIFFERENTIAL-GEAR SYSTEMS

A differential-gear system alone will not produce a variable-speed drive but must be supplemented by some other variable-speed mechanism to control the relative rotation of the various gears. This is commonly done by adding to the differential gearing a control mechanism based upon the principle of the V belt or chain drive with adjustable cone pulleys. The relative capacity of the combined V drive and differential-gear systems depends upon the ratio range of the V drive and the required ratio range of the differential system.

The advantage of this combination over the use of the V drive alone lies in the fact that only a portion of the power transmitted by the combined system passes through the V-governing drive. Thus the size and weight may be reduced as compared to V drives of equal rating, and units may be constructed to handle larger horsepower requirements. Units may be constructed to operate as low as zero speed or through reverse. In some cases the differential gears are employed to supplement the accuracy of the V drive alone; in others to reduce the effect of ratio change vs. transmitted torque.

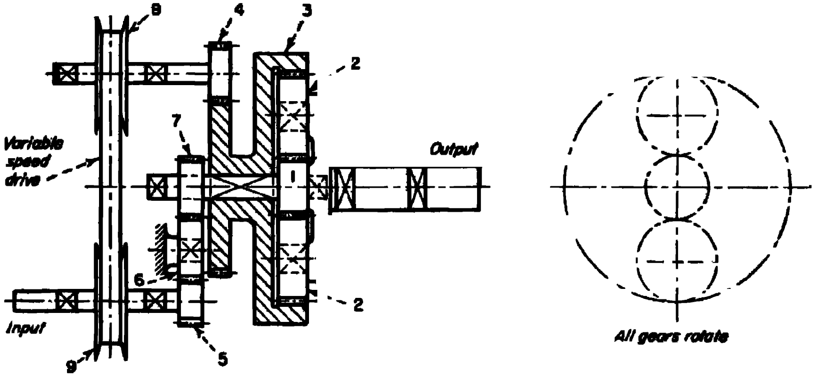


Fig. 14-73. Controlled single differential.

The type of differential-gearing system will vary from one manufacturer to another, some being of the bevel-gear type and others of the planetary-gear type. Some devices employ a double differential-gear system. In the case of the single differential, the torque transmitted through the variable element deviates considerably, depending upon the relative ratio range. With the double-differential type, the torques transmitted through the variable elements do not vary so widely as with the single-differential type.

The gear system shown by Fig. 14-73 is typical of a single-differential device,[9] although numerous other combinations are possible. This uses planet gears (2) located symmetrically about, and meshing with, a pinion (1) which is driven by the input shaft through gears (5, 6, and 7). The inner end of the output shaft has projecting fingers which form the shafts on which the planet gears (2) revolve. The speed of rotation of the output shaft is governed by controlling the relative speed of rotation of a ring gear (3) encompassing the planet gears (2) with respect to the rotation of the input shaft. This control is maintained by a V belt or chain mounted on two adjustable cone pulleys (8 and 9), one of which (9) is on the input shaft, and the other (8) is on a shaft geared through gear (4) to the ring gear.

The single-differential type, in sizes up to 100 hp, may be designed to allow continuous variation from maximum to zero speed or through zero to reverse.

A double-differential-gear system with V belt or chain control mechanism is illustrated in Fig. 14-74. The pinion (15) on the input shaft drives two spider arm gears (1 and 2), which respectively carry the free-turning bevel gears (3 and 4), with them. The relative rotation of the two bevel gears (5 and 6), is controlled by the belt or chain on the two variable V wheels (7 and 8). With pulleys (7 and 8), at the 1:1 position, the bevel gears (5 and 6), rotate at equal speeds, although the other gears (3, 4, 9, and 10), may rotate at sundry speeds dependent upon the ratio of the gears (1 and 2), to pinion (15) and also gears (11 and 12), to gear (14).

With equal ratios in gear sets (1 and 2), to (15), (11 and 12), to (14), and in the two differential-gear sets, equal forward to reverse speed is obtainable by changing the variable-speed V transmission an equal amount or range either side of the mean or 1:1 position, *e.g.*, +400 through 0 to −400 rpm.

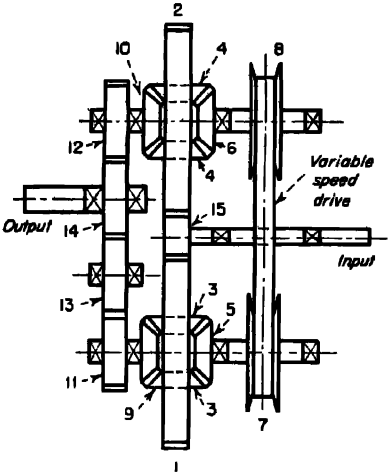


Fig. 14-74. Controlled double differential.

With unequal ratios in gear sets (1 and 2) to (15) and/or (11 and 12) to (14), substantially any desired variable-speed range may be obtained, as follows:

Zero to maximum, *e.g.*, 0 to +800 rpm

Forward through zero to reverse, (unequal values), *e.g.*, +250 through 0 to −75 rpm

Minimum to maximum, *e.g.*, +10 to +600 rpm

Manufacturers of Various Types of Variable-speed Units

Number indicates reference in text to type of unit.

1. Graham Transmissions Inc.
2. Morse Chain Co.
3. Equipment Engineering Co.
   Lewellen Manufacturing Co.
   Mechanical Handling Systems, Inc.
   Reeves Pulley Co.
4. The American Pulley Co.
   The DoAll Co.
   Ideal Commutator Dresser Co.
   Lovejoy Flexibility Coupling Co.
   Stephens-Adamson Mfg. Co.
   Toledo Timer Co.
5. Allis-Chalmers Mfg. Co.
   Lewellen Mfg. Co.
   Reeves Pulley Co.
   Sterling Electric Motors, Inc.
   U.S. Electrical Motors, Inc.
6. Link-Belt Company
7. Master Electric Company
8. Worthington Pump and Machinery Corp.
9. Lombard Governor Co.
   Mechanical Handling Systems, Inc.
   Speed Control Corp.

GEAR REDUCTION UNITS AND PLANETORQUE REDUCERS

BY J. N. MORRELL

President, E. I. M. Company (Formerly Chief Engineer, Philadelphia Gear Works)

MAJOR DESIGN FEATURES

Many factors require consideration in the design of gear reduction units. It is not the object of this subsection to treat on detail design. However, the engineer should be familiar with the major design features. These are as follows:

Horsepower rating for strength of gear teeth
Horsepower rating for surface durability of gear teeth
Suitability of material for the gears
Practical combination of tooth height, pressure angle, and/or helix angle
Tooth runout, spacing, lead
Relation of center distance and face width to avoid undue concentration of pressure arising from deflection underload
Shaft stresses
Bearing loads and speeds
Foundation and bearing-bolt stresses
Rigidity of housing substantially to maintain shaft position under maximum loads
Thermal horsepower capacity of the unit

The American Gear Manufacturers Association was established during 1917. Within the Association is an engineering body which has formulated standards for the manufacture of gearing and gear reduction units. These standards fully consider the major design features outlined above and are looked upon as the basis for the design, manufacture, and application of the modern gear reduction unit.

Gear reduction units and planetorque reducers are built in many forms, using a wide variety of gears to meet practically every kind of power-transmission and speed-reduction application. Each of the more commonly used speed-reduction units will be treated separately.

PROPER SELECTION

The general advantages of gear reduction units are manifold and must be considered properly to select the correct type for the specific application.

General Advantages. 1. *Power Conservation.* Because of accurate gear design, high-grade workmanship, proper bearings, and adequate lubrication, there is an assurance of minimum loss between applied and delivered power.

2. *Low Maintenance.* If the correct design and power capacity for the requirements are selected and occasional attention is given to the lubrication system, low maintenance costs will result.

3. *Operating Safety.* All gears, bearings, and shafts are enclosed in oiltight, strongly built, cast-iron or steel housings.

4. *Silent Operation.* The entire mechanism is tightly sealed and operates in a bath of oil; therefore noise and vibration are reduced to a minimum.

5. *Space Conservation.* Units are entirely self-contained and extremely compact; therefore they require small space. This also enables them to be installed in out-of-the-way locations.

6. *Adverse Operating Conditions.* Efficient enclosure protects the mechanism from dirt, dust, soot, abrasive substances, moisture, or acid fumes.

7. *Economy.* Units permit the use of high-speed motors direct-connected to low applied speeds. High-speed motors cost less initially.

8. *Life Expectancy.* The life of a unit is practically unlimited if it is properly lubricated and correctly aligned.

9. *Horsepower and Ratios.* Units are available in almost all desired ratios and for all practical power requirements.

10. *Adaptability.* Regardless of the direction between applied and delivered power, a reduction unit can be obtained to fill the requirements.

Efficiency. Efficiency means power conservation. While several types of gear reduction units of identical ratio, identical input speed, and identical horsepower could be selected for a given application, the efficiency of the drive may vary from 50 to 96 per cent. Table 14-94 shows anticipated efficiencies of the various types of speed-reduction units.

Table 14-94

Type of reduction unit	Ratio	Stage of reduction	Efficiency, per cent
Herringbone and helical	Up to 10:1	Single	98
	10:1–60:1	Double	97
	60:1–360:1	Triple	96
Spiral bevel and hypoid	Up to 6:1	Single	98
Spiral-bevel herringbone and spiral-bevel helical	6:1–40:1	Double	96
	20:1–240:1	Triple	95
Gearmotors	4:1–10:1	Single	98
	10:1–20:1	Double	97
	20:1–100:1	Triple	96
Worm gear	3½:1–100:1	Single	*
	100:1–10000:1	Double	†

* Efficiency of single-reduction units is 100 − (ratio/2).
† Efficiency of double-reduction units is 100 − (ratio of first reduction/2) × 100 − (ratio of second reduction/2).

Space conservation also means efficiency. Gear reduction units, being extremely compact, comprising types built with various relations of input shaft to output shaft, permit remarkable adaptability. The unit that satisfies the requirements of relation between applied and delivered power, both as to location and rotation of input and output shafts, should be selected. Then the most efficient of the several types of drives available should be chosen.

All other features listed under general advantages should automatically be realized.

Before a gear reduction unit can be selected for any given application, the equivalent horsepower must first be computed by multiplying the specified or actual horsepower by a "service factor" for the particular class of service for which the gear reducer is to be used. It is necessary for low maintenance, operating safety, and reasonable life expectancy that the gear reduction unit selected have a capacity equal

Table 14-95. Service Factors

Character of load on driven machine	Source of power								
	Electric motor drive			Multicylinder gas or steam engine or turbine			Single-cylinder gas engine		
	Intermittent, 3 hr per day	8–10 hr per day	24 hr per day	Intermittent, 8 hr per day	8–10 hr per day	24 hr per day	Intermittent, 3 hr per day	8–10 hr per day	24 hr per day
Uniform	0.8	1.0	1.25	1.0	1.25	1.5	1.25	1.5	1.75
Moderate shock	1.0	1.25	1.5	1.25	1.5	1.75	1.5	1.75	2.0
Heavy shock	1.5	1.75	2.0	1.75	2.0	2.25	2.0	2.25	2.5

to or in excess of the equivalent horsepower. The recommended service factors for various load characteristics and duration of service are given in Table 14-95.

Certain typical and more common speed-reduction-unit applications are classified in Table 14-96 into the three commonly recognized load classifications according to the nature of the load and the usual duty cycle. No attempt has been made to include all possible applications of gear reduction units, but a sufficient number and variety have been covered to serve as a guide for proper application classification.

Table 14-96. Load Classification

Uniform load		Moderate-shock load		Heavy-shock load
Agitators, liquid and semiliquid	Fans, centrifugal and propeller	Beaters, pulp	Feeders, apron, belt, screw,	Compressors, single- or two-cylinder
		Blowers, lobe		
Blowers, centrifugal or vane	Feeders, disk type	Compressors, centrifugal or rotary reciprocating (three or more cylinders)		Conveyors, reciprocating
	Generators		Hoists, reversing Skip	
Conveyors (uniformly loaded or fed) Apron Assembly Belt Oven Screw	Line shafts, light duty		Kilns, cement Lime	Crushers, ore Stone
	Mixers, liquid Semiliquid	Conveyors, heavy duty Apron Belt Live roll Screw	Mills, ball Pebble	Dredge cutterhead
	Pumps centrifugal Geared Rotary		Mixers, dough	Feeders, reciprocating
Can-filling machines				
		Drawbench	Pumps, reciprocating (three or more cylinders)	Jordans
Elevators, bucket (uniformly loaded or fed) Centrifugal discharge Gravity discharge Continuous bucket	Screens, traveling, water, intake	Dryers, rotary		Mills, hammer Pug Rod Rubber
	Stokers	Elevators, heavy-duty bucket Freight	Screens, rotary, gravel or stone	Pumps, reciprocating (single or two cylinders)
	Textile machinery			
		Fans, mine Induced draft	Stock chests	
			Washers, reversing	
			Wire-drawing machines	

While Table 14-96 is a suitable application-classification guide, consideration must be given applications such as the following:

High Energy Loads. Extreme repetitive shock and those applications where exceedingly high energy loads must be absorbed, as when stalling, require special consideration and are, therefore, not covered by service factors given in Table 14-95.

High Momentary Loads. Where the actual maximum momentary or starting load exceeds 200 per cent of normal, the equivalent horsepower is obtained by dividing the maximum momentary load by 2. If this value exceeds the value for equivalent horsepower obtained from a consideration of the service and duty, it must be used in selecting the size of reducer.

Brakes. Where motors are equipped with brakes and the torque rating of the brake exceeds the rating of the motor, the rating of the brake dictates the selection of the gear reduction unit.

High Input Speed. Gear reduction units of the types listed in Table 14-94 are suitable for application where the input speed does not exceed 3,600 rpm and where the pitch-line velocity of the gearing or the rubbing velocity of the worm in worm gearing is not in excess of 4,000 fpm. Special consideration must be given drives where these values are exceeded.

Speed-increasing Units. The single-stage reduction unit of the several types within Table 14-94 may be used for speed-increasing purposes as well as speed-reducing units. However, the output speed in such instances must be limited to 3,600 rpm, and the pitch-line velocity of the gearing or the worm rubbing velocity may be limited to 4,000 fpm. When higher output speeds and higher pitch-line velocity are required, these need special consideration.

Overhung Loads. Where the gear reduction unit is not directly coupled to the motor or other high-speed prime mover and also where the gear reduction unit is not coupled directly to the load, special consideration must be given the resulting overhung loads. A gear, sprocket, sheave, or pulley may be mounted on either or both input and output shafts. It is advisable with such applications to provide an outboard bearing. However, if structural restrictions interfere, the manufacturer should be advised of the pitch radius of the gear, sprocket, sheave, or pulley and the load to be transmitted by the gear, sprocket, sheave, or pulley so that full consideration may be given the bearings of the unit.

WHAT CARE TO GIVE THE GEAR REDUCTION UNIT

Proper selection of the gear reduction unit ensures low maintenance. But, this does not mean "no maintenance."

Prime consideration must be given alignment of the equipment upon installation. Gear reduction units are frequently supplied coupled with the driving motor or other prime mover and mounted on a common bedplate. Such assembly ensures proper alignment and checking of alignment in the machine shop before delivery to the field.

Upon field installation the bedplate assembly must be properly aligned and coupled to the driven machine. Much damage not immediately noticeable has occurred when misalignment has been initially present.

Observation of coupling action when first operating a drive provides a simple reasonable check of alignment. Misalignment has been the cause of coupling failure, shaft breakage, and bearing replacements, all of which can be avoided by good alignment initially.

After initial installation and alignment, the gear reduction unit should be filled with lubricant specified by the manufacturer. All gear reduction units built in accordance with the American Gear Manufacturers Association standards carry a name plate bearing full instructions for the selection of the correct lubricant.

After a gear unit is first installed and filled with oil, the original oil should be changed or filtered at the end of 2 weeks. The casing should be thoroughly flushed out with a light flushing oil. After this, change of oil every 6 months should be sufficient unless there are unusual temperature conditions combined with intermittent high loads, where the temperature of the case rises rapidly and then cools off quickly. Sweating of the casing walls occurs under these conditions, thus contaminating the oil and forming sludge. Should the unit be subject to such conditions or to wet or exceptionally moist atmosphere, it is advisable to change oil at 90-day intervals or more frequently.

Greased Bearings. Most bearings are lubricated by splash through the medium of the revolving gears. Some bearings, however, are grease-lubricated. These bearings are normally provided with grease fittings and should be checked at regular intervals of 90 days.

Venting of Units. Gear reduction units are usually provided with "breathers." The breathers should be examined not less than every 90 days to ensure efficient operation of the gear reduction unit. Where equipment is subject to excessive dirt, dust, soot, abrasive substances, moisture, or acid fumes, more frequent inspection of the breathers is warranted.

Forced-feed Lubrication. Some gear reduction units are equipped with a force-feed system of lubrication. Driver of the pump may be the gear reduction unit or a separate motor. In either event, daily checking of the pump drive and the filter which is normally a part of the system is necessary. Upon starting up after idle periods, particular attention should be given the pump.

Lubrication of Gear Reduction Units. The American Gear Manufacturers Association standard specification for lubrication of enclosed gear drives is applicable to the types listed in Table 14-94.

The specifications apply only when the gears and enclosures are designed and rated in accordance with American Gear Manufacturers Association standards. The ambient-temperature range for which they apply is from 0 to 150°F. Gear drives operated outside this range or those operating in extremely humid or chemical-laden atmosphere should be referred to the manufacturer or lubricant supplier for his recommendation.

The areas of contact on gear teeth are relatively small, and the unit pressures produced in transmitting the load are relatively high. It is, therefore, essential to provide a film of lubricant of sufficient strength to withstand the localized pressure during the period of contact. The peripheral speed of the gears governs the period of tooth contact and determines the length of time the film must withstand the pressure. When speeds are high, the time is very short, and the loads are usually light. As a result, an oil of comparatively low viscosity should be used. When speeds are low and the loads heavy, the contact time is considerably longer, and an oil of higher viscosity should be used.

Table 14-97

AGMA lubricant No.	Viscosity range SUV sec	
	At 100°F	At 210°F
1	180–240	
2	280–360	
3	490–700	
4	700–1,000	
5		80–105
6		105–125
7		125–150
7 comp.*		125–150
8		150–190
8 comp.*		150–190

* The oils marked "comp." are those compounded with 3 to 10 per cent acidless tallow or other suitable animal fat.

Temperature. The ambient temperature has considerable bearing on the viscosity of oil to be used. The viscosity of mineral oil varies inversely with the temperature, which makes it desirable to use an oil of heavier grade at high ambient temperature. Also, for low temperatures an oil must be selected that has a pour point lower than the minimum temperature expected for the application.

Type of Oil. Lubricating oils should be high-grade, high-quality, well-refined petroleum oil. They must not be corrosive to the gear, or to ball, roller, or sleeve bearings. They must be neutral in reaction, free from grit or abrasive, and have good defoaming properties. For high operating temperatures they must have good resistance to oxidation.

Viscosity. The viscosity range for the various American Gear Manufacturers Associations lubricants is shown in Table 14-97.

Table 14-98 shows the grade of oil to use for drives of all types except those employing worm gears. The latter group is covered by Table 14-99.

Table 14-98. Recommended Lubricants for Enclosed Units of All Types Except Worm Gears

Type of unit	Size of units	Ambient temperature, deg F		
		0–40, use AGMA No.	41–100, use AGMA No.	101–150, use AGMA No.
Parallel-shaft units	Low-speed centers up to 20 in.	2	4	5
	Low-speed centers over 20 in.	3	5	6
Planetary-gear units	OD of housing up to 16 in.	2	3	4
	OD of housing over 16 in.	3	4	5
Gearmotors	All sizes	2	4	5
Spiral or straight bevel-gear units	Cone distance up to 12 in.	2	4	5
	Cone distance over 12 in.	3	5	6
High-speed units	All sizes	1	2	3

Table 14-99. Recommended Lubricants for Enclosed Worm-gear Units Only

Service and worm speed	Ambient temperature, deg F		
	0–40, use AGMA No.	41–90, use AGMA No.	91–120, use AGMA No.
Intermittent operation, all worm speeds*	5	5	7 comp.
Continuous operation, worm speeds below 600 rpm	7 comp. diluted†	8 comp.	8 comp.
Continuous operation, worm speeds 600 rpm and over	7 comp. diluted†	7 comp.	8 comp.

* Where period of operation is insufficient to produce any appreciable rise in oil-bath temperature.

† Diluted AGMA No. 7 comp. oil should be diluted with a lighter oil, preferably not exceeding 500 sec viscosity at 100°F, until the desired fluidity is obtained. The lubricant used for dilution should be of the same basic crude as that of the recommended oil. The lubricant supplier should be consulted if there is any doubt.

Antifriction Bearings. A good mineral grease should be used for antifriction bearings. Choice of lime-soap grease or soda-soap grease is recommended dependent upon bearing temperatures.

Lime-soap greases are suitable for bearing temperatures of −20 to +45°C. Soda-soap greases have a higher drop point and, therefore, can be used between −20 and +70°C.

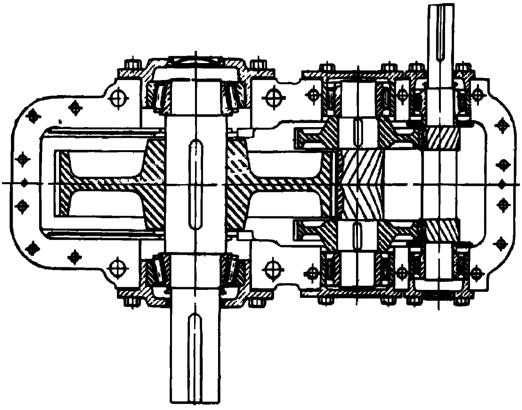


FIG. 14-75. Sectional view of double-reduction herringbone-gear unit.

Ball and roller bearings require only a small amount of lubricant, and lubrication intervals are generally long. However, lubricants must always be present in the bearings; otherwise their life will be affected.

Thermal Capacity. Since most oils lose their lubricating quality at temperatures of approximately 200°F, it is standard practice to limit the temperature of the oil in the sump to 200°F. Normal allowable temperature rise is 90°F above ambient but not to exceed an oil-sump temperature of 200°F.

The heat generated by the gear reduction drives must, therefore, be radiated by the casing enclosing the gearing. Relatively large radiating areas must be provided, particularly where low-efficiency drives are used.

There are instances where the radiating surface of the gear reduction unit will limit the rating of the drive. It then becomes the thermal rating. Where the mechanical rating exceeds the thermal rating, the selected gear reduction drive may be used provided that direct cooling by means of water-cooling coils or oil cooling by heat exchanger is employed.

HERRINGBONE-GEAR REDUCTION UNITS

Herringbone-gear reduction units operate quietly, produce uniform torque, and transmit smooth, even flow of power free from vibration or pulsation.

The units, normally, are for floor mounting with parallel shafts. Input and output shafts may extend on same or opposite sides. Units also may be arranged with one shaft above the other or for wall mounting where conditions require this.

In these units, the gears are located symmetrically, not offset or staggered. The symmetrical arrangement of the gears permits the use of two standard bearings of identical size on each shaft, the equidistant spacing resulting in equal loading of the bearings.

The gears may be of the continuous herringbone type, generally known as the "Sykes form," or opposed single helical gears.

A sectional view of a double-reduction herringbone-gear unit is shown by Fig. 14-75. It indicates the symmetrical arrangement of gears and bearings.

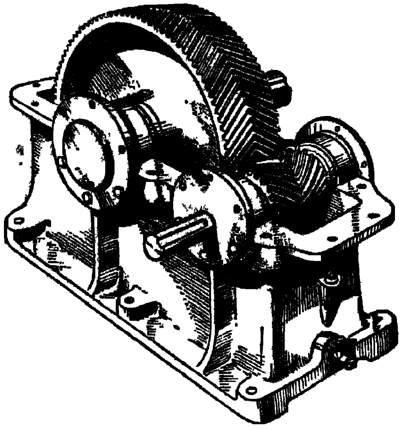


Fig. 14-76. Single-reduction herringbone-gear unit.

Figures 14-76, 14-77, and 14-78 show, respectively, single-reduction, double-reduction, and triple-reduction units with covers removed.

A double-reduction herringbone unit rated 50 hp at 1,150 rpm, ratio 45.8:1, is shown in Fig. 14-79 driving a chemical-plant dryer.

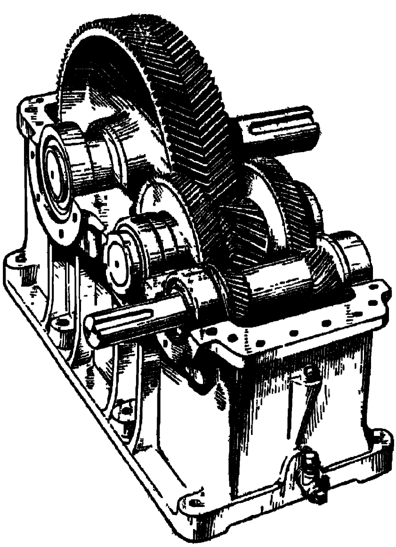


Fig. 14-77. Double-reduction herringbone-gear unit.

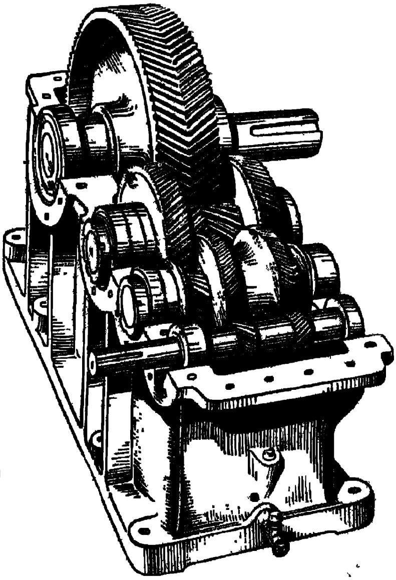


Fig. 14-78. Triple-reduction herringbone-gear unit.

High-speed Herringbone-gear Reduction Unit. For applications where shaft speeds and pitch-line velocity exceed the limits stated, high-speed herringbone units are available. These units are equipped with special force-feed lubrication of gears and a water-cooled heat exchanger to dissipate the heat generated in high-speed operation. Special bearings are employed Noise and vibration are held to a minimum through the balancing of the gears combined with rigid bearing mounting. Such units are used for turbine drives, to pumps or other machinery, and for speed increasers from low-speed motors or engines to high-speed pumps, fans, etc.

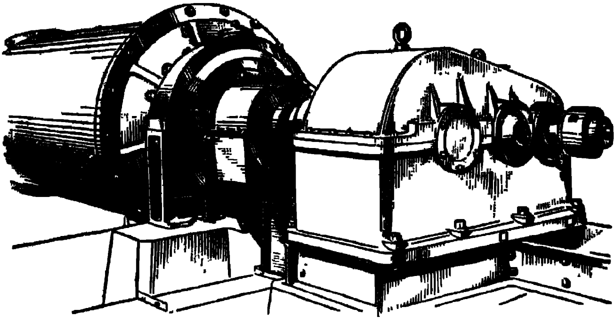


FIG. 14-79. Typical installation of double-reduction herringbone-gear unit.

Figure 14-80 shows a high-speed herringbone unit complete with pump, filter, cooler, and piping.

Helical-gear Reduction Units. Helical-gear reduction units are normally built in the same arrangement and the same housing employed for herringbone-gear reduction units. The features, service factors, and application data are the same as for herringbone-gear reduction units.

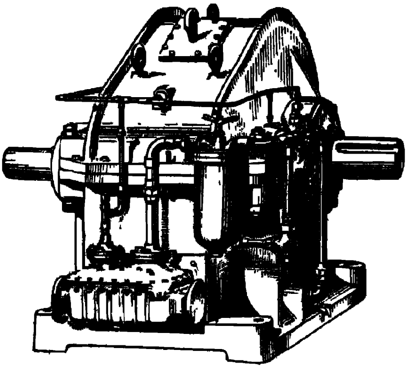


FIG. 14-80. High-speed herringbone-gear unit equipped with oil pump, filter, and cooler.

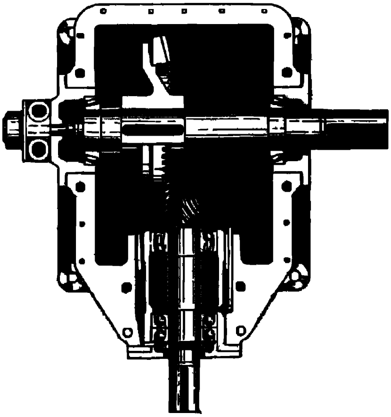


FIG. 14-81. Sectional view of spiral-bevel-gear reduction unit.

An additional load is imposed on the bearings owing to the thrust action of the single helical gears and must be considered in bearing selections.

Spiral-bevel-gear Reduction Units. The parallel-shaft speed reducer of the herringbone-gear and helical-gear type combine simplicity of design with comparatively low cost. These units are utilized in many power-transmission applications. However, when space is limited, it is often necessary to take advantage of the compactness offered by the right-angle drive.

The right-angle drive has certain decided advantages when compared to the parallel-shaft drive, and where the low-speed shaft must be vertical, the use of the right-angle drive is preferable.

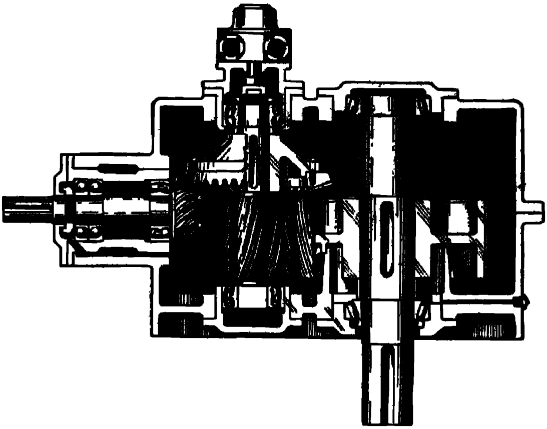


Fig. 14-82. Sectional view of vertical spiral-bevel helical-gear unit.

For single-reduction units, only spiral bevel gears are used. For double and triple reductions, helical gears or herringbone gears are used.

Sectional view of horizontal spiral-bevel-gear reduction unit is shown by Fig. 14-81. Vertical spiral-bevel helical-gear double-reduction unit, sectional view, is shown by Fig. 14-82. Sectional view of horizontal spiral-bevel helical-gear triple-reduction unit is shown by Fig. 14-83.

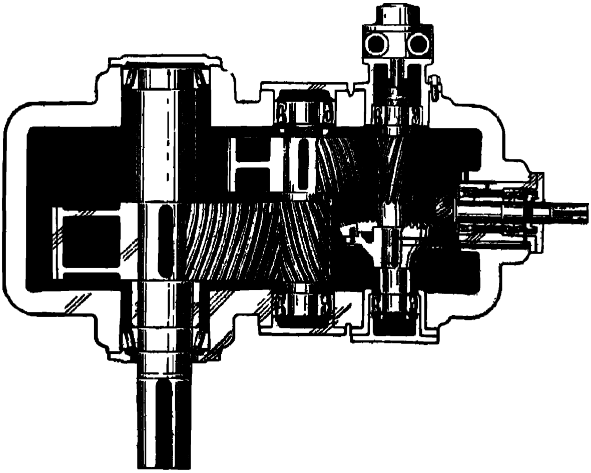


Fig. 14-83. Sectional view of horizontal spiral-bevel helical-gear unit.

Arrangement of Shafts and Rotations. When purchasing spiral-bevel or combination right-angle drives, the hand of assembly must be specified for horizontal units, and the direction or rotation of both input and output shafts must be given for all

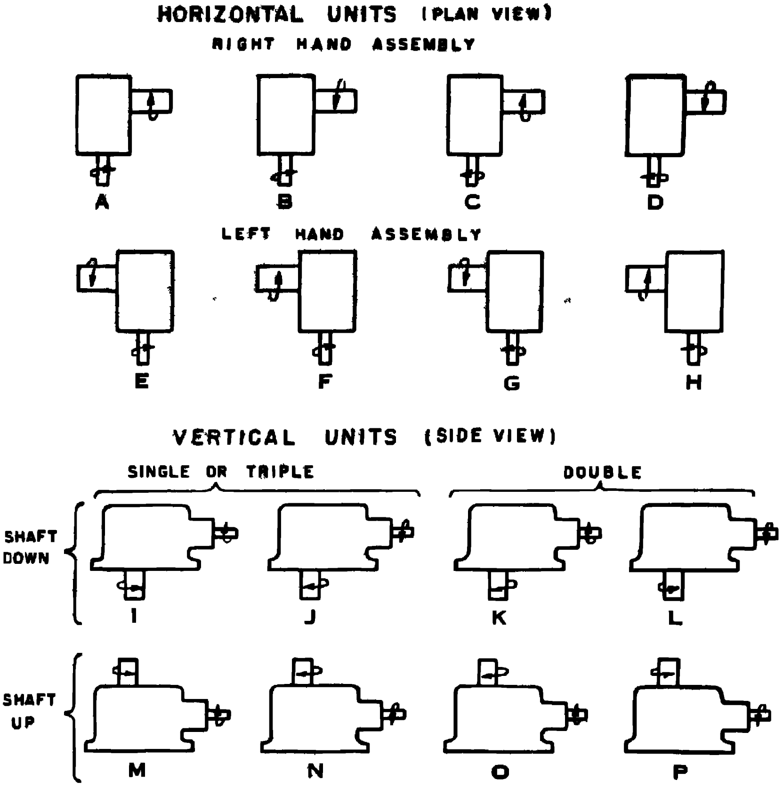


Fig. 14-84. Proper shaft arrangement and rotation for spiral-bevel and combination right-angle drives.

units. In most instances, the direction of rotation of the output shaft only is fixed, and the input shaft is driven by a motor which may rotate in either direction. Proper shaft arrangement and rotation are shown by Fig. 14-84.

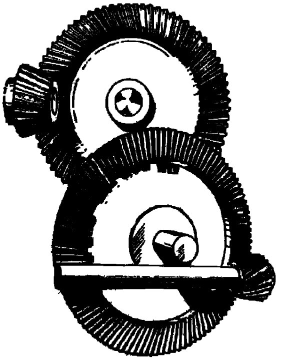


Fig. 14-85. Hypoid gears.

Hypoid-gear Reduction Units. General features of the hypoid reduction unit are those of the spiral-bevel-gear reduction unit. However, the hypoid gear has the additional advantage over the spiral bevel gear in that it permits transmission of power between offset shafts.

Hypoid-gear reduction units are designed for transmitting power between shafts at practically any angle. These drives should be used for speeds in excess of 800-fpm pitch-line velocity where smooth and quiet operation, together with minimum size, is required. Continuous pitch-line contact of hypoid gears makes it practical to obtain superior performance with a smaller number of teeth in the pinion than is possible with straight bevel gears.

The shaft offsets for all ratios above 3:1 should not exceed 20 per cent of the diameter of the gear. For 1:1 ratio, the maximum offset should not exceed 30 per cent of the diameter of the gear. Ratios between miter gears and 3:1 ratio should have proportionate shaft offsets (see Fig. 14-85).

GEARMOTORS

Gearmotors are built in many forms, using a wide variety of gears to meet practically every kind of power-transmission and speed-reduction application. Of the several types of gearing, the planetary finds extensive use for certain drives. Nonplanetary gearing is also employed. Gearmotors may be of the parallel- or concentric-shaft types.

Spur, helical, and, in some unusual cases, herringbone gears are used for both planetary and nonplanetary systems.

The simple planetary system consists of a sun pinion driving several idler gears mounted in a spider inside a fixed internal ring gear meshing with the idlers. The output shaft is part of the idler spider. The three-idler system is the one most widely used. All rotating elements are normally mounted on antifriction bearings and contained in a single enclosure.

Gearmotors are available in both horizontal and vertical types, with the motor built in or bolted on. Motors are standard open type, totally enclosed, fan cooled, totally enclosed nonventilated, explosionproof, or with other variations, both electrical and mechanical.

Figure 14-86 shows the vertical type of two-stage reduction. Figure 14-87 shows horizontal-type two-stage reduction with built-in motor. Figure 14-88 shows nonplanetary type three-stage reduction-gear motor with bolted-on motor.

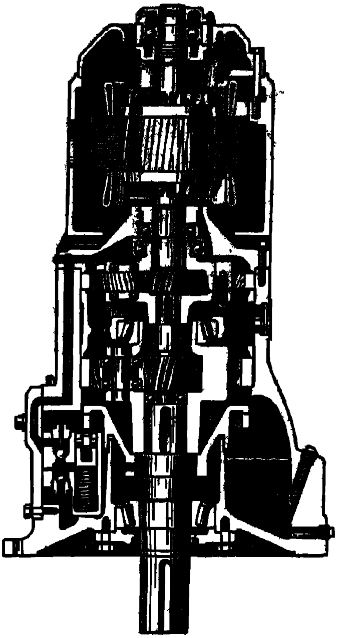


Fig. 14-86. Vertical two-stage gearmotor.

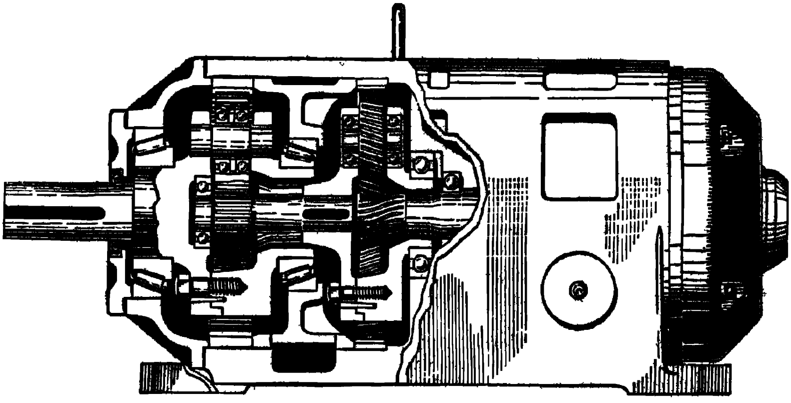


Fig. 14-87. Horizontal two-stage gearmotor.

Gearmotors fall within three general classifications of the American Gear Manufacturers Association. These are as follows:

Class I. For steady loads not exceeding normal rating of motor and 8-hr-per-day service; moderate shock loads where service is intermittent

Class II. For steady loads not exceeding normal rating of motor and 24 hr per day; moderate shock loads for 8 hr per day

Class III. For moderate shock loads for 24 hr per day; heavy shock loads for 8 hr per day

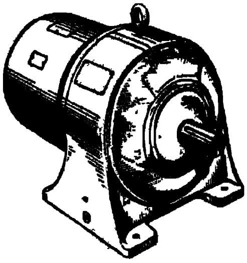


FIG. 14-88. Three-stage reduction-gear motor with bolted-on motor.

Certain typical and more common gearmotor applications have been classified in Table 14-100 according to the nature of the load and the usual duty cycle into the three commonly recognized classifications. No attempt has been made to include all possible applications of gearmotors, but a sufficient number and variety have been covered to serve as a guide for proper application classification.

Extreme repetitive shock and those applications where exceedingly high energy loads must be absorbed, as when stalling, require special consideration and are, therefore, not covered by classes I, II, or III. These include gearmotors for application to runout tables, sheet catchers, etc. Certain types of machines equipped with flywheel on the output shaft may come under this classification and should be checked to determine that the flywheel is adequate to absorb the peak loads. Flywheels are not to be used on the input shaft of gearmotors.

Where motors are equipped with brakes and the torque rating of the brake exceeds the rating of the motor, the rating of the brake dictates the selection of the gear.

Planetorque Reducers. Gearmotors are available with a built-in overload protective device known as "Planetorque." This protective device, operable in either direction of shaft rotation, is ideal for applications like stoker drives, conveyor drives, mixers, roll drivers, etc., where overloads would stall and seriously damage expensive machinery.

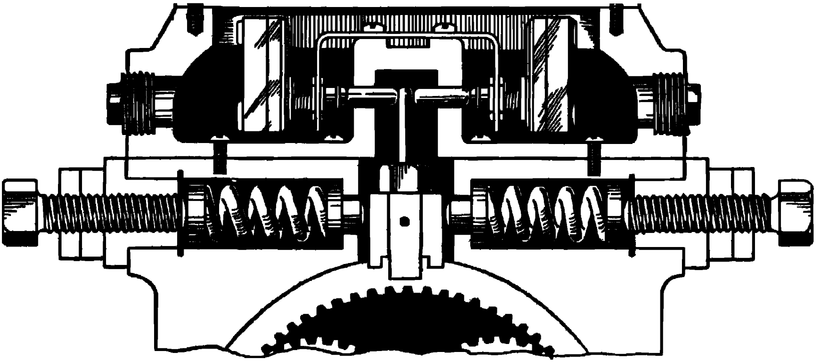


FIG. 14-89. Sectional view of the Planetorque device.

A sectional view showing internal construction of the Planetorque device is shown in Fig. 14-89. Figure 14-90 shows a horizontal gearmotor equipped with Planetorque device.

Figure 14-89 shows the internal ring gear of the planetary set arranged in the housing so that it may rotate. However, the ring gear is restrained from rotating by the compression springs on either side of the projection fastened to the periphery of the ring gear.

Table 14-100. Typical Gearmotor Applications

Application	Class	
	8- to 10-hr service	24-hr service
Agitators		
Pure liquid	1	2
Variable density	2	2
Blowers		
Centrifugal	1	2
Lobe	2	2
Brewing and distilling		
Bottling machinery	1	2
Brew kettles, continuous duty	2	2
Cookers, continuous duty	2	2
Mash tubs, continuous duty	2	2
Scale hopper (frequent starting peaks)	2	2
Car dumpers	2	3
Car pullers	2	3
Clarifiers	1	2
Classifiers	2	2
Clay-working machinery		
Brick press	3	3
Briquette machine	3	3
Clay-working machinery	2	2
Pug mill	2	2
Compressors		
Centrifugal	1	2
Lobe	2	2
Reciprocating, multicylinder—Adequately flywheeled (within 3 per cent cyclic variation)	2	3
Reciprocating, single cylinder	Refer to factory	
Conveyors (uniformly loaded or fed)		
Apron	1	2
Assembly	1	2
Belt	1	2
Flight	1	2
Oven	1	2
Screw	1	2
Conveyors (heavy-duty or dual drive, not uniformly fed)		
Apron	2	2
Assembly	2	2
Belt	2	2
Flight	2	2
Oven	2	2
Screw	2	2
Reciprocating	3	3
Shaker	3	3
Cranes and hoists		
Main hoists, medium duty	2	2
Main hoists, heavy duty	3	3
Skip hoists	2	2
Travel motion	2	2
Trolley motion	2	2
Dredges		
Cable reels	2	2
Conveyors	2	2
Cutterhead drives	3	3
Jig drive	3	3
Screen drive	3	3
Stackers	2	2
Utility winches	2	2
Fans		
Centrifugal	1	2
Cooling towers	2	2
Large (mine, etc.)	2	3
Light small diameter	1	2
Food industry		
Beet slicer	2	2
Cereal cookers	1	2
Dough mixer	2	2
Meat grinder	2	2
Hoists (see Cranes)		
Laundry washers	2	2
Laundry tumblers	2	2
Line shafts		
Driving processing equipment	2	2
Other line shafts	1	2
Machine tools		
Auxiliary drives (feed, traverse, etc.)	1	2
Notching press (belt driven)	1	2
Punch press (gear connected to load) and shears	3	3
Plate planers	3	3
Other machine tools, main drives	2	2
Metal mills		
Drawbench carriage and main drives	3	3
Forming machines	3	3
Pinch, dryer, and scrubber rolls (reversing).	Refer to factory	
Slitters	3	3
Small rolling-mill drives	3	3
Table conveyors (nonreverse)	2	2
Table conveyors (reversing)	Refer to factory	
Wire-drawing and, flattening machines	3	3
Mills (rotary type)		
Ball	3	3
Cement kilns	..	3
Dryers and coolers	..	2
Hammer	3	3
Kilns	2	3
Pebble	2	3
Rod	3	3
Tumbling barrels	3	3
Mixers		
Concrete mixers, continuous	2	2
Concrete mixers, intermittent	1	
Constant density	1	2
Irregular density	2	2
Oil industry		
Oil-well pumping (not over 150 per cent peak torque)	3	3
Refineries		
Chillers	2	2
Paraffin filter press	2	2
Rotary kiln	2	2
Paper mills		
Agitators (mixers)	..	2
Bleacher	..	2
Beater and pulper	..	3
Calenders	..	3
Conveyors	..	2
Couch	..	3
Cylinders	..	2
Dryers	..	3
Felt stretcher	..	2
Jordans	..	3
Presses	..	3

Table 14-100. Typical Gearmotor Applications (*Cont.*)

Application	Class	
	8- to 10-hr service	24-hr service
Paper mills (*Cont.*)		
Stock chests	..	3
Suction roll	..	3
Winders	..	2
Rubber industry		
Mixer	3	3
Rubber calender	2	3
Rubber mill	3	3
Sheeter	3	3
Tire-building machines	2	2
Tire- and tube-press openers	1	1
Tubers or strainers	2	2
Pumps		
Centrifugal, with surge tanks or equivalent	1	2
Centrifugal, without surge tanks	2	2
Gear and rotary, constant-density fluid	1	2
Gear and rotary, variable-density fluid	2	2
Proportioning pumps	2	3
Reciprocating, with open discharge	1	2
Pumps (*Cont.*)		
Reciprocating—multicylinder, double acting	2	2
Reciprocating, single cylinder	Refer to factory	
Sewage-disposal equipment		
Inside service	1	2
Screens		
Rotary, stone or gravel	2	2
Traveling water intake	1	1
Textile industry		
Batchers	2	2
Calenders	2	2
Card machine	2	3
Dry cans	2	2
Dyeing machinery	2	2
Looms	Refer to factory	
Mangles	2	2
Nappers	2	2
Soapers	2	2
Spinner	2	2
Tenter frames	2	2

Spring loading may be varied by means of the adjusting screw.

Thus, when the setting of the spring has been reached owing to overloading of the drive, the internal gear tends to rotate, compressing the spring and actuating the switch in the enclosure directly above the springs. Motor control circuit is automatically broken, and the equipment remains inactive until the cause of the overload has been determined and corrected.

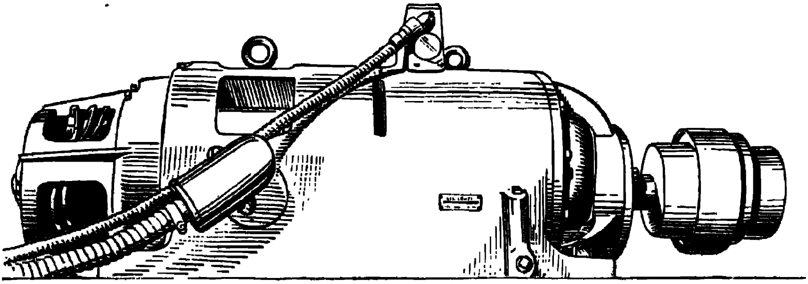


FIG. 14-90. Horizontal gearmotor equipped with Planetorque device.

On removal of the excessive load, equipment can be started without resetting the switch.

This device offers many advantages over other protective devices such as shear pin couplings, thermal overload relays, etc.

WORM-GEAR REDUCTION UNITS

Although worm gearing has been employed since the early centuries, it has been only within the last 30 years that the most rapid progress has been achieved. This has been due, primarily, to development of equipment for production of modern designs of exceptionally high efficiency.

Worm-gear reduction units transmit power by continuous shockless action. Three or more teeth are in contact at all times, giving an even flow of torque and uniform angular velocity, eliminating vibration, pulsation, and chatter.

The worm-gear reduction unit is a compact right-angle drive that provides larger ratios of speed reduction in a single stage than available in other types of gear reduction units.

Worm gears are normally produced from a special gear bronze, the worm normally from an alloy steel, the threads being carburized, hardened, tempered, machine ground, and polished. The bronze gear teeth necessarily maintain their correct form owing to the regenerating action of the hardened-steel worm, which retains its original shape throughout years of wear. A worm-gear reduction unit actually improves with service. This fact is peculiar to worm gearing and contributes to its recognized long life.

Assuming proper lubrication, worm-gear drives operate with little attention even under the most adverse conditions. The small number of moving parts in a worm-gear drive tend to safety and ease of maintenance.

Figure 14-91 shows the section through the worm and worm-gear shafts of a drive where the worm is placed below the worm gear.

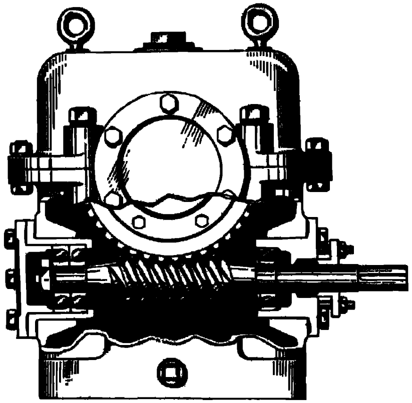


Fig. 14-91. Section through worm-gear unit with worm below worm gear.

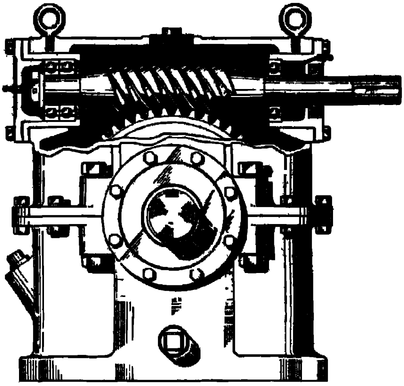


Fig. 14-92. Section through worm-gear unit with worm above worm gear.

Figure 14-92 shows a section through worm of a drive where the worm is assembled above the worm gear.

Figures 14-93 and 14-94 show sections through the gear shaft of vertical units, Fig. 14-93 showing the shaft projecting downward, Figure 14-94, with the shaft projecting upward.

Figure 14-95 shows a sectional view of a two-stage or double-reduction worm-gear unit.

Worm-gear reduction units are normally applied in accordance with the following recommended classifications of the American Gear Manufacturers Association:

Class I—Normal 8- to 10-hr service, free from recurrent shock loads (*i.e.*, shock loads that recur at approximately even and frequent intervals). The mechanical ratings of the manufacturer are to be applied directly in this classification, provided only that the thermal rating is not exceeded.

Class II— Eight to ten hours' service where recurrent shock loading is encountered, or 24-hr service where no shock loading is experienced. The mechanical ratings of the manufacturer must be reduced by dividing by 1.2 for this service. The thermal rating must not be exceeded.

Class III—Twenty-four hours' shock-load service. The mechanical rating must be reduced by dividing by 1,333 for this service, and the thermal rating must not be exceeded.

Class IV—Intermittent service where the maximum cycle of operation calls for not more than 15 min running in a 2-hr period. The mechanical rating may be divided by 0.7 to obtain a rating for the service classification. The thermal rating may be ignored.

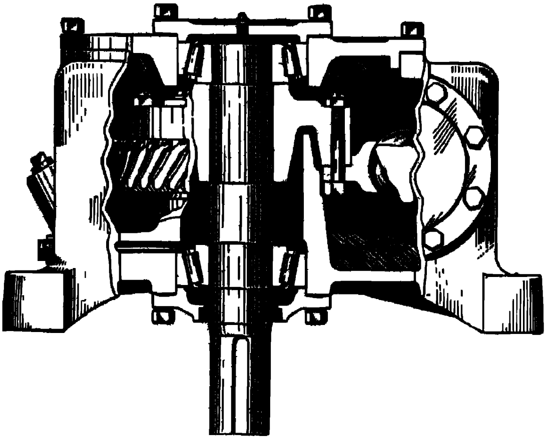


FIG. 14-93. Section through vertical worm-gear unit with driven shaft downward.

Class V—Low-speed service (where worm speed is less than 100 rpm) will require output-torque ratings in inch-pounds, independent of the nature or duration of the loads.

Worm-gear reduction units are rated mechanically and thermally by all manufacturers. The mechanical rating is based on the wear resistance of the gear teeth. The worm-gear tooth form is such that the gear teeth are under a crushing rather than a bending load, and for this reason extremely high momentary and shock loads damaging to many forms of gearing can be successfully withstood. The beam or structural strength of the tooth is considerably in excess of the wear-resistant rating.

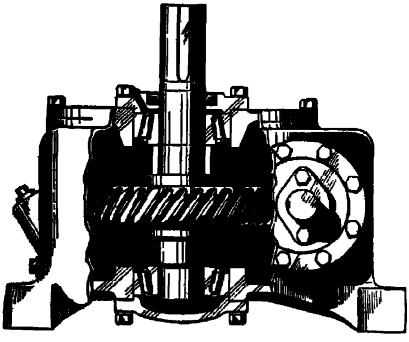


FIG. 14-94. Section through vertical worm-gear unit with driven shaft upward.

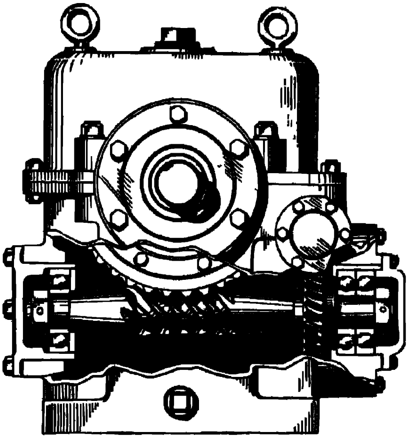


FIG. 14-95. Section through double-reduction worm-gear unit.

The thermal rating indicates the amount of power that can be delivered through the unit continuously without overheating the unit. This rating is based on the radiating area of the housing and varies only with the efficiency of the unit which in turn is dependent on the ratio of reduction. Thermal ratings, therefore, are independent of service classification.

For continuous service, the unit must be selected according to the lesser of the two ratings. If the thermal rating is lower than the mechanical rating, the unit must be selected on the thermal rating. However, for class IV service, the thermal rating may be neglected, for the service classification does not permit sufficient time for the unit to heat up.

Fan Cooling. The use of coils carrying circulating water to reduce temperature of the oil bath is employed to increase the thermal capacity of worm-gear reduction units. But water usage in such volume to be adequate for cooling is not economical in some plants. Some applications utilize a circulating system for the lubricant and a heat exchanger. But this is costly. To improve upon these methods and to increase the over-all efficiency of worm-gear reduction units, the fan-cooled unit has been developed.

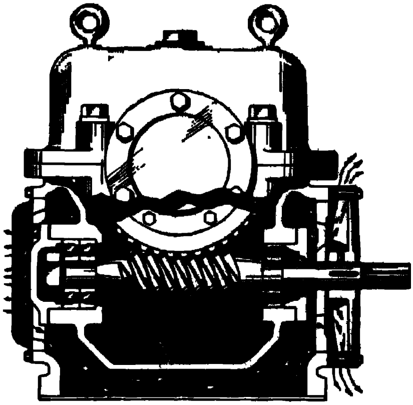


Fig. 14-96. Fan-cooled worm-gear reduction unit.

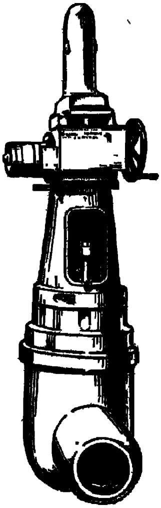


Fig. 14-97. Gear-unit valve control mounted on gate valve.

Conventional worm-gear reduction units are provided with dual walls. The ribbed inner housing provides a large heat-exchange surface. The outer housing wall which carries gear and shaft loads is kept cool, since it does not contact the oil bath. Thus, distortion due to temperature changes is eliminated, and bearings, shaft, and gears are maintained in proper alignment.

A fan is mounted on the worm shaft. This fan draws air from the opposite end of worm shaft over the ribbed inner housing reducing the temperature of the oil bath to approximately 200°F. Temperature of the drive is thus held to a limit permitting satisfactory lubrication and accordingly increasing capacity over drive without such cooling feature.

As an example, suppose the requirements were for 20-hp at 1,750 rpm, ratio 4.667:1. A worm-gear unit of 6.75 in. center distance would be needed to satisfy this requirement when considering the standard type. The fan-cooled unit of 4.75 in. center distance would amply satisfy these requirements.

Figure 14-96 illustrates the fan-cooled worm-gear reduction unit showing a view through the worm shaft.

Electric Operators for Valves. The characteristics of electric operators for valves parallel, in so far as the gear reduction unit is concerned, the conventional worm-gear reduction unit. While valve operators are subject to momentary peak loads at the time of shutting off the valve or cracking the seat of the valve, the loading at other times is negligible involving usually the weight of the moving parts and friction only.

The gear reduction unit is equipped with a flange-mounted motor rated in pounds-feet torque and of short-duty cycle. Valve-operator motors are infrequently rated in horsepower.

The gear reduction box is also equipped with switches to control the opening and closing travel of the valve, as well as switches to provide actuation of indicating-light system and interlocking system where necessary.

Housed in the gear reduction unit is the valve yoke nut which raises or lowers the stem, opening or closing the valve. Valves of the nonrising-stem type are also driven by the yoke nut within the valve-control gearbox, but instead of being threaded to mate with the valve stem, the yoke nut is bored and keywayed to fit the stem of the nonrising-stem valve.

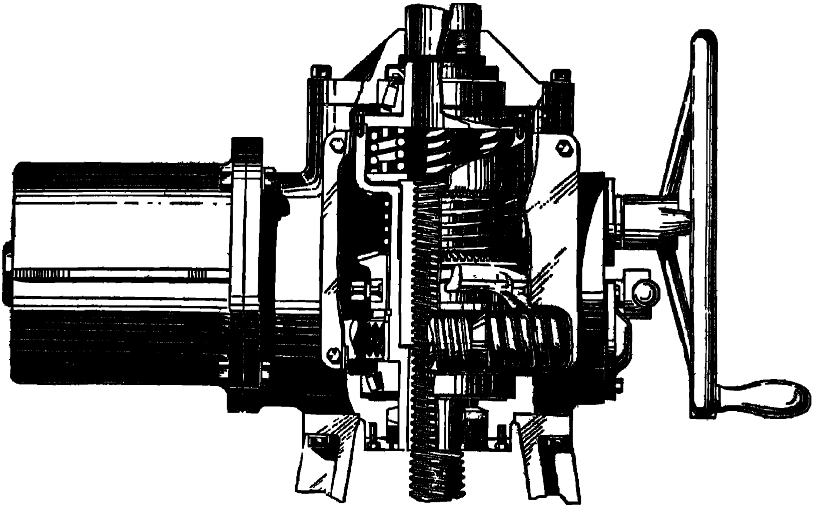


FIG. 14-98. Sectional view of valve operator.

Figure 14-97 shows valve control mounted on a gate valve. Figure 14-98 shows a sectional view of what is termed the operator.

A handwheel is provided for manual operation. The handwheel does not revolve during motor operation. During hand operation, the motor is mechanically disengaged and does not rotate. A lever is provided to change from motor to hand operation, but the unit returns automatically to motor operation.

A spring compensator is provided which permits the valve stem to expand or contract with temperature changes but ensures tight closure of the valve regardless of temperature variations.

A torque seating switch ensures tight valve closure without damage to the valve or the mechanism. This switch may be adjusted to the desired seating thrust and can be varied over a wide range. The torque switch also protects the valve in the event of an obstruction being encountered during the closing operation.

What to Do When Trouble Occurs. Heating does not mean trouble. Overheating does mean trouble. If the temperature of the oil-bath lubricant is 200°F, this may be considered normal for most gear reduction units. When this temperature is exceeded, a search should be made for the cause.

Naturally there are spot temperatures, such as the teeth at point of mesh, that are higher than 200°F. But if the oil in the sump is in the neighborhood of 200°F, the unit can be considered operating normally.

Table 14-101. Trouble Chart

Trouble	*What to inspect*
Heating	Is unit overloaded?
	Has recommended oil level been exceeded, or is level too low?
	Are couplings in alignment?
	Have bearings been properly adjusted?
	Are oil seals or stuffing boxes the cause?
	Perhaps breathers are not clean.
	Is oil clean, or is sludge content high?
	Have you cleaned oil filter?
	Is oil pump functioning?
Shaft failure	Check alignment. Most shafts fail owing to misalignment.
	Some troubles are caused by use of rigid couplings.
	Is overhung load beyond capacity of unit?
	Is unit subject to high energy loads or extreme repetitive shocks not previously considered?
Bearing	Rust formation due to entrance of water or humidity.
	Unsuitable lubricant.
	Abnormal loading causing excessive deflection results in flaking, cracks, fractures.
	Improper adjustment causes abnormal loading if bearings are pinched or abnormal gear wear if bearings are too free, dependent upon type of bearing.
	Lack of lubrication.
Oil leakage	Check oil seals and replace if worn.
	Check stuffing boxes and adjust or replace packing.
	Check tightness of drain, level, and other plugs or fittings.
Wear	Backlash may be insufficient.
	Misalignment due to worn bearings.
	Incorrect lubrication.
	Insufficient lubrication.
	Lubricant carrying foreign matter, viz., abrasive dirt or particles of metal worn from teeth.
	Excessive temperature.
	Excessive speeds.
	Excessive loads.
Noise	Bad alignment.
	Loose or worn bearings.
	Insufficient lubrication.
	Lack of lubrication.
	Excessive lubrication.

While gear reduction units are suitable for momentary overloading up to 100 per cent of their normal rating, there have been instances where this permissible momentary overloading has been exceeded. The user frequently overloads the unit up to 100 per cent or more for a period of from several minutes to perhaps an hour or more daily. Such cases of overloading cause excessive heating during the period of overload. And this, while seldom resulting in breakage immediately, does cause more rapid wear of the gear teeth and in time may be the cause of bearing or shaft breakage.

Improper lubrication certainly covers a large percentage of gear reduction-unit failures. Too frequently units are started up without any lubricant at all. Sometimes the belief seems to be that a higher oil level than called for is better lubrication. This usually results in more of the input power going into churning of the oil creating excessive temperatures with detrimental results to the bearings and gearing. Likewise, insufficient lubricant effects the same results.

Almost 98 per cent of gear reduction unit failures have been attributable to misalignment, improper lubrication, and misapplication. Therefore, if these major points are recognized, the modern gear reduction unit will perform satisfactorily.

AIRFLEX CLUTCHES

BY THE DIRECTOR OF SALES AND ENGINEERING

Fawick Airflex Co., Inc.

The Airflex clutch was developed in an effort to apply sound and thoroughly proved mechanical principles to obtain the maximum efficiency of power transmission using conventional materials. Airflex clutches are manufactured in five basic design types.

CLUTCH TYPES

In three of the types, the torque is transmitted through the rubber tube, thus providing resilient cushioning action between the driving and the driven shafts. Type CB, shown by Fig. 14-99, with dimensions, is a constricting design with the friction-shoe assemblies mounted on, and driven by, the rubber tube. Type EB is an expanding design with the friction shoes similarly mounted. Type ER is an expanding design driving through direct contact between the rubber tube and the clutch drum. These types are shown by Fig. 14-100 with dimensions. In two of the types, the torque is transferred directly through torque bars from the friction-shoe assemblies to the clutch housing, the air tube acting only as the means of applying uniform pressure. Type C is a constricting design, and type E is an expanding design. These types are shown by Fig. 14-101 with dimensions. The variations in construction of the several types are shown in Figs. 14-99, 14-100, and 14-101.

Common Characteristics for all Types.

1. All portions of the friction block contact the drum at equal velocity.
2. All friction blocks contact the drum with uniform pressure.
3. Approximately 360-deg radial contact is obtained between friction blocks and the drum.
4. All designs are pneumatically operated.
5. Disengagement is complete when air pressure is released. When the clutch is in the released position, an open air space between the drum and the friction shoes eliminates any possibility of drag effect which would result in heat, wear, or noise.
6. Friction-shoe assemblies are removable and replaceable without special tools or equipment.
7. All types are self-compensating for wear and require no adjustment throughout the life of the friction block.
8. Automatic compensation eliminates any negative effects of misalignment since pressure balancing occurs in the air tube.
9. Control is accomplished by manually or electrically operated valve. This valve can be either of the "on-and-off" type or of the modulating (controllable variable pressure) type. Remote control or multiple-station control can be obtained through the use of solenoid-operated valves with proper accessory equipment.
10. Maximum heat dissipation is ensured since drum design permits maximum circulation of air.
11. The three types in which the torque is transmitted through the rubber tube ensure prolonged life and reduced maintenance on all power-transmission elements through the elimination of peak shock loads.
12. The three types in which the torque is transmitted through the rubber tube compensate automatically for accidental and reasonable angular- and parallel-shaft misalignment.
13. Since no adjustment is required, all possibility of improper adjustment is eliminated.

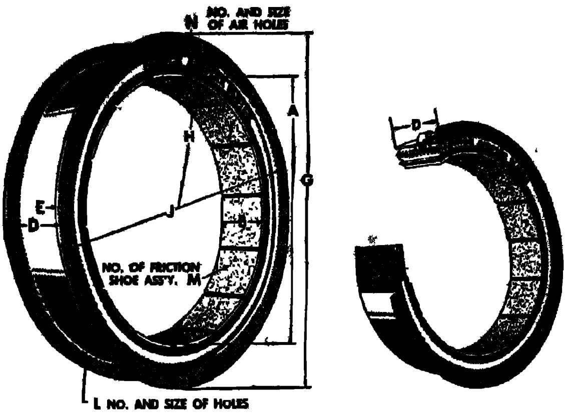


Fig. 14-99. Clutch type CB elements.

Type CB

Clutch size*	Torque rating, in-lb†	Weight, lb	Dimensions, in.									
			A	*B*	*D*‡	*E*	*G*	*H*	*J*	*L*§	*M*	*N*
6CB200	2,040	7	6 5/32	2	2 15/16	5/32	10 752 $^{+000}_{-005}$	5	10	8 377	6	1 5/16
8CB250	4,290	9	8 5/32	2 1/2	3 7/16	5/32	12 875	6 1/16	12 1/8	8.377	8	1 5/16
10CB300	8,150	19	10 1/8	3	4 1/8	3/16	15.375	7 5/16	14 5/8	12.377	10	1 5/16
12CB350	13,300	26	12 1/8	3 1/2	4 23/32	3/16	17 625	8 7/16	16 7/8	14.377	12	1 5/16
14CB400	19,700	31	14 1/8	4	5 7/32	3/16	19.625	9 7/16	18 7/8	16.377	14	1 5/16
16CB500	35,200	74	16 5/16	5	6.625	7/32	23.500	11 1/8	22 1/2	8.502	10	1 3/8
18CB500	44,000	81	18 5/16	5	6.625	7/32	25.500	12 3/16	24 3/8	12.502	11	1 3/8
20CB500	53,600	88	20 5/16	5	6.625	7/32	27.500	13 3/16	26 3/8	12.502	12	1 3/8
22CB500	62,300	95	22 5/16	5	6.625	7/32	29.500	14 3/16	28 3/8	12.502	13	1 3/8
24CB500	75,000	102	24 5/16	5	6.625	7/32	31.500	15 3/16	30 3/8	16.502	14	1 3/8
26CB525	92,400	133	26 5/16	5 1/4	6.938	9/32	34.000 $^{+000}_{-008}$	16 1/4	32 3/4	16.627	16	1 1/2
28CB525	106,000	140	28 5/16	5 1/4	6.938	9/32	36.000	17 1/4	34 3/4	16.627	17	1 1/2
30CB525	121,000	148	30 5/16	5 1/4	6.938	9/32	38.000	18 1/4	36 3/4	16.627	18	1 1/2
32CB525	137,000	157	32 5/16	5 1/4	6.938	9/32	40.000	19 1/4	38 3/4	18.627	19	1 1/2
36CB525	172,000	178	36 5/16	5 1/4	6.938	9/32	44.625 $^{+000}_{-010}$	21 5/8	43 1/8	18.752	22	1 5/8
40CB525	211,000	201	40 5/16	5 1/4	6.938	9/32	48.625	23 5/8	47 1/8	20.752	24	1 5/8
45CB525	260,000	262	45 5/16	5 1/4	6 938	9/32	53.625	26 1/8	52 1/8	24.752	27	2 5/8
50CB650	406,000	460	50 5/16	6 1/2	8.938	7/16	62.000	30	60 1/2	28.877	30	4 3/8
55CB650	490,000	520	55 5/16	6 1/2	8 938	7/16	67.000	32 1/2	65 1/2	30.877	33	4 3/8
60CB650	577,000	600	60 5/16	6 1/2	8.938	7/16	72.000	35	70 1/2	32.877	36	4 3/8
66CB650	707,500	725	66 5/16	6 1/2	8.938	7/16	78.000	38	76 1/2	32.877	40	4 3/8
72CB650	838,000	870	72 5/16	6 1/2	8 938	7/16	84.000	41	82 1/2	36.877	44	4 3/8
84CB650	1,130,000	1,200	84 5/16	6 1/2	8.938	7/16	96.000	47	94 1/2	42.877	50	4 3/8

* Drum diameter is exact inch size +0.000 in. −0.010 in., viz., 16CB500 drum is 16.000 in. diameter.

† Torque ratings are based on 75 psi air pressure at element at speed up to 100 rpm. For exact rating at higher speed, consult manufacturer for centrifugal-force effect.

‡ Dimensions as given may vary ± 0.010 in. Tolerance on width of rim is +0.000 in. −0.010 in.

§ All drilled hole sizes, dimension L, are subject to tolerance +0.005 in. −0.000 in.

Inset cut at right above shows single flange construction used on sizes 6CB200 to 14CB400 inclusive.

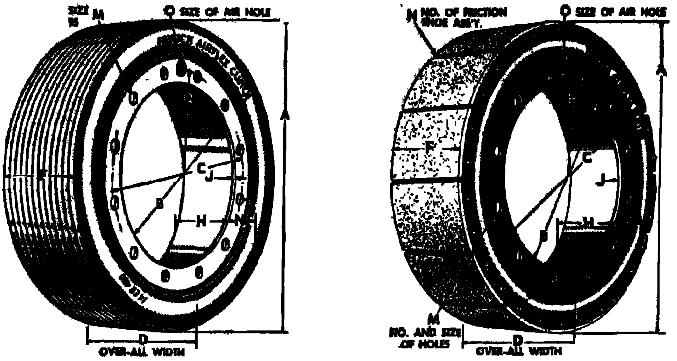


FIG. 14-100. Clutch type EB and ER elements.

Clutch size*	Torque rating, in.-lb.†	Weight, lb	Dimensions, in.											
			A	*B*	*C*	*D*	*F*	*H*	*J*	*M*‡	*N*	*O*	*Q*	*R*
						Type EB								
6EB200	910	7	5 29/32	....	...	2 3/4	2	2	..	.. ..	6	5/16	3/8	
8EB250	2,220	19	7 27/32	. ... .	....	3 1/4	2 1/2	2 1/2	.	.. ..	8	5/16	3/8	
9EB325	3,750	9	8 27/32	3.500^{+005}_{-000}	4 5/8	4 1/8	3 1/4	3 1/4	3/16	8.502	9	5/16	7/16	2 5/16
10EB300	4,275	10	9 27/32	4.250	5 1/4	3 7/8	3	3	5/32	8 502	10	5/16	7/16	2 25/32
12EB350	7,500	16	11 27/32	6 250	7 1/4	4 3/8	3 1/2	3 1/2	3/16	12 502	12	5/16	7/16	3 25/32
14EB400	12,000	23	13 27/32	8 250	9 1/4	4 7/8	4	4	3/16	12.502	14	5/16	7/16	4 25/32
16EB475	18,750	40	15 13/16	$8\ 250^{+010}_{-000}$	9 5/8	6 3/8	4 3/4	4 3/4	1/4	8 502	12	3/8	13/16	4 13/16
19EB475	27,000	49	18 13/16	9 625	11	6 3/8	4 3/4	4 3/4	1/4	6 752	14	3/8	13/16	5 1/2
21.5EB475	36,000	62	21 5/16	12 125	13 1/2	6 3/8	4 3/4	4 3/4	5/16	8 752	16	3/8	13/16	6 3/4
24EB475	45,000	68	23 13/16	14 625	16	6 3/8	4 3/4	4 3/4	5/16	8.752	18	3/8	13/16	8
28EB525	72,500	80	27 13/16	18.625	20	6 7/8	5 1/4	5 1/4	5/16	12 752	21	1/2	13/16	10 9/16
						Type ER								
6ER200	1,540	7	5 29/32	.....	..	3 1/16	2	2	...	......	1 7/32	5/16		
8ER200	3,550	18	7 29/32		..	3 9/16	2 1/2	2 1/2	..	.... .	1 7/32	5/16		
10ER300	6,600	8	9 29/32	$4\ 250^{+005}_{-000}$	5 1/4	4 1/4	3	3	5/32	8.502	5/8	5/16	2 23/32	
12ER350	12,900	13	11 29/32	6.250	7 1/4	4 3/4	3 1/2	3 1/2	3/16	12.502	5/8	5/16	3 23/32	
14ER400	22,600	17	13 29/32	8 250	9 1/4	5 1/4	4	4	3/16	12.502	5/8	5/16	4 23/32	
16ER475	32,600	31	15 13/16	8.250^{+010}_{-000}	9 5/8	6 5/8	4 3/4	4 3/4	1/4	8.502	1 5/16	3/8	4 13/16	
19ER475	47,600	39	18 13/16	9 625	11	6 5/8	4 3/4	4 3/4	1/4	6 752	1 5/16	3/8	5 1/2	
21.5ER475	63,000	52	21 5/16	12 125	13 1/2	6 5/8	4 3/4	4 3/4	5/16	8 752	1 5/16	3/8	6 3/4	
24ER475	83,500	61	23 13/16	14 625	16	6 5/8	4 3/4	4 3/4	5/16	8 752	1 5/16	3/8	8	
28ER525	115,000	72	27 13/16	18 625	20	7 1/8	5 1/4	5 1/4	5/16	12.752	1 5/16	1/2	10 9/16	

* Dimensions as given may vary ± 0.010 in.

† Torque ratings are based on 75 psi air pressure at element at speed up to 100 rpm. For exact rating at higher speed, consult manufacturer for centrifugal-force effect.

‡ All drilled hole sizes, dimension M, are subject to tolerance +0.005 in. −0.000 in.

§ Drum diameter is exact inch size +0.010 in. −0.000 in., viz., 12EB350 drum is 12.000 in. diameter.

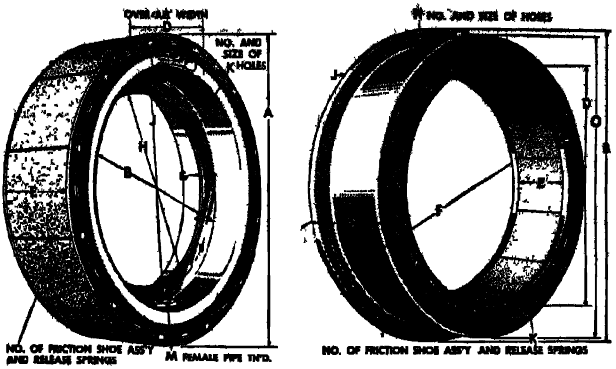

OVERALL WIDTH
D
NO. AND SIZE OF K HOLES
A
H
B
C
L
NO. OF FRICTION SHOE ASS'Y AND RELEASE SPRINGS
M FEMALE PIPE THD.
NO. AND SIZE OF HOLES
J
G
D
A
E
F
K
NO. OF FRICTION SHOE ASS'Y AND RELEASE SPRINGS

FIG. 14-101. Clutch Type E and C elements.

Clutch size*	Torque rating, in-lb†	Weight, lb	Dimensions, in											
			A	B	C	D	E	F	G	H	J	K‡	L	M
						Type E								
12E475	11,300	25	11²⁹⁄₃₂	8⅛		5⁷⁄₁₆	4¾		³⁄₁₆	6 000$^{+010}_{-000}$	7	10 377	8	¼
14E475	16,000	32	13²⁹⁄₃₂	9¹³⁄₁₆		5⁷⁄₁₆	4¾		³⁄₁₆	7 625	8¾	12 382	10	¼
16E475	21,500	42	15²⁹⁄₃₂	11¹³⁄₁₆		5⁷⁄₁₆	4¾		³⁄₁₆	9 625	10¾	6 507	12	⅜
19E475	31,500	53	18²⁹⁄₃₂	14¾		5½	4¾		¼	12 125	13¾	10 507	12	⅜
21 5E475	40,500	60	21¹³⁄₃₂	17¼		5½	4¾		¼	14 250	15¾	8 757	14	⅜
24E475	52,000	67	23²⁹⁄₃₂	19¾		5½	4¾		¼	16 750	18¼	10 757	16	⅜
27E475	67,000	75	26²⁹⁄₃₂	22¾		5½	4¾		¼	19 750	21¼	12 757	18	⅜
30E600	106,000	125	29²⁹⁄₃₂	24¹⁵⁄₁₆		7	6		⁵⁄₁₆	21 000$^{+005}_{-000}$	23	14 757	14	½
34E600	137,000	156	33²⁹⁄₃₂	28¹⁵⁄₁₆		7	6		⁵⁄₁₆	25 000	27	16 757	16	½
40E700	225,000	220	39²⁹⁄₆₄	34¹¹⁄₁₆		8⅛	7		⁵⁄₁₆	30 000	32	18 757	18	½
						Type C								
18C475	40,900	152		25½$^{+000}_{-005}$	5⅞	18³⁄₃₂	4¾	24⅜		12 502	1¹⁄₁₆	12		23
21C475	54,500	175		28½$^{+000}_{-005}$	5⅞	21³⁄₃₂	4¾	27⅜		12 502	1¹⁄₁₆	14		26
25C475	75,300	207		32¾$^{+000}_{-005}$	5⅞	25³⁄₃₂	4¾	31⅜		16 627	1¹⁄₁₆	16		30
27C600	112,000	334		37 $^{+000}_{-008}$	7⁷⁄₁₆	27³⁄₃₂	6	35⅜		16 627	⅞	14		33
32C600	155,000	394		42 $^{+000}_{-008}$	7⁷⁄₁₆	32³⁄₃₂	6	40¾		18 627	⅞	18		38
40C700	360,000	480		53⅝$^{+000}_{-010}$	9	40³⁄₃₂	7	52⅛		24 752	1⅛	18		46¼

* Drum diameter for type E is exact inch size +0 010 in −0 000 in, viz., 16E475 drum is 16 000 in diameter

† Torque ratings are based on 75 psi air pressure at element at speed up to 100 rpm For exact rating at higher speed, consult manufacturer for centrifugal-force effect.

‡ All drilled hole sizes, dimension K, are subject to tolerance +0 005 in. −0 000 in.

§ Drum diameter for type C is exact inch size +0.000 in. −0.010 in., viz., 21C475 drum is 21.000 in. diameter. Dimensions as given may vary ± 0.010 in.

Table 14-101A.—Service Factors

Type of driven machine	Service factor: Clutch coupling	Service factor: Bushed clutch
Agitators	1.2	1
Ball mills	1.8	1.5
Blowers:		
Centrifugal	1.2	1
Lobe	1.4	1.2
Brakes	Consult manufacturer	
Calenders (see paper and rubber)		
Compressors:		
Single stage	2.4	2
Multiple stage	1.4	1.2
Rotary	1.4	1.2
Conveyors: Belt, chain, screw or flight	1.2	1
Cranes:		
Bridge type Wheel or crawler type	Consult manufacturer	
Crushers	1.8	1.5
Draw-works, oil well	Consult manufacturer	
Dryers (see Kilns, or see Paper-mill drives)		
Dynamometers	Consult manufacturer	
Fans:		
Light duty	1.2	1
Induced-draft or mine ventilating	1.4	1.2
Generators	1.2	1
Grizzlies	2.4	2
Hammermills	1.8	1.5
Hogs	1.8	1.5
Hoists	Consult manufacturer	
Kilns	Consult manufacturer	
Mixers:		
Rubber mill		
Agitator type	1.2	1
Calender type	1.4	1.2
Pug mill type	1.2	1
Oil-field machinery	Consult manufacturer	
Paper-mill drives:		
Agitator	1.2	1
Calender	2.4*	2*
Dryer	2.4*	2*
Fourdrinier	1.8	1.5
Press	1.4	1.2
Suction couch	1.8	1.5
Propellors, agitator, marine propulsion (consult Falk Corporation, Milwaukee, Wisconsin)	1.2	1
Pulverizers	1.4	1.2
Pumps:		
Centrifugal	1.2	1
Single-stage reciprocating	2.4	2
Multistage reciprocating	1.4	1.2
Mud or slush	2.4	2
Punch presses	Consult manufacturer	
Road rollers:		
Engine	1.8	1.5
Transmission	1.2	1
Rod mills	Consult manufacturer	
Rubber mills:		
Single calender mills	2.4	2
Two or more calender mills	1.8	1.5
Saw-mill machinery	Consult manufacturer	
Shovels, power:		
Boom hoist Crowd Drag drum Hoist drum	Consult manufacturer	
Master engine	1.8	1.5
Propel Retract Swing Swing brake	Consult manufacturer	
Tube mills	Consult manufacturer	
Tumbling barrels	1.4	1.2
Veneer machinery	Consult manufacturer	
Wire:		
Drawing machinery	Consult manufacturer	

* Apply factors shown to starting torque load.

14. Smooth starting is obtainable through controlled application of the operating air pressure.

15. Applied torque can be sensitively regulated through a modulating (controllable variable pressure) type of control valve.

16. Torque limiting can be obtained through regulation of maximum pressure applied.

17. Location of control station can be determined independently of clutch location.

18. Minimum axial space is required for installation.

19. The standard clutch in the three types in which the torque is transmitted through the rubber tube combine the dual function of a flexible coupling and a clutch.

20. Type CB clutch is generally applicable to most industrial clutch requirements. It is equally suitable for those requirements where two shafts are to be connected and

for those applications where either the drum or the clutch element and supporting spider are bushed on a common shaft, with the other member keyed to the shaft. It makes available the greatest amount of peak-torque cushioning, as well as the reduction of vibration which would otherwise be mechanically transmitted. The rubber tube tends to eliminate these mechanical vibrations. In the type CB clutch, the rubber tube is permanently bonded to the steel rim at the time the rubber tube is cured. The friction-shoe assemblies are secured to the tube with removable shoe pins which are held in place by heavy lock wires. The friction-shoe assemblies consist of an asbestos-base synthetic-resin block permanently bonded (without rivets) to the cadmium-plated steel backing plate. These friction-shoe assemblies are removable and replaceable without use of special tools. The openings between adjacent friction-shoe assemblies are provided with stainless-steel cavity shields. These shields eliminate any possibility of harmful or destructive effect of any material that may accumulate between the friction blocks.

Service factors to be used in the selection of the proper size of an Airflex clutch are given in Table 14-101A.

Since the friction-shoe assemblies and the nonrigidly supported part of the clutch tube are subject to centrifugal-force effect, consideration must be given to the operating speed. Centrifugal-force effect at speeds up to 100 rpm is negligible. On applications operating at higher speed, proper correction for the operating speed should be made. The value of the necessary correction is shown on the service-rating chart. All torque ratings shown in the dimension tables for the various sizes of cluthces are based on 75 psi inflation pressure in the clutch tube. All Airflex clutches are suitable for operations at applied pressures up to 110 psi maximum. The necessary formulas to obtain the rating at pressures other than 75 psi are shown below the service rating graph (see Fig. 14-102).

The type EB clutch is generally applicable to most industrial clutch requirements where the operating speed is low. This limitation is imposed by the fact that centrifugal force on the expanding-type element may prevent positive disengagement on release of air pressure. It is equally applicable to those requirements where two shafts are to be connected and to those applications where either the drum or the clutch element and supporting spider are bushed on a common shaft, with the other member keyed to the shaft. Since the entire torque is transmitted through the rubber side wall of the tube, maximum cushioning against peak torques is obtained, and maximum elimination of mechanical vibration is effected. When installed as a flexible-coupling connection, compensation is provided for reasonable angular- and parallel-shaft misalignment. Torque ratings shown in the type EB dimension table are based on 75 psi inflation pressure in the tube. Correction of rating for higher or lower air pressures should be made through use of the formula shown below the type CB service-rating graph. The friction-shoe assemblies on the type EB are similar to the type used on, and described under, the type CB.

The type ER clutch is a special-purpose design intended for use on disconnecting clutch coupling applications, such as dual drives and stand-by drives. On such installations, the rubber-element assembly is properly mounted on the shaft, which normally remains idle. Inflation pressure in this type can be provided through a drilled hole in the shaft, a rotorseal and "on-and-off" valve from a compressed-air line, or be provided by manual inflation. On those applications where manual inflation is required, the element assembly is furnished with a tank valve similar to that used on automobile tires. The type ER is also suitable where low inertia values are involved and where an almost instantaneous braking function is needed. Typical of this type of installation are the brakes on the collapsible drums of tire-building machines. The drum is automatically collapsed by arresting the rotation of the shaft, allowing the inertia of the tire to unlatch the collapsing mechanism. On both the type EB and type ER, the rubber tube is permanently bonded to the steel rim at the time the rubber tube is cured. The torque ratings shown in the type ER dimension table (Fig. 14-100) are also based on 75 psi inflation pressure in the element tube.

The type E Airflex element may also be used on general industrial clutch or brake requirements. It is particularly suited to those requirements where frequent accelera-

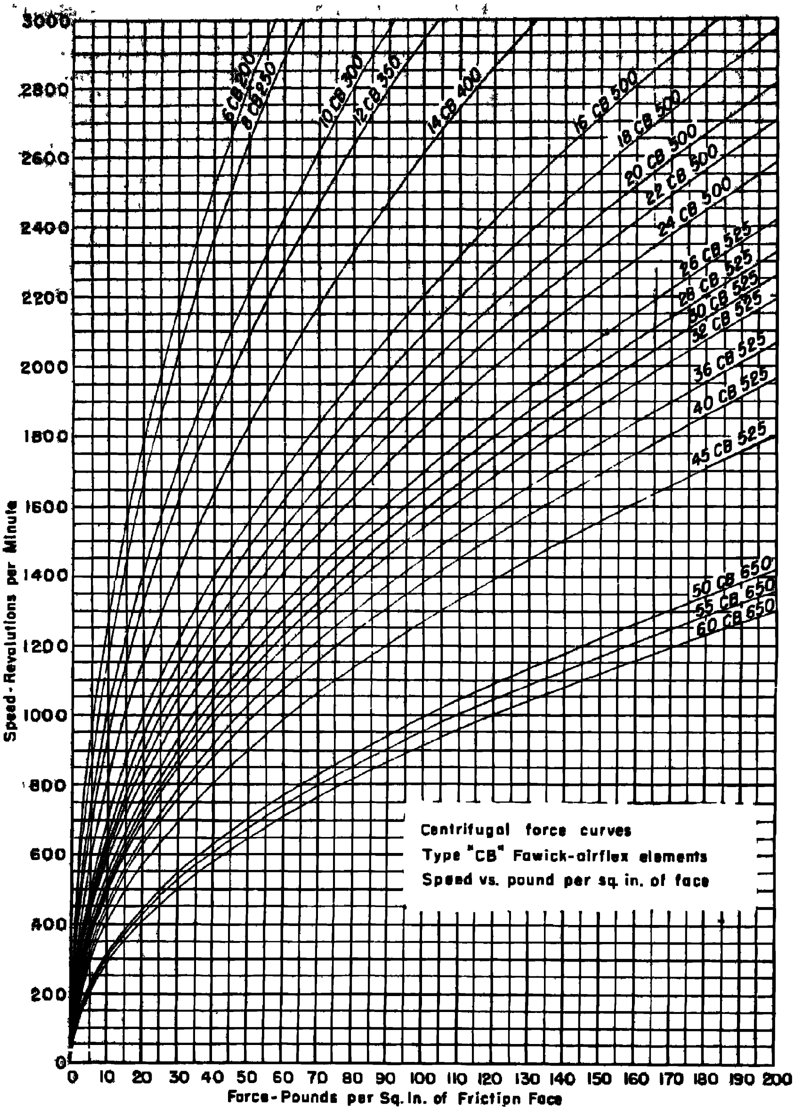


FIG. 14-102. Clutch service ratings.

For applications up to 100 rpm

$$\text{Required torque rating} = \frac{\text{hp} \times 63{,}000}{\text{rpm}} \times \text{service factor} \times \frac{75}{\text{minimum air pressure}^*}$$

For applications above 100 rpm

$$\text{Required torque rating} = \frac{\text{hp} \times 63{,}000}{\text{rpm}} \times \text{service factor}$$

$$\times \frac{75}{\text{minimum air pressure} - \text{centrifugal force}^*}$$

tion or braking duty is imposed or where continuous controlled torque slip is required. Typical of common applications are swing clutches on power shovels and drag lines; feed and return travel on sawmill carriages and similar reverse-travel mechanisms; slip clutches on coilers, slitters, and winders. Another typical application is drag brakes or differential drag mechanisms on ferrous and nonferrous metal, paper winders, coilers, and rewinders.

The type E element has also been employed on stationary Prony-brake-type applications. It is ideally suited to high-inertia starting-duty requirements and makes practical the use of a synchronous motor or normal-starting-torque induction motor of proper rating for the actual operating load, instead of the more expensive motor and control required by the high-inertia starting requirements if directly coupled to the drive.

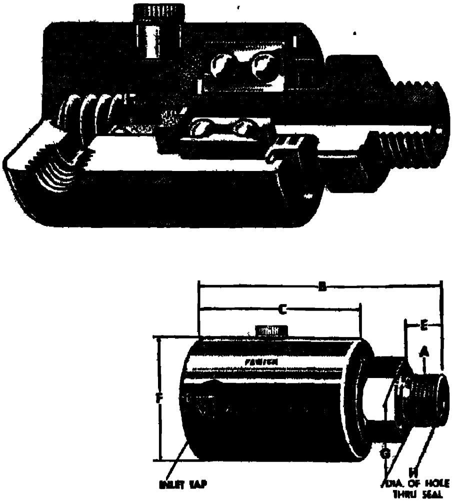


Fig. 14-103. ABC Rotorseal.

The Rotorseal

Size	Max speed, rpm	Max air pressure, psi	*A* th'd diam,	Dimensions, in.							
				B	*C*	*D*	*E*	*F*	*G*	*H*	Pipe tap
A	2,500	150	⅝-18 N.F.	3⅛	2	⅜	½	1½	⅞	¼	¼
B	2,000	150	⅝-18 N.F.	3 7/16	2¼	7/16	½	1⅝	⅞	⅜	¼
C	1,500	150	1-14 N.F.	4⅜	3	1 1/16	¾	2½	1⅜	⅝	½

This type is also ideally suited to cyclic braking requirements on equipment such as punch presses, shears, tile trimmers, and similar machines. On applications of this type, the motor can be started and allowed to come to full speed without load. The full running torque of the motor can then be utilized through controlled application of air pressure through a modulating valve to accelerate the driven equipment. Starting cycles as long as 3 min can be obtained. Where synchronous motors are preferred to obtain the benefits of power-factor correction, the use of this type of clutch permits continuous operation of the motor and cyclic or periodic use of the driven equipment.

Torque Ratings. The torque ratings shown under type E clutches are based on 75 psi inflation pressure in the tube. Correction for higher or lower operating pressures should be made through use of the formula shown under the type CB service rating graph. On continuous slip applications, the torque rating of the type E element is 40 per cent of the value shown in the torque rating column. This decreased rating is predicted on the use of a maximum of approximately 30 psi inflation pressure.

The type E clutch can be supplied with special high-heat-dissipating drums or with drums suitable for use with auxiliary water cooling. The type E clutch consists of two similar stamped-steel housings, one of which is provided with an opening to permit connection of the valve in the tube to the air supply. These housings are cadmium plated to prevent any damage from corrosion. The tube is an individual member supported by the steel housings. The friction-shoe assemblies consist of an asbestos-synthetic-resin-type friction block substantially secured to a cadmium-plated steel backing plate. This backing plate is mounted in an offset position on the friction block. This offset construction permits removal and installation of the friction-shoe assemblies without disturbing the housing mounting on those installations where provision has been made to permit drawing back the drum a sufficient distance to clear the friction shoes. The torque is transmitted from friction-shoe assemblies

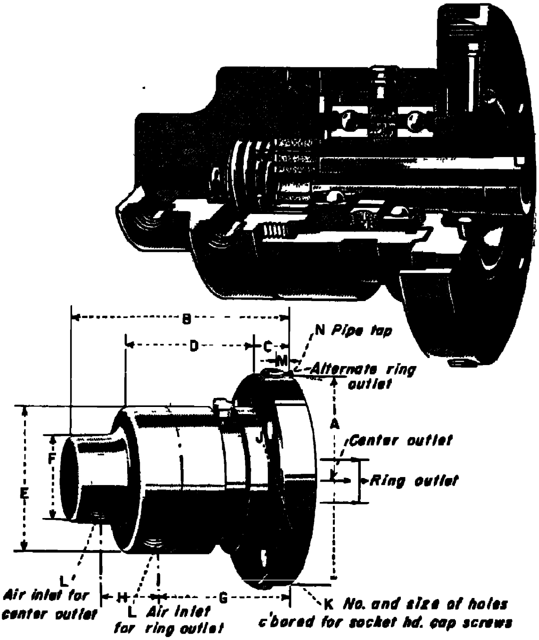

B
D
C
M
N Pipe tap
Alternate ring outlet
A
Center outlet
Ring outlet
E
F
J
L'
Air inlet for center outlet
H
G
L Air inlet for ring outlet
K No. and size of holes c'bored for socket hd. cap screws

FIG. 14-104. BD and FD Rotorseal.

Dual-passage Rotorseal

Size	Max speed, rpm	Max air pressure, psi	Dimensions, in.												
			A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	*K*	*L*	*M*	*N*
BD	1,500	150	$4\frac{1}{4}$	$4\frac{7}{8}$	$\frac{3}{4}$	$3\frac{1}{32}$	$2\frac{15}{16}$	$1\frac{3}{4}$	$3\frac{11}{32}$	$1\frac{1}{8}$	$3\frac{9}{16}$	$4\frac{3}{8}$	$\frac{1}{4}$ P. tap	$\frac{5}{8}$	$\frac{1}{4}$
CD	1,000	150	$5\frac{7}{8}$	7	$1\frac{3}{16}$	$5\frac{7}{16}$	$4\frac{7}{16}$	3	$4\frac{23}{32}$	$1\frac{5}{8}$	$5\frac{5}{16}$	$6\frac{3}{8}$	$\frac{1}{2}$ P. tap	$\frac{19}{32}$	$\frac{1}{2}$

through torque bars of substantial proportions to the steel housings. The friction-shoe assemblies are held in place in the housings and are retracted when air is exhausted by release springs inserted in a slot in the friction block and supported by lips on the housings. These release springs are of highest quality alloy spring-steel material and are cadmium plated to prevent corrosion. A series of release springs is available for each size of element. The choice of the correct release spring is based upon the maximum operating speed of the clutch (or brake) so that the necessary force to overcome the effect of centrifugal force will be available to ensure complete disengagement.

The type C element assembly is also applicable to either clutch or brake function. Its useful field of application consists of those requirements where heavy-duty acceleration or braking loads are imposed and where space considerations lead to a preference for a constricting type of unit. It is essentially applied on applications where the element may operate when disengaged at overrunning speeds higher than the speed at which the clutch normally operates. Typical of this type of application are cable-drum clutches where the free falling speed of the load on the cable may cause the drum to operate at speeds considerably higher than the speed obtained when the drum is driven by the prime mover. The construction of the type C element is similar to that of the type E, except that it is of the constricting type. The housing is a combination of cast Meehanite and steel-plate construction rather than steel stampings.

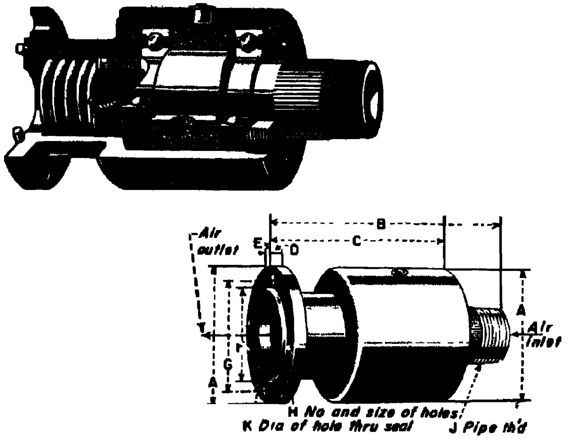


FIG. 14-105. RH Rotorseal.

Size, in.	Ass'y No.	P/L No.	Max speed, rpm	Air pressure, psi	Weight, lb	Dimensions, in.									
						A	B	C	D	E	F	G	H	J	K
¾	12061	5487	2500	150	4	3	5 15/16	4 5/16	⅜	⅛	1.749/1 747	2⅜	4 11/32	¾	¾
1	12050	5488	2250	150	4½	3¼	6½	5	⅜	⅛	1.999/1.997	2⅝	4 15/32	1	1
1¼	12060	5489	1750	150	6	3¾	7 5/16	5½	⅜	⅛	2.499/2.497	3⅛	4 13/32	1¼	1¼
1½	10941	5461	1500	150	7	4	7 23/32	5⅝	½	⅛	2.749/2 747	3⅛	4 19/32	1½	1½

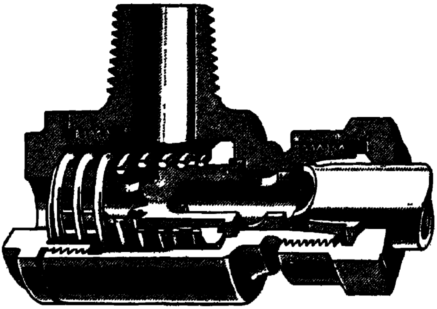
The torque ratings for type C elements are based on 75 psi inflation pressure in the tube. Correction for higher or lower operating pressures should be made through use of the formula shown under the type CB service-rating graph.

THE ROTORSEAL

The Rotorseal (Fig. 14-103) is a positive seal for introducing air, liquids, or gases, under pressure, into a rotating shaft. It can also be used for vacuum applications.

The Rotorseal operates continuously or intermittently at high speeds in either direction.

As shown in Fig. 14-103, sealing is effected by a nonmetallic bushing held by spring pressure against a spindle face. Sealing surfaces are given the finest precision lapping obtainable. The spindle rotates on a sealed double-row bearing.



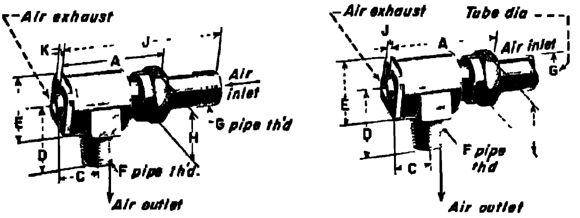


FIG. 14-106. Quick-release valve.

With tube connection

Size, quick release valve, in.	Ass'y No.	P/L No.	Dimensions, in.								
			A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*
3/8	10785	5406	2 5/16	2	7/8	1 3/16	1 1/8	3/8	1/2	7/8	1/16
1/2	9112	5407	2 3/4	2 3/8	1	1 7/16	1 7/16	1/2	5/8	1 1/8	3/16
3/4	10532	5141	3 1/16	2 5/8	1 1/8	1 9/16	1 7/16	3/4	3/4	1 1/4	3/16
1	11145	5413	3 1/2	3 1/16	1 5/16	1 15/16	2	1	1	1 1/2	1/4

With pipe connection

Size, quick release valve, in.	Ass'y No.	P/L No.	Dimensions, in.									
			A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	*K*
3/8	11671	5468	2 5/16	2	7/8	1 3/16	1 1/8	3/8	1/4	7/8	3 3/4	1/16
1/2	11672	5469	2 3/4	2 3/8	1	1 7/16	1 7/16	1/2	1/2	1 1/8	4 3/8	3/16
3/4	11673	5470	3 1/16	2 5/8	1 1/8	1 9/16	1 7/16	3/4	3/4	1 1/4	4 3/4	3/16
1	11674	5471	3 1/2	3 1/16	1 5/16	1 15/16	2	1	1	1 1/2	5 1/2	1/4

The Rotorseal is connected to the shaft simply by screwing the male end of the seal into a properly aligned, drilled, and tapped mating hole in the shaft end. The connection from Rotorseal to stationary piping should be made through a flexible hose connection. This prevents artificial loading of the Rotorseal bearings. Either rubber or flexible metallic hose connections can be supplied.

Two rotating elements on one shaft can be separately supplied and controlled from one shaft end with the dual-passage Rotorseal BD (Fig. 14-104). Where both shaft ends are available, four separate clutches can be individually actuated with two dual-passage rotorseals. Each line of the seal can be used to conduct a different medium (liquids, gases, etc.).

This seal is flange-mounted on any size of shaft by means of adapters designed for every type of application. Mounting components are recommended and available to suit the particular requirements of the job.

Flexible hose connections should be used between stationary piping and the dual-passage Rotorseal. This prevents artificial loading of the bearings. Either rubber or flexible metallic hose connections can be supplied.

Type RH Rotorseal (Fig. 14-105) is a single-passage seal for applications where large volumes of air, gases, or liquids are required. The operating principle of the type RH is identical with that of the smaller Rotorseal units.

It is flange mounted to a shaft end. The housing rotates, and the inlet is connected to stationary piping through a flexible hose connector.

The weight of type RH Rotorseals is very small, eliminating the need for auxiliary support arrangements. Decided space economy is also accomplished.

For use with other mediums than air, the manufacturer should be consulted for installation recommendations.

The quick-release valve (Fig. 14-106) provides a positive and efficient method of quickly discharging air under pressure from clutch or brake units, air cylinders, or any system where it is necessary quickly to evacuate air under pressure. Its operation is quick and positive, regardless of its location relative to the operating or control valve.

The valve operates on the principle of pressure differential. Normal discharge occurs through the control-valve discharge port until a 5-lb differential is developed in the quick-release valve. This actuates the discharge plunger, opening the exhaust port and exhausting air directly to atmosphere.

The unit is self-adjusting and requires no lubrication.

The standard quick-release valve is supplied with a male pipe-thread outlet connection, and a tube connection fitting on the inlet side. It is also available with an adapter for pipe-to-tube connection on the inlet side.

TORQUE CONTROL, TENSION CONTROL, AND TORQUE AMPLIFICATION

BY JAMES P. MADDEN, M.E.

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Torque control has always been a major problem in industrial-plant operations. Much has been done in the design of electric motors to solve this, but accomplishments have been in the direction of providing sufficient torque in the form of "minimum guaranteed" torque rather than accurate maximum torque.

It is of paramount importance that mechanical power drives have sufficient torque capacity available at all times to start equipment and keep it running under all reasonable load conditions, but excessive torque is frequently *undesirable and sometimes* the cause of serious breakdowns.

Torque control and tension control are synonymous in so far as the problem of the plant engineer is concerned because torque is the product of the pull (or force) multiplied by the radius at which the pull is exerted, whereas tension is the conventional term used to designate the pull exerted on a piece of material, usually in linear motion, without reference to the radius of the core or coil on which the material is being wound or from which it is being unwound. To produce any desired tension, it is necessary to maintain a torque or torques equal to the desired tension multiplied by the radius or radii at which the material is being wound or unwound.

The difference, therefore, between torque control and tension control lies in the type of apparatus used to accomplish the control.

TORQUE CONTROL

In designing or selecting a mechanical power drive with torque control for any piece of mechanical equipment, it is necessary to have the following information:

1. The torque necessary to start the equipment under the most adverse starting conditions. This is called "the starting torque."
2. The torque necessary to keep the piece of equipment running during all reasonable operating load conditions. This is called the "running torque."
3. The maximum operating torque which it is desired to impose on the equipment or motor. This is called "maximum overload torque" and is largely a matter of judgment by the plant engineer.

Inasmuch as the majority of engineers refer to starting torque, running torque, and overload torque in terms of horsepower, this practice will be followed as far as possible in this subsection.

In many instances the "running torque" will be the determining guiding factor in selecting the motor and the chain, belt, gears, or gear reducer which make up the drive. These are conventional drives and present no problem in torque control.

In some mechanical power drives, owing to the weight or mass of the driven equipment or the product which is being handled in the equipment, it takes considerably more horsepower to start than it does to run it. To meet this condition, a larger motor may be used or a motor with high-starting-torque characteristics. The first choice is objectionable because of increased motor cost and lower efficiency and power factor. The second choice is objectionable because high-torque motors develop a starting torque of from 240 to 300 per cent. This high torque is exerted instantly upon closing of the motor circuit, with the result that such torque is imposed on the component parts of the drive and the driven equipment under practically shock conditions. With machinery that is subject to frequent starting and stopping, the application of this high torque often results in damage to the drive, the machine, or the product being handled.

This condition is very similar to that which would prevail if a long heavy freight train were provided with an enormous locomotive which was capable of starting the train instantly and accelerating up to full speed in a few seconds. The couplers would undoubtedly be jerked out of the leading cars every time the locomotive started.

Starting shock can be materially alleviated by the use of wound-rotor motors with suitable controls which provide slow start and gradual acceleration. This method, however, is expensive and tends to complicate the drive.

The second item, "running torque" (or horsepower), is that which should determine the size of motor for maximum efficiency of operation.

The third item, "maximum overload torque" (or maximum horsepower), is of interest as a safety factor. The parts of all machinery and motors are designed for some definite fiber stress, and it inevitably follows that, if the maximum stresses for which the machinery is designed are exceeded, breakage will result, especially if the maximum load occurs suddenly, thus creating a shock condition.

An ideal arrangement would be where a motor would build up gradually to the horsepower capacity necessary to start a machine, and having done so, would accelerate the machine to full speed at a rate that would not impose more than the safe fiber stress on any of the parts, as determined by the designing engineer. This is what some engineers have called a steam-engine start. This ideal setup can be very closely approximated by the use of a piece of mechanical power equipment known as a slip-ring starter.

This device, which is of mechanical construction, is a modified form of centrifugal friction clutch whose torque capacity varies directly as the square of the speed of operation, and whose horsepower capacity varies as the cube of the operating speed. For any given horsepower capacity, therefore, the higher the speed, the smaller will be the size of this device.

Practically every clutching device has a slipping or releasing point which is variable because it is directly dependent on the coefficient of friction of the friction material used. As the coefficient of friction of all friction linings varies widely at any and all times, friction devices as such are not sufficiently accurate or constant to be used as safety devices. The slip-ring starter, however, possesses a novel characteristic in that, although the coefficient of friction will vary as much as 50 per cent, its torque and horsepower capacities do not change substantially. This seemingly paradoxical situation is explained in the following mathematical treatment of the subject.

Theory of the Slip-ring Starter. There are three elements in the slip-ring starter: the spider, the friction band, and the drum. (In actual construction there are two friction bands, but this is merely to provide a balanced drive; theoretically, one band is sufficient.) The spider is keyed on the motor shaft and the drum on the shaft of the driven machine; the band is weighted and lies inside the drum, one end engaged by a lug projecting from the spider, by which the band is pulled around. The rear or trailing end of the band is free. As the motor speeds up, centrifugal force causes the band to expand and exert a frictional drag on the drum, the torque thus transmitted depending upon the motor speed, diameter of the drum, and weight of the friction band.

Torque transmitted by the slip-ring starter is practically independent of coefficient of friction. Such a characteristic completely differentiates this from ordinary friction devices and results in a constancy which is novel and apparently at variance with previous engineering experience. Since the operation of the slip-ring starter depends upon a combination of wrapping friction and centrifugal force, both of which involve principles difficult to visualize, explanation of the constancy of transmitted torque cannot, unfortunately, be given in simple words but must be expressed in the form of a mathematical proof.

Figure 14-107 shows a drum which may be considered stationary and within which an arm revolves, in the direction indicated by the arrow. At its extremity this arm is connected to one end of the flexible weighted friction band. Now, let the centrifugal force per unit length of the friction band = c. (This will depend upon speed, weight of band, radius of drum r, etc., all of which are considered constant.) Taking the elementary portion of band da, the radial force due to centrifugal effect will be $cr\,da$, part of which, however, is neutralized by the tension in the band. In Fig. 14-108, let these tension forces be f and $f + df$. Then the radial resultant will be $f \sin (da/a) + (f - df) \sin (da/a)$, or simply $f\,da$.

The residual of radial force P pressing the band against the drum will then be $cr\,da - f\,da = P = (cr - F)\,da$.

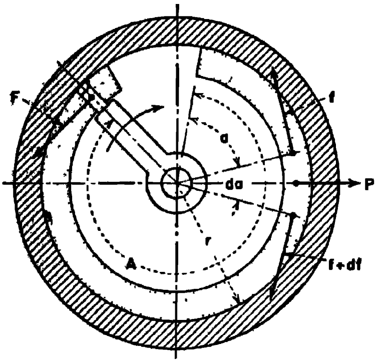


FIG. 14-107.

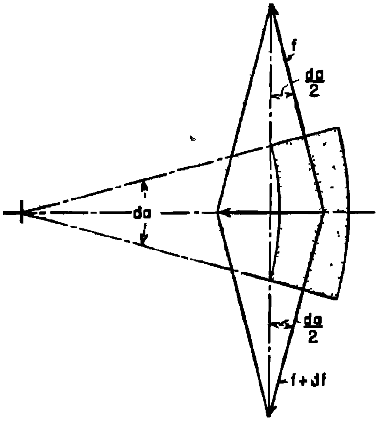


FIG. 14-108.

The tangential force due to friction will be $P\mu$, where μ = coefficient of friction. But $P\mu = df$; then $df = (cr - f)\mu\,da$.

Integrating this and noting that when $a = 0, f = 0$, we get

$$\log [cr/(cr - f)] = \mu a \qquad \text{or} \qquad f = cr\,[1 - (1/e^{\mu a})]$$

(e is the logarithmic base 2.71828).

The torque T on the arm will be rF.

$$T = rF = cr^2\,[1 - (1/e^{\mu A})]$$

where F and A are the values of f and a corresponding to the point of attachment of the arm.

This formula shows that a wide change in μ causes only a slight change in the torque transmitted

To take extreme values of μ, assuming $A = 6$ radians, where

$$\mu = 0.35 \qquad T = 0.875cr^2$$
$$\mu = \text{infinity} \qquad T = 1.000cr^2$$

Thus it is obvious that the coefficient of friction, while necessarily a factor in the formula, here appears in such a manner as to render it of very little importance in the final answer.

Figure 14-109 shows a curve of the horsepower transmitted by a slip-ring starter operating on the foregoing principle at different values of the coefficient of friction. The full line portion of the curve, from coefficient 0.35 up, covers the working range of the device. While this range may seem extreme, as a matter of fact any friction material may be expected to vary as much as this under normal operating conditions for short periods, owing to effects of heating, wear, dampness, glaze, etc.

The purpose of the slip-ring starter now becomes apparent. The coefficient of friction cannot be stabilized or controlled, but as shown mathematically and illustrated by the starter curve, these variations have practically no effect on the final result.

The example shown is that of a unit designed to transmit power of 1.15 hp. At 0.35 coefficient it transmits 1.1 hp. As the coefficient of its friction bands rises, the power transmitted is shown in the curve as approaching but never reaching the line of 1.25 hp.

The curve Fig. 14-109 explains the action of the starter in providing against injurious overload.

Application Characteristics. The following application characteristics are obtainable with a slip-ring starter:

1. Load-speed acceleration can be varied from a few seconds to several minutes.
2. Motor, driven machine, and transmission equipment are relieved from injurious shock and strain.
3. Simplest electrical starting equipment can be used.
4. The motor starts under no load and is free of load until it reaches almost full speed.

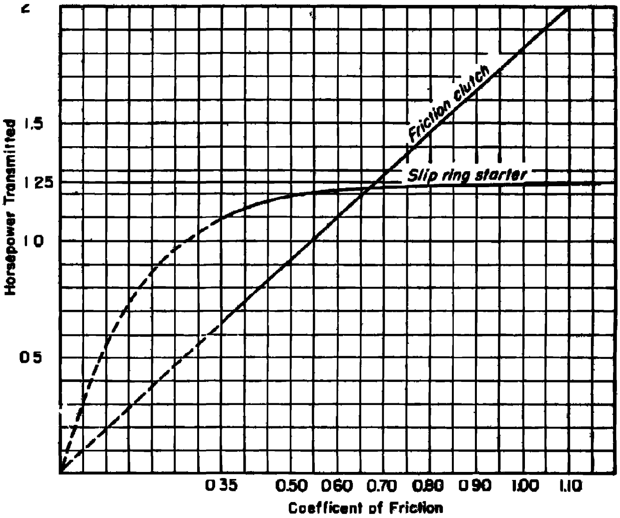


FIG. 14-109. Chart showing relation of horsepower to coefficient of friction.

5. The device is inherently an overload device. It slips as soon as the load reaches its capacity but continues to exert its torque and resumes positive drive as soon as the load returns to the predetermined limit.
6. The capacity to transmit torque is the same whether the driven machine is operating at full speed or is in a stalled condition.
7. Any desired overload factor can be obtained.
8. The entire unit consists of a small number of separable parts which can be disassembled and reassembled by any mechanic.
9. When acting as a coupling between the motor and the driven machine, the starter, in addition to its other characteristics, serves as a flexible coupling.
10. It is applicable to any constant-speed prime mover and in certain cases to variable-speed prime movers for horsepower capacity ranging from fractional to 1,500 hp.
11. The friction lining is the only wearing part. It usually lasts for from 4 to 10 years and can be cheaply and quickly renewed.
12. Slippage occurs only in starting and in cases of overload.
13. Accuracy of performance is maintained throughout the life of the unit.
14. The device can be incorporated as an integral unit with flat-belt pulley, chain sprocket, V-belt sheave, or gear pinion.

Figure 14-110 is a chart which shows a typical situation in which a motor is equipped with a slip-ring starter with a capacity of 125 per cent of the motor rating. The motor starts under no load, and as it accelerates to full speed, the torque and horsepower capacity of the slip-ring starter develop very gradually at first, which serves to take up the slack in chains, belts, or gears of the drive before applying the full torque.

The "driven machine" starts to turn over as soon as the slip-ring starter has sufficient torque capacity to overcome the frictional or other load. However, the driven machine starts slowly, and most of its acceleration takes place after the motor has reached full speed. The motor is under the load capacity of the slip-ring starter until the driven machine reaches full speed, and then it drops off to the running load.

In Fig. 14-111 are shown several motor-speed torque curves of commonly used electric motors. The curves numbered 5 to 13 are the torque capacity of different slip-ring starters which range in capacity from 103 to 270 per cent of the full-load torque of the motor.

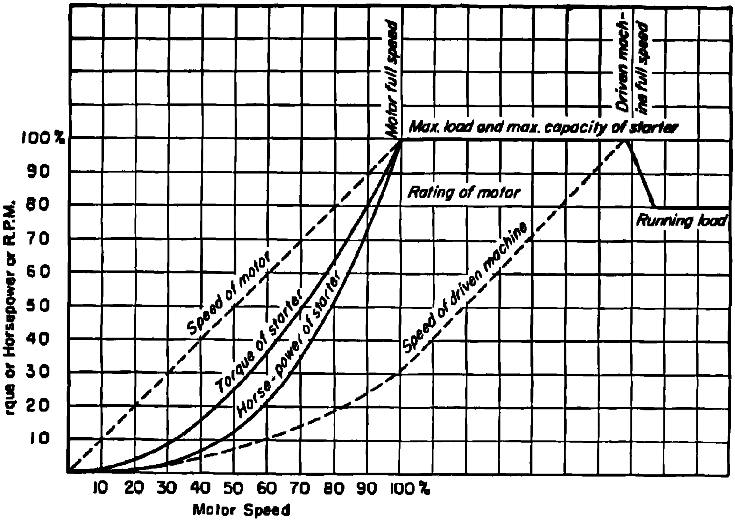


FIG. 14-110. Chart showing smooth gradual start and acceleration of driven machine and no-load start of motor.

The nature of the starting conditions or predetermined overload release of the machine to be driven will determine the selection of the correct size of slip-ring starter for the job.

From Fig. 14-111 it will be seen readily that the general-purpose or normal-starting-torque motor is best adapted for starting against a very high starting-torque load caused either by inertia or breakaway conditions. The general-purpose motor, because of its extremely high pull-out torque, provides the highest possible torque for starting when it is equipped with a slip-ring starter.

It will also be seen that the high-amperage starting current lasts only while the motor itself is getting up to speed. When the maximum load of the slip-ring-starter capacity comes on, the current has dropped to very reasonable proportions. In practically every case, therefore, the motor can be started on full voltage.

Having established the fact that accurate torque control can be obtained, it is now possible to approach the problem of providing an economical and efficient motor drive with the following salient facts in mind:

Any normal-starting-torque or general-purpose squirrel-cage motor which will run the driven machine will also start it. The horsepower rating of the motor will be determined by the running load on the driven machine. The motor can be started on full voltage provided there are no local conditions that make this prohibitive. In this connection it should be mentioned that, whereas in extreme cases it may not be permissible to start under full voltage with a solid coupling on the motor because of the long-drawn-out starting current, it is frequently permissible to do so on account of the very short period of inrush current which prevails when the motor is started under no load.

As a guide in determining the correct capacity of slip-ring starter which should be used for driving various machines and equipment, Table 14-102 shows the relation between the motor rating and that of the slip-ring starter. These figures are based on actual industrial operating experience.

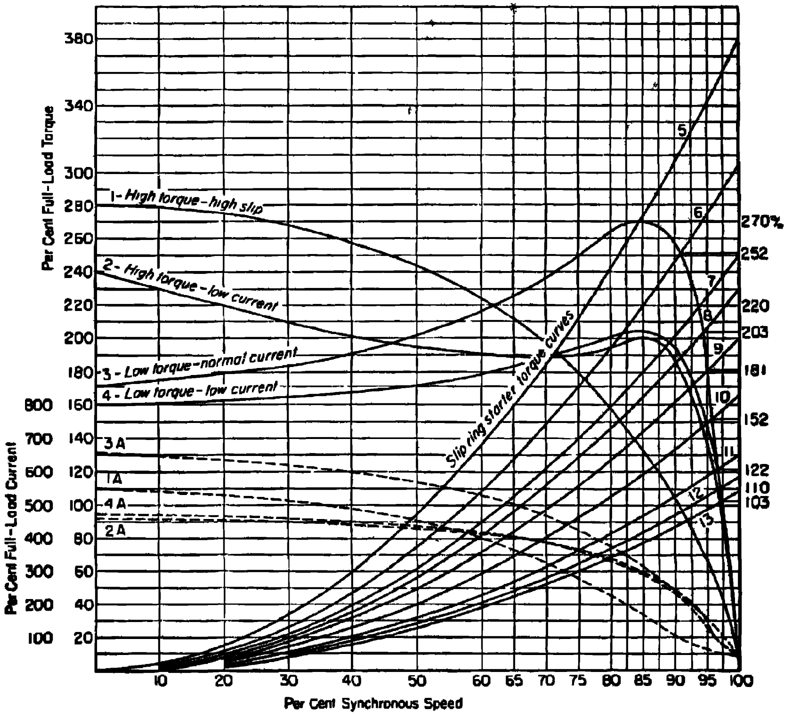


Fig. 14-111. Chart showing several capacities of slip-ring starters which can be applied to motors, to suit various load conditions.

After the capacity of the slip-ring starter is decided upon, either a V-belt, flat-belt, chain, gear, or direct-connected drive may be used. The size of the slip-ring starter can be selected from Table 14-103, and the dimensions can be obtained from Table 14-104.

A large number of sizes of slip-ring starter other than those listed are available, but those given in Tables 14-103 and 14-104 will be sufficient to guide the plant engineer in making his preliminary arrangements.

The majority of mechanical power drives run continuously in one direction, but where operation in both directions of rotation is required, slip-ring starters correspondingly are made reversible, and the same accuracy of torque control prevails in both directions of rotation. This is provided for in the construction of the driving bands.

Table 14-102. Relation of Motor and Slip-ring Starter Rating

Driven machine	Ratio of slip-ring-starter rating to motor rating, per cent	Reasons for using slip-ring starter
Dough mixers, chance coal cleaners, vacuum pans, pumps handling high-viscosity materials, machines subject to short-period high-peak loads.	250:270	Extremely hard-starting breakaway load. Low-starting-torque motor can be used which reaches full speed to exert its maximum "pull-out" torque before it encounters any material load.
Chaser mills, coal shakers, hammer mills, pulverizers, multiple-plunger pumps, jute spinners, heavy textile equipment, cardboard machines, chance cones, cards, etc.	200:225	Hard-starting heavy equipment with occasional high-peak loads.
Conveyors, fans, centrifuges, air compressors, lead pumps, refrigerator compressors, pickers, bridge-cable wire wrappers, rovers, shears, punches, Bradford breakers, barking drums, shakers, air pumps, blowers, ball mills, chemical equipment, turbine drives, coal breakers.	150:200	General run of industrial equipment, fairly hard start with maximum load limitation.
Silk winders, spinners, nylon equipment, cable-taping machines, light textile equipment, gasoline-engine drives, crockery and china conveyors, light machinery with very slow start.	100:150	Light slow-starting equipment with load limitation to low percentages of overload.

Table 14-103. Maximum Horsepower Capacities of Slip-ring Starters

Size of starter, in.	Maximum horsepower capacity									
	375 rpm	425 rpm	490 rpm	570 rpm	680 rpm	710 rpm	860 rpm	1,160 rpm	1,450 rpm	1,750 rpm
5½ × 2½	....	....	....	0.4	0.63	0.7	1 3	3	5	7 5
6 × 3	....	.. .	...	0.7	1.15	1 3	2.3	5.5	10	16 5
6½ × 3	....	....	...	0 85	1.4	1 6	3	7	11.3	20
7 × 3	....	....	0.8	1.2	2	2.2	4	9.5	15.3	24
8 × 3	....	...	1.3	2	3.4	4	7	16	26	41 5
9 × 3	1.1	1.5	2.3	3.7	5.7	6.5	11.5	24	40	60
10 × 4	2	2.8	4.3	6.7	11.5	13	23	56	75	110
11 × 5	4	5.5	8.6	13	23	26	46	77	150	
12 × 5	5	7.5	11	17	29	33	58	100	190	
12 × 6	6.5	9	14	22	37	43	75	125	240	
13 × 5	8	12	17	27	45	52	92	145		
13 × 6	9.5	13	20	33	54	62	110	165		
14 × 6	11.5	15 5	25	40	65	75	132	198		
15 × 6	15.5	21	33	52	88	100	175	270		
16 × 6	20	30	43	69	115	131	200	310		

The coupling type of slip-ring starter, shown in Fig. 14-112, consists of a driving spider *A*, driven drum *B*, and driving bands *C*. The direction of rotation is indicated by the arrow.

The reversible-coupling type of slip-ring starter for operation in both directions of rotation is illustrated by Fig. 14-113. It will be noted that the driving bands are equipped with driving cleats at both ends and that the spider has two driving arms for each band.

The torque effort is transmitted from the driven spider to the bands through flexible rubber cushions. In the case of the coupling type, these rubber cushions, by their flexing, will compensate for angular misalignment and eccentricity. However, in erection, a definite effort should be made to obtain accurate alignment.

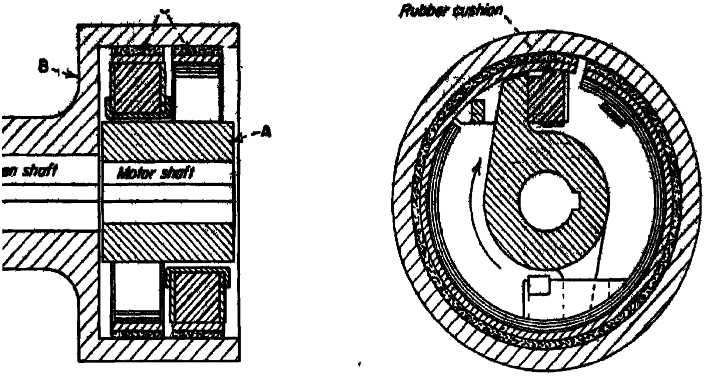
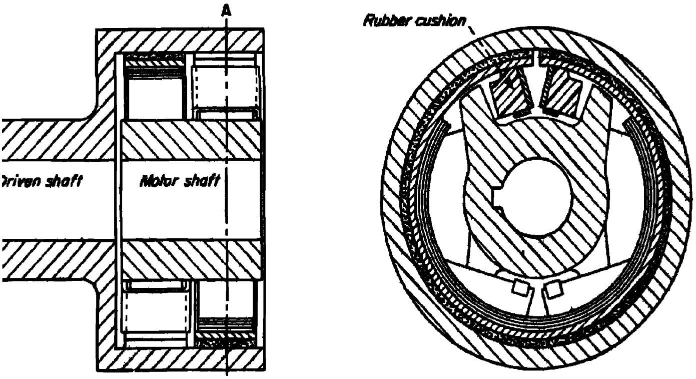
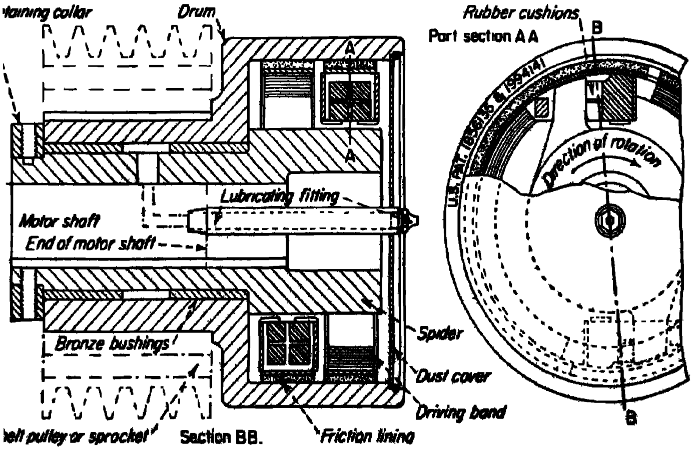


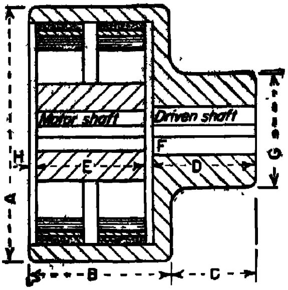
Fig. 14-112. Coupling type—one direction.



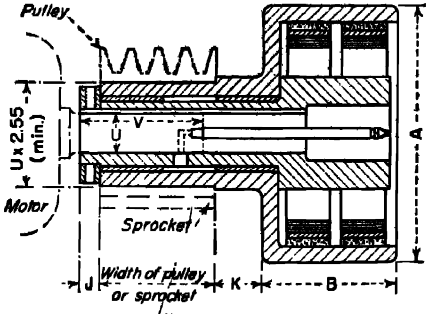
Section AA.

Fig 14-113. Coupling type—reversible.





Coupling type



Pulley or sprocket type

Table 14-104

Size of starter	Dimensions, in.										Wt., lb	Max bore
	A	B	C	D	E	F	G	H	J	K		
5½ × 2½	6¼	3⅝	1½	1⅞	2⅞	3/16	2¼	3/16	⅝	1	30	1½
6 × 3	7	4¼	2½	3	3¼	¼	3	¼	⅝	1¼	35	1⅝
6½ × 3	7½	5	2¾	3¼	3⅝	¼	3¼	⅝	¾	1¼	45	1¾
7 × 3	8	5⅛	2⅞	3½	3¾	¼	3½	½	¾	1¼	60	1¾
8 × 3	9¼	5⅛	3⅜	4⅛	3¾	¼	4	⅜	1	1½	90	2¼
9 × 3	10¼	5¼	3½	4¼	4	¼	4	¼	1	1½	105	2¼
10 × 4	11¼	6¼	3¾	4½	5	¼	5½	¼	1¼	2	145	2¾
11 × 5	12½	6⅞	3¾	4½	5⅝	¼	6½	¼	1¼	2	195	2¾
12 × 5	13½	7¼	3¾	4⅝	5⅝	¼	7	½	1¼	2	235	3½
12 × 6	13½	7¼	3¾	4⅝	5⅝	¼	7	½	1¼	2	245	3½
13 × 5	14½	8¼	3¾	4⅝	6⅝	¼	7½	½	1¼	2	285	3¾
13 × 6	14½	8¼	4⅛	5	6½	¼	7½	⅝	1¼	2	300	3¾
14 × 6	15¾	8¼	4⅜	5¼	6½	¼	8	⅝	1¼	2	335	4
15 × 6	16¾	8¼	4⅜	5¼	6½	¼	8	⅝	1¼	2	365	4
16 × 6	17¾	8¼	4⅜	5¼	6⅝	¼	8	½	1¼	2	410	4

The pulley and sprocket type of slip-ring starter is shown in Fig. 14-114. This type is provided with a bearing of renewable bronze bushings in which the driving spider rotates during slipping periods. A flat-belt pulley, V-belt sheave, sprocket, or gear can be mounted on the hub of the drum (see Figs. 14-115, 14-116, 14-117). A coupling-type starter is shown by Fig. 14-118.

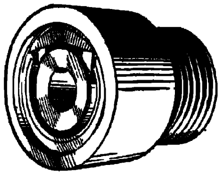


Fig. 14-115. Slip-ring starter arranged for V-belt drive.

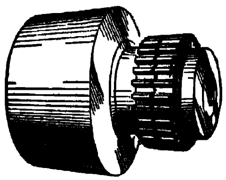


Fig. 14-116. Slip-ring starter arranged for silent chain or gear drive.

The pulley, sprocket, or gear is located in the same position in relation to the motor shaft as it would be if it was mounted directly on the motor shaft. If the diameter of the pulley or sprocket is small, instead of making them as separate pieces, they can be made as integral parts of the slip-ring starter by cutting the grooves or teeth on the hub of the drum.

The bearing functions for short periods and very infrequently. Therefore very little lubrication is needed. Radial holes in the motor shaft and driving spider convey the lubricant into the bearing reservoir from the lubricating fitting which is screwed into the end of the motor shaft.

Normally one end of the drum is left open, but when deleterious atmospheric conditions exist, a "cover" is used to protect the bands and drum surface. This is illustrated in Fig. 14-114.

Applications Where Torque Control Is Advantageous. It is not possible to give a complete list of equipment in the driving of which torque control is desirable or necessary. The majority of machinery presents no problem of this character, but there are a certain number of drives where high starting torque causes so much trouble that the attention of plant engineer is demanded continuously.

These drives include ball mills, grinders, pulverizers, hammer mills, dough mixers, compressors, heavy fans, cards, conveyors, centrifuges, drawbenches, rotary cutters, etc.

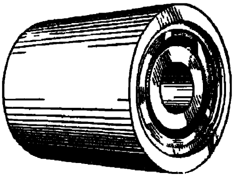


FIG. 14-117. Slip-ring starter arranged for flat-belt drive.

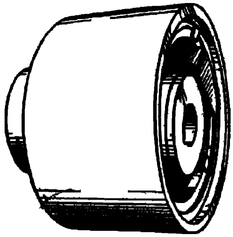


FIG. 14-118. Slip-ring starter, coupling type, for direct drive.

The high starting torque is caused by the inertia of the large masses which have to be set in motion. If the drive is of the positive type with no provision for slippage, motion of the driven machine begins instantly, and inasmuch as the motor acceleration is very rapid, the acceleration of the driven machine will be correspondingly so. This produces very high fiber stresses in every part of the drive causing either early fracture or fatigue which eventually results in breakdown.

Practically all the machines mentioned above can be started and driven satisfactorily and with high efficiency by general-purpose motors in combination with torque control. The maximum torque imposed may be as low as half of that which would normally be required. In addition, the elimination of the shock conditions is practically a positive guarantee against fractures or damage. The maximum operation load will determine the maximum horsepower capacity of the slip-ring starter.

Another high-starting-torque condition is found on machines in which the inertia factor is low, but where there is a breakaway load resistance which will exist for periods of time varying in length from a few seconds to several minutes. These machines are dough mixers, vacuum pans, pumps handling high-viscosity liquids, chance coal cleaners, and several types of chemical equipment. The high starting torque is caused by the fact that the machines contain materials in a state of rest which must be agitated or moved bodily by an agitator of one form or another. A slip-ring starter with a torque capacity equal to the pull-out torque of the motor, which will be from 250 to 270 per cent of full-load torque, will start these machines smoothly and safely. If torque control is not used, it will be necessary to use a wound-rotor motor with the necessary electrical control to produce low speed with high torque, under which condition it is necessary that the operator exercise care and good judgment, or he is likely to damage the motor and the machine.

For the benefit of the engineer who is considering the problem of providing accurate torque limitation, the question of its application to several machines will be

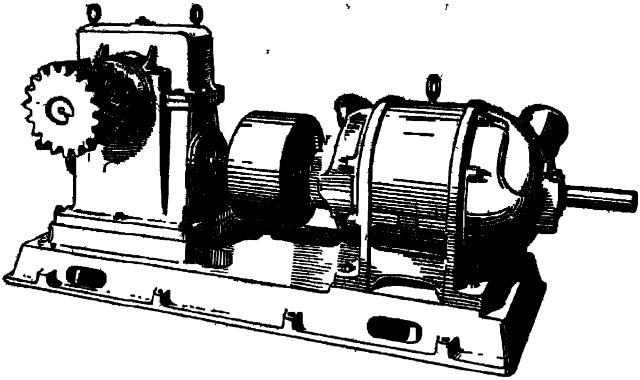


Fig. 14-119. Typical motor direct-connected drive.

discussed. Each machine is typical of a group requiring a specific percentage of overload horsepower capacity in relation to the full-load rating of the motor.

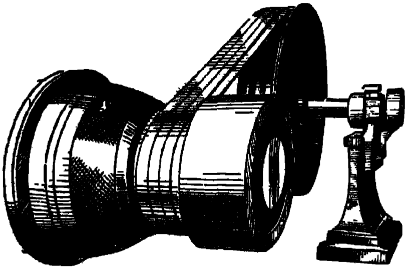


Fig. 14-120. Gill-box drive with pulley-type slip-ring starter.

1. A batch mixer, vacuum pan, cone coal cleaner, or agitator in which the impeller or agitator is immersed in a more or less solid material. To start any one of these machines requires a very high torque in order to move the impeller or agitator. It is assumed that the machine will be driven by a 50-hp 1,200-rpm motor. The correct size of slip-ring starter should have a maximum torque capacity of 270 per cent of motor full-load rating. Under this torque load a low-torque motor will operate at a speed of 1,020 rpm (approximately). The starter will therefore have a capacity of 119 hp at 1,020 rpm. The maximum running (pull-out) torque is used for starting, and it is applied gradually and without shock, and therefore there is no danger of shearing keys or twisting shafts. The starter will at no time allow the load on the motor or machine to exceed 119 hp.

2. Hammer mill, subject to plugging or occasional overfeeding. The mill is driven by a 150-hp 720-rpm motor. The normal running load is 145 to 155 hp. As a matter of judgment, it is desired that the maximum load should not exceed 210 per

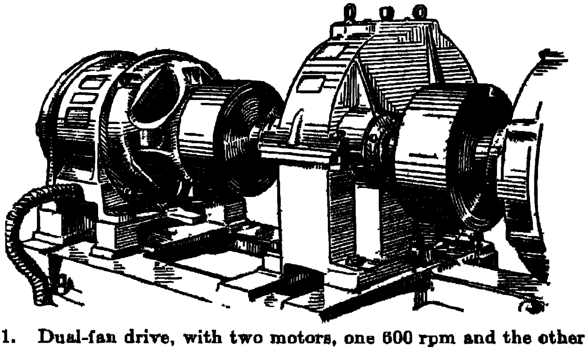


Fig. 14-121. Dual-fan drive, with two motors, one 600 rpm and the other 900 rpm.

cent of full-load rating. A 150-hp, low-torque motor will operate at 600 rpm under this amount of overload. The slip-ring starter therefore will have a maximum capacity of 300 hp. If the load goes beyond 300 hp, the starter will slip instantly.

3. Ball mill, conveyor, elevator, or air compressor. It is customary to regard these machines as requiring 200 per cent torque to start them with conventional drives.

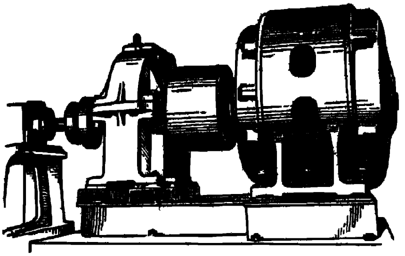


Fig. 14-122. Synchronous motor fan drive through reducer.

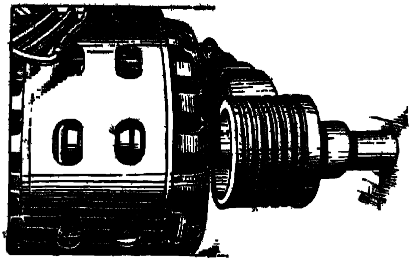
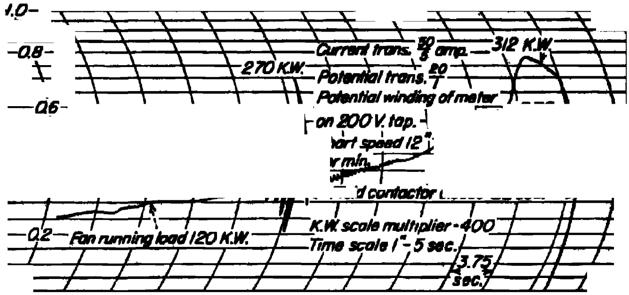
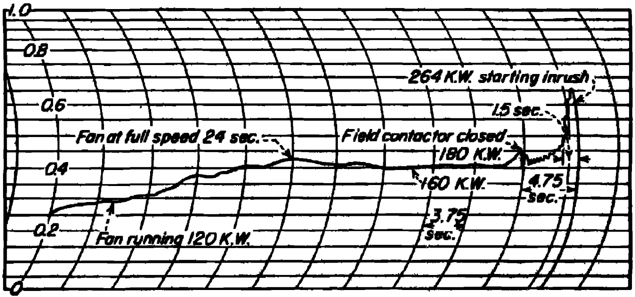


Fig. 14-123. Large pulverizer motor drive.



Curve with motor coupled to fan through flexible coupling



Curve with motor coupled to fan through slip ring starter

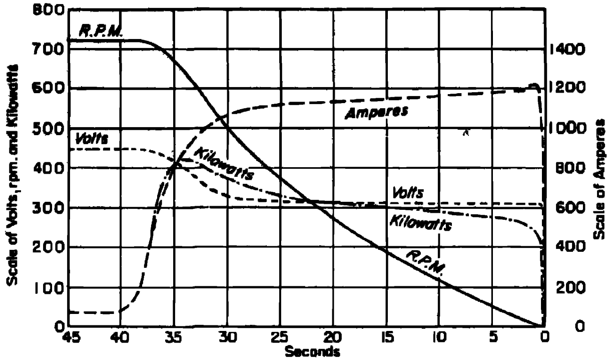
Fig. 14-124. Charts showing comparison between fan drive with flexible coupling and slip-ring starter.

This high torque is the result of inertia in the moving parts or the materials that have to be quickly set in motion. Assume a 50-hp 900-rpm low-torque motor is used for the drive, the correct size of slip-ring starter will be 75 hp maximum capacity at 840 rpm.

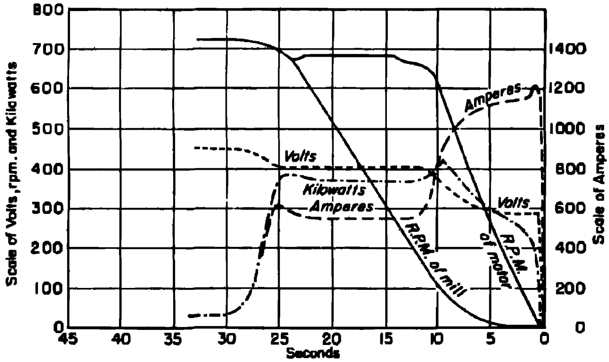
4. Large fan driven by a 200-hp 600-rpm motor. In this case the starting torque is caused entirely by inertia, and the time lapse to reach full speed is usually in the neighborhood of 30 sec. A slip-ring starter of 270 hp at 565 rpm will provide ample

torque to accelerate the fan against the air load which develops as the fan increases in speed and approaches full speed.

5. A conveyor that transports articles in process of manufacture must be started very slowly to prevent a jerky start which would disturb or upset the articles. The running load should be known with a fair amount of accuracy. If the conveyor itself is not too light in weight, a slip-ring starter with about 20 per cent capacity more than the running load will give a satisfactorily slow start. If a very slow start is desired, additional weight can be built into the driven side of the slip-ring starter to create a predetermined amount of inertia which will slow the acceleration down to any desired number of seconds or even minutes.



Motor coupled to pulverizer through flexible coupling



Motor coupled to pulverizer through slip ring starter

Fig. 14-125. Charts showing comparison between pulverizer drive with flexible coupling and slip-ring starter.

An example of this type of drive is in textile winding machines which are very light in weight. The product is also very light. These machines would normally start and reach full speed in about 2 or 3 sec, but the introduction of an additional amount of inertia will slow the acceleration down to 35 to 50 sec. This is very necessary in operations where light webbings or yarns are being transferred from one roll to another, and a quick start would be almost sure to break the material. In these light-power drives, the running load and not the motor rating should be used as the determining factor in arriving at the correct amount of torque limitation to be provided by the slip-ring starter.

An interesting characteristic of the slip-ring starter is that synchronous motors with low "starting torque" and low "pull-in torque" may be used in high-starting-torque drives in which synchronous motors would normally be inoperable. The slip-ring starter allows the synchronous motor to start easily and "pull in" to synchronous speed without any difficulty (see Figs. 14-122 and 14-123).

Figures 14-119, 14-120, 14-121, 14-122, and 14-123 show typical applications of slip-ring starters, and Figs. 14-124 and 14-125 are charts which show the starting conditions before and after installation of the slip-ring starter. The improvements which resulted are obvious.

LIMIT TORQUE COUPLING

The limit torque coupling is intended for use as an addition to the ordinary flexible coupling. It functions as a positive drive until the predetermined torque limit for which it is set is reached. At this torque load it slips but continues to exert the predetermined torque.

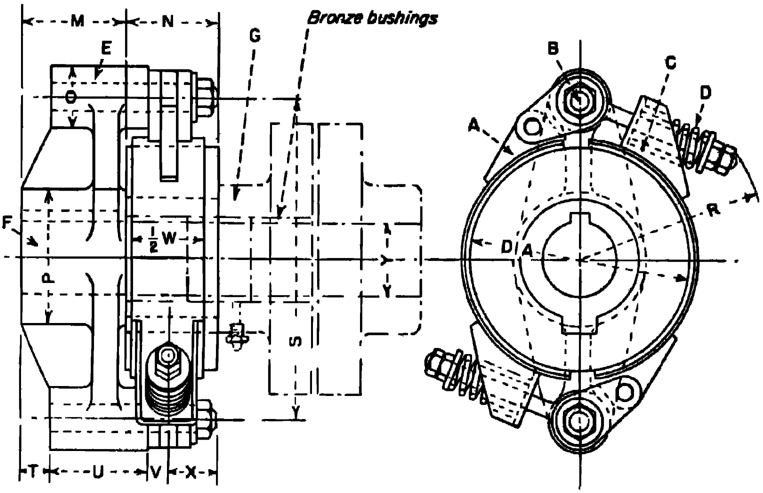


Fig. 14-126. Limit torque coupling.

In principle it is exactly the same as the slip-ring starter except that the pressure on the friction lining is provided by helical steel springs instead of centrifugal force.

It is particularly suitable for operation at low speeds, below 500 rpm. It has the same accuracy, as to torque and horsepower capacity, as the slip-ring starter. It slips in one direction only, but the bands can be reversed for either direction of rotation.

It is primarily intended for use on apparatus that is subject to extreme overload conditions or jams, and it is used definitely to limit the number of foot-pounds of torque, or fiber stress, which can be imposed on the weakest shaft or other part of the machine to a predetermined figure.

It can be tested periodically to check its accuracy and adjustment.

The limit torque coupling is symmetrical and therefore balanced, and it transmits a balanced torque at all times. It can be adjusted to slip at any desired load between 80 and 100 per cent of its maximum capacity.

Referring to Fig. 14-126, it will be seen that one end of each band *A* is anchored to a driving pin *B*, and the other end *C* is also attached to this same driving pin but through the medium of a helical spring *D*.

The spring pulls the band around the drum and applies the necessary pressure on the friction lining, thereby gripping the drum with the correct amount of pressure to prevent slipping until the predetermined torque load is reached.

In case of lining wear, the spring will become longer, but by adjusting it back to its correct length, constancy of torque capacity will be maintained.

Table 14-105 gives the dimensions of several sizes of limit torque couplings and their torque capacity in foot-pounds.

It should be observed that the driving arm *E* is keyed to shaft *F*, and one-half the standard coupling *G* is free on the shaft and preferably fitted with bronze bushings.

Table 14-105. Limit Torque Coupling Dimensions

Diam × *W*	Capacity, ft-lb	*M*	*N*	*O*	*P*	*R*	*S*	*T*	*U*	*V*	*X*	*Y*
5 × 3	100	2	2	1¾	4	5	8½	¾	2	⅝	1⅜	2
6 × 3	200	2½	2¼	2	5	5¼	9	¾	2⅝	⅝	1⅝	2½
7 × 3	300	3	2¼	2	5½	6	9¾	¾	2⅝	⅝	1⅝	3
9 × 4	600	3½	3	2¼	6	7	11	1	3	¾	1¾	3
10 × 5	800	4	3½	2¾	6½	8	13¼	1¼	3½	¾	2	3¼
11 × 6	1,100	4¾	4¼	3	7	9	14½	1½	4	¾	2	3½
12 × 6	1,350	6	5	3	9	10	16	1½	6	1	2¼	3½
14 × 7	2,000	7½	6	3½	10	11	18	2	6¾	1¼	3¼	5
16 × 8	3,000	7½	6	3½	10	13	21	2	6¾	1¼	3¼	5

TENSION CONTROL

Accurate tension control is obtained in an unwinding operation by providing a brake with a torque retardation which is proportionately lessened as the diameter of the coil of the material being unwound is reduced.

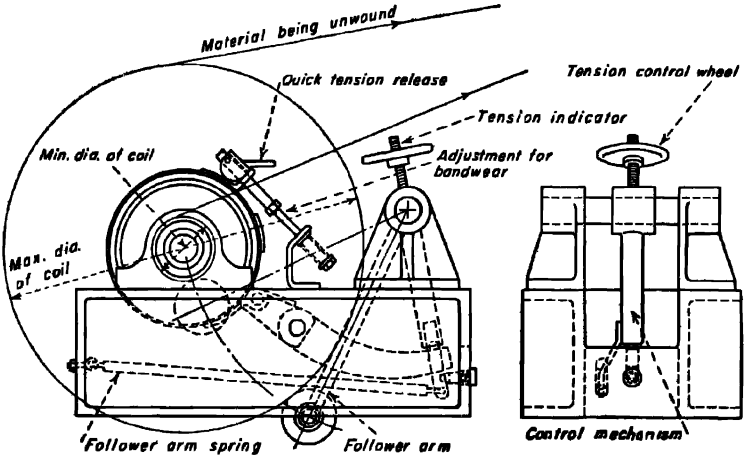


FIG. 14-127. Typical tension control for unwinding.

In the winding operation the conditions are reversed, and the torque exerted by the winder must be increased in direct proportion to the diameter of the coil of material being wound.

Obviously, therefore, there must be some connection between the coil of material and the brake or mechanism which determines the amount of the torque or tension.

In the tension control illustrated in Fig. 14-127, a follower arm or lever is so mounted that its end contacts the surface of the coil of material and follows it from the maximum coil diameter down to the minimum diameter. It is fitted with a roller of suitable diameter and length so that under a comparatively small pressure the material will not be marked or damaged.

The follower arm may also be connected by levers, shafts, and links which transmit its motion to the control mechanism.

The control mechanism contains a calibrated spring which exerts a pressure on the curved lever. The radius at which the spring imposes its pressure is directly controlled by the diameter of the coil of material. As the coil of material gets smaller, the radius at which the spring acts becomes less, and the torque capacity of the brake bands is less in proportion. The tension on the material remains constant throughout the operation.

The amount of tension on the material can be regulated within wide limits by adjusting the compression of the spring. The amount of tension is indicated by the amount of the projection of the indicator rod which projects above the "control handwheel." The amount of tension can be regulated up or down at any time, either at rest or during unwinding operations.

The tension will be constant, regardless of friction conditions and variations, at any amount of tension within the maximum and minimum ranges of the device with an accuracy of 1 per cent. The reason for this phenomenal accuracy in a purely frictional device is explained in the following discussion.

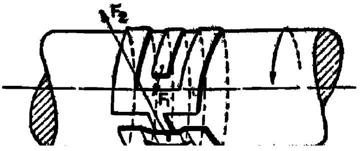


FIG. 14-128.

Theory of This Tension Device. Why the Pull Is Constant Regardless of Wide Variations of the Coefficient of Friction. The principle on which a reel tension works is as follows: In Fig. 14-128, F_1 and F_2 represent the forces on the two ends of the brake band illustrated. With the drum turning in the direction of the arrow, the relation between these two forces is represented by the usual formula for wrapping friction

$$(F_2/F_1) = e^{\mu\alpha}$$

where e is the logarithmic base (2.71828), μ is the coefficient of friction of the brake lining, and α is the angle of wrap in radians.

The retarding force T which is applied to the brake drum is expressed by the following equation:

$$T = (F_2 - F_1)$$

and therefore

$$T = F_2[1 - (1/e^{\mu\alpha})]$$

In the above equation, the value of α being determined by the design of the brake band is constant. In order to determine the effect of a change in the coefficient of friction, let us take a common value of this quantity, say $\alpha = 11$ radians, and assume for the moment that F_2 is 100 lb. Solving this equation for two extreme values of the coefficient of friction, where $\mu = 0.35$ and $\mu = 0.9$, in the first case T will equal 97.921 lb, and in the second case T will equal 99.995 lb.

Therefore, with this extreme variation of the coefficient of friction, the retarding force varies only 2 lb, which for practical purposes can be considered constant. With the ordinary friction brake, the braking force varies in direct proportion to the coefficient of friction, and if such a brake gave a retarding force of 97.921 lb with a coefficient of friction of 0.35, it would have a retarding force of about 252 lb with a coefficient of 0.9.

In the above discussion, the force (or torque) F is constant, whereas, of course, with a roll of material whose diameter is continually changing, the torque must be proportional to the roll diameter in order that the pull or tension may be constant. In the tension control herein described, this condition is provided for by the "follower lever" (or arm), which by its movement increases or decreases the torque proportionately.

TORQUE AMPLIFICATION

Many operations in industry are such that they must be performed manually because of widely varying conditions as to speed of movement or degree of movement. If the operation requires considerable effort, it is necessary to provide more or less gear reduction to make the mechanical advantage such that the operation can reasonably be performed by the human hand.

Sometimes this results in so much gear reduction that the operation becomes extremely slow, and if less gear reduction is used, the effort becomes too laborious.

Some of these manual operations are the operation of large valves, feeding heavy plates into punches, screw downs for strip mills, inching and maneuvering single-purpose machines, tilting heavy ladles, etc.

All these operations call for variability of movement from time to time which can only be determined by the judgment of the operator and at the time the operation takes place.

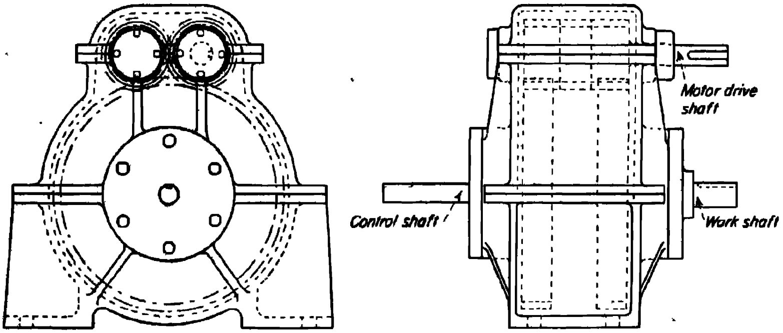


FIG. 14-129. Reversible torque amplifier.

It is generally recognized that a man cannot do more than about 1/10 hp of work turning a crank on a winch or pump. If, however, he has to be precise in his movements, such as in the adjustment of a heavy plate for punching holes with reasonable accuracy, which calls for frequent starting, stopping, and reversing movements, he will only be able to do a small percentage of 1/10 hp.

When a man is obliged to exert a large effort to turn over a machine for setting up or correction, it has been found that he cannot stop with the necessary precision accuracy.

The mechanical torque amplifier is a device that permits the operator to draw on the power of a continuously running motor by the exertion of only one-tenth of the torque effort needed to perform the operation. In effect, the torque amplifier gives the operator the use of ten times his own strength without any loss of that sensitivity of touch which he normally possesses. He can start and stop as he would on his own power, and he can make his movements as fast or as slow and as variable in speed as he would on his own power.

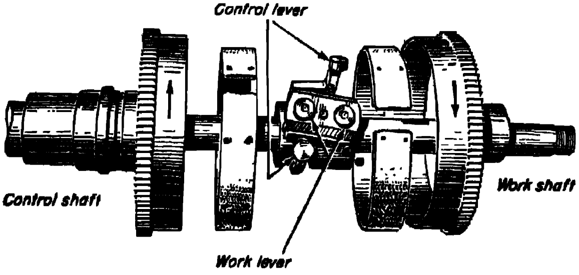


FIG. 14-130. Disassembled view of torque amplifier drums, bands, etc.

Referring to Figs. 14-129 and 14-130, which show a torque amplifier used for manual operation of the screw downs on a rolling mill, it will be noted that there are two drums which rotate continuously in opposite directions. The drums are driven by conventional spur gears and an electric motor connected to the "motor drive shaft."

Inside each drum there is an internal expanding friction band. On the "control shaft" there is a double-armed lever, the arms of which are staggered. Each arm of this lever engages in a hole in the trailing end of each band. On the "work shaft" there is also a similar double-armed lever whose arms engage in holes in the leading ends of the two bands. The "control shaft" and "work shaft" are physically connected to each other through the medium of the friction bands which repose with light touching contact within the drums.

Any movement of the "control shaft" will expand one of the bands pressing it against the drum and will contract the other band by a like amount. With the drums rotating in opposite directions, as soon as either band is expanded and makes pressure contact with the drum, the drum tends to drag the band along with it. As the band is quite flexible, the servo action increases the effort applied by the control shaft so that the front or leading end of the band presses against the arm of the lever of the "work shaft" with a greatly increased pressure. This increase in effort is represented by the equation:

$$(W/c) = e^{\mu\alpha}$$

where c = control effort
W = resultant work effort
e = natural logarithmic base (2.71828)
μ = coefficient of friction
α = angle of wrap of the band, radians

In Fig. 14-130, $\alpha = 5.9$, $\mu = 0.4$, and assume that $c = 5$ ft-lb. Then $W = 50$ ft-lb. The amplification is 10.1. The difference in the torque efforts between 50 ft-lb and 5 ft-lb is furnished by the electric motor through the "motor drive shaft."

It is obvious that an average man is not capable of exerting a torque effort of 50 ft-lb for any appreciable length of time. Moreover, his rate of motion would be very irregular, and he could not stop at any specific point with accuracy. Exerting an effort of only 5 ft-lb, he is capable of turning a crank of 1-ft radius easily and with a smooth regular motion at any rate of speed from 1 up to about 60 rpm, and he can stop at any desired point on the circle with an accuracy of plus or minus $\frac{1}{64}$ in.

The smallest fractional movement of the "control shaft" causes an exactly similar movement of the "work shaft."

The torque amplifier can be applied to operations requiring one-way or reciprocal movements in either a straight-line or circular motion.

With some applications the work force may be so large that an amplification of 10:1 is not sufficient to keep the "control force" down to a figure which is reasonable for a man to exert under the existing operating conditions as to frequency and continuity of effort. In such cases, it is possible to provide larger amplification by increasing the angle of wrap from 5.9 radians to 11.0 radians which will give an amplification of 80:1.

In Fig. 14-129, the "control shaft" was fitted with a 15-in. diameter handwheel, and the "work shaft" was connected through gearing to the screws of the rolling mill which determine the thickness of the material being rolled. Each revolution of the handwheel produces a change of 0.001 in. in the thickness of the material. It is therefore possible for the operator to adjust the rolls with extreme accuracy to produce any desired thickness of material. The effort required was approximately 5 lb on the handle of the 15-in. diameter handwheel.

A 1-hp gearmotor was used to furnish power to the revolving drums. Full power is used only while the "control shaft" is being turned. When the "control shaft" is at rest, the motor is subjected to merely an idling load.

An additional use for the torque amplifier is where synchronization of motion has to be provided between a very light piece of apparatus and a relatively massive piece of equipment. This can be done by installing a torque amplifier with a torque amplification of 100:1. The light apparatus is connected to the control shaft and the massive equipment to the work shaft of the torque amplifier. The massive equipment will then be driven by the light apparatus at exactly the same speed and will impose only one-hundredth of the torque load on the light apparatus.

BEARINGS FOR INDUSTRIAL ELECTRIC MOTORS*

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INTRODUCTION

The electric motor is, from a mechanical viewpoint, the simplest of all power sources. This simplicity, as is usually the case with simple and functional designs, is the reason the electric motor is so dependable and easy to operate. By pressing a button or throwing a switch automatically, a motor may be started on a year-long continuous run or a short predetermined working cycle with no further attention and perhaps under very adverse conditions, such as heavy overloads, a gritty and dusty surrounding atmosphere, which may in addition be loaded with corrosive or explosive gas fumes, extremely hot or extremely cold surroundings, nearby water spray, or even submersion in water or other liquids.

These characteristics of the electric motor have become so well associated with the concepts of what the motor will stand that also the bearings, although no different from the bearings in any other machine, are expected to give equally good service with little or no attention. The design and the selection of proper bearings for the electric motor has therefore become a very important feature with all motor manufacturers. Much money and extensive engineering efforts have been expended by the leading manufacturers in attempts to make the bearing performance match, as nearly as possible, other features of the electric motor.

If a motor is ordered and built for a particular purpose to operate under certain known conditions, the bearings may be designed or selected to meet such purpose or condition, and the experienced designer may not have much difficulty in making the proper choice. In the majority of cases, however, the designer of the motor does not know the purpose for which the motor will be used or under what conditions it will operate, and therefore the problem is usually to select a bearing, which may be classed as universal, to take care of most prevailing conditions.

The purpose of this subsection is to clarify some of the reasons governing the selection of certain bearings for use in electric motors.

SLEEVE- AND ANTIFRICTION-TYPE BEARINGS (GENERAL)

Sleeve Bearings. A sleeve bearing is the stationary support or guide for a revolving shaft. Usually the bearing is made from a metal having very different characteristics and properties from the material in the shaft. As one of the desired characteristics is softness, low strength is a consequence. Therefore the bearing is usually made in two parts, a soft, thin lining for wear and adaptability and an outer shell for the support of the soft lining. In machines with very rigid shafts, where the deflection of the shaft inside the bearing is negligible but where shock loads and severe vibrations would pound out a soft lining, as, for example, in railroad trucks and in crankshaft bearings for large internal-combustion engines or in geared applications, the bearings are usually made from comparatively hard bronzes without lining or with a very thin lining, as thin as 0.005 in.

Some linings are made from expensive silver alloys, others from inexpensive lead-base babbitt. Recent experiments with an aluminum–tin alloy bonded by a process developed by the Al-Fin Corporation of Jamaica, New York, is claimed to be good. The Aluminum Company of America is offering solid wall bearings from its alloy No. 750, intended for large diesel engines. As this alloy is not too expensive, it is possible that the simpler method of manufacture will make this one-piece bearing most economical.

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* Superior number refers to a specific reference listed at the end of this subsection.

The metallic lining or shell does not support the shaft directly when running; only during prolonged periods of rest does the shaft come in metallic contact with the lining. Since this condition usually exists when the machine is started from rest, the frictional resistance is high, and the wear on the lining may be excessive. During running, the shaft is supported on an oil film which, depending upon the design and the dimensions of the bearing, as well as the type of oil and the means for its circulation, may vary from only microscopic thickness to several thousandths of an inch. It may also exert considerable hydrodynamic pressure on the shaft, sufficient actually to support the shaft and whatever load the shaft may carry. The friction loss in the properly lubricated bearing is purely the result of fluid shear in the oil film, and the magnitude of the loss depends upon (1) the area of shear, (2) the rate of shear, which increases with increased velocity and decreases with increased film thickness, and (3) the viscosity of the lubricant. In properly designed and lubricated bearings, the friction loss when running is of relatively small order.

In this class are included all bearings known as slide or glider bearings for both radial and axial or thrust loading. Best known of these bearings are perhaps the Kingsbury thrust bearing and the closely related Michell bearing, which have found extensive use in large water wheels and turbogenerators. Simple modifications of this type of bearing are also used as thrust bearings in small high-speed vertical motors.*

Other types of cam bearings or multiple-collar bearings, sometimes made from nonmetallic materials such as rubber or plastic, although important in certain fields and applications, have found little use in electric motors.

One type of sleeve bearing which is extensively used in small electric motors is the oil-impregnated powdered-metal bearing, but applications of this bearing are limited to light-load and -duty operation, such as household appliances, small fans, etc.

Antifriction Bearings. Antifriction bearings are usually understood to be ball and roller bearings, including the needle roller type. The term "antifriction bearing" must not be construed to mean that these bearings are not subjected to the fundamental laws of friction. Owing to distortion of the rolling elements and the raceways under load, the motion is not a pure rolling motion, but includes some local sliding between balls and raceways which introduces minor frictional forces. In addition, the retainer or separator, which keeps the rolling elements distributed over the entire bearing, introduces sliding friction of appreciable magnitude, although far smaller than the frictional losses found in the average sleeve bearing.

Ball bearings of different makes and of different types have very important design features which will make one bearing differ very much from another as far as its load capacity and life are concerned. It is therefore important to select the proper bearing for each application.

As the balls are always of the same geometrical shape, the difference is usually found in the ball tracks of the two races, *e.g.*, deep-groove bearings, angular-contact bearings, double-row bearings, self-aligning bearings, etc. The assembling method and the construction of the retainer are also factors affecting the capacity and the serviceability of the bearings and must therefore be taken into account when selecting a bearing for a certain application.

Roller bearings, by virtue of greater area of contact between rollers and races, are theoretically capable of carrying heavier radial loads and are for this reason often used in heavy-duty motors like the mill and traction types. The cylindrical roller bearing, however, has very small thrust capacity, and also, as it is more apt to be noisy, particularly at high speed, it has not found much use in the industrial motor field.

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* The Westinghouse Electric Corporation some 15 years ago, in the development of a new high-speed vertical rayon-spinning motor, made very successful use of a flexibly supported shoe thrust bearing for the lower bearing of this 10,000- to 12,000-rpm motor.

SELECTION OF PROPER BEARINGS FOR THE ELECTRIC MOTOR

The general trend is toward the use of ball bearings in the electric motor. There are many sound engineering reasons for this, but demands from the trade have to some extent accelerated the adoption of the ball bearing for this service.

The standardization of ball and roller bearings over a wide range of sizes, which permits interchangeability of different manufacturers' products and simplifies the obtaining of spare bearings for replacements, is a factor that has promoted the popularity of these bearings. The demand for ball bearings in the electric motor has steadily increased so that today the great majority of motors built are equipped with ball bearings.

The advantages of the ball bearing are many and varied, and any one of its good points may be the most important feature in a certain application. For example, the less space occupied may be of prime importance. This is often true in the machine-tool industry and marine applications. Practically no danger of oil leakage and very infrequent lubrication may be of paramount importance in another installation, *e.g.*, in the food-processing or clothing industry. Still, in another case, low friction at starting may be the most important (in dynamometers, for example), while in some other service, little or no end play may be a necessary feature.

The one outstanding feature and advantage of the ball bearing, which by itself alone would justify the important place it occupies in the electric-motor industry, is its ability to carry a heavy thrust load in addition to a radial load. It would be difficult to picture many present-day machines without the ball bearing to carry thrust in a simple and efficient manner. The electric motor is better because of this important feature of the ball bearing.

Another feature of the ball bearing which is valuable to the electric motor is its negligible wear. This feature practically guarantees no change in the rotor air gap, and thus the efficiency of the motor is maintained through its entire life. Ball-bearing wear is usually caused by dirt or by corrosion. Generally such conditions are accompanied by noise and vibration, indicating trouble. Replacement of the bearing can thus be accomplished before any damage is done to the motor.

Ball Bearings for Motors. The majority of ball bearings in use are lubricated with grease, of which there are many types with different characteristics to meet different conditions with regard to temperature, speeds, atmospheric conditions, etc. In the electric motor, grease is the ideal lubricant because of the comparative ease whereby it is introduced and retained in the bearing housing, a feature which is particularly important as most lubricating oils are detrimental to the commonly used insulating materials in the motor windings and connections.

More ball bearings fail from too much grease than from too little. More electric motors have failed from overgreasing than from insufficient lubrication, the implication being that excess grease from the bearings may be very harmful to the motor as a whole. Frequent and indiscriminate greasing subjects the bearings and also the motor to many hazards.

To remove such hazards from the most common sizes of commercial motors, the prelubricated and sealed ball bearing was introduced on certain frame sizes and for certain service by the Westinghouse Electric Corporation, and after careful study of closely kept service records from thousands of motors, the use of these bearings has now been extended to include general-purpose motors up to 75 hp at 1,800 rpm.

These bearings are now a standard product with the leading ball-bearing manufacturers and are available with different types of greases for different applications, including silicone grease for extremely high temperatures.

The life expectancy of the prelubricated bearing, without any regreasing or upkeep, is the same as for a conventional bearing receiving the best of care and lubrication.

The prelubricated and sealed bearing has proved to be a most valuable contribution toward trouble-free operation of electric motors, and as the service records become more complete, it is probable that its use will be extended to even larger motors and to cover longer life expectancy.

This bearing is equally satisfactory for vertical mounting. It probably has even greater value as an aid to trouble-free operation in such applications, inasmuch as an excess of grease in the upper bearing of a vertical motor more readily finds its way down onto the motor winding than the excess from a horizontally mounted bearing.

The physical dimensions of available prelubricated and sealed bearings correspond to both the single- and double-row widths of both the light and medium series. The ball complement, however, is the same as in the single-row bearing. It is quite possible that regular-width double-row bearings with seals will soon be available.

Ball-bearing Mountings. The simplest mounting is obtained when end thrust is taken in one direction only by each bearing (see Fig. 14-131). This mounting is preferred wherever a small amount of end play is permissible so that no binding may occur from end to end; such binding may easily load the bearings to destruction. The end play must provide for the aggregation of all machining tolerances in the component parts of the assembly, plus any difference in expansion between the shaft and the stationary assembly. Such expansion depends upon how the motor is ventilated and how high the total temperature may go. For every 12 in. of distance between bearings, 0.003 in. should be allowed.

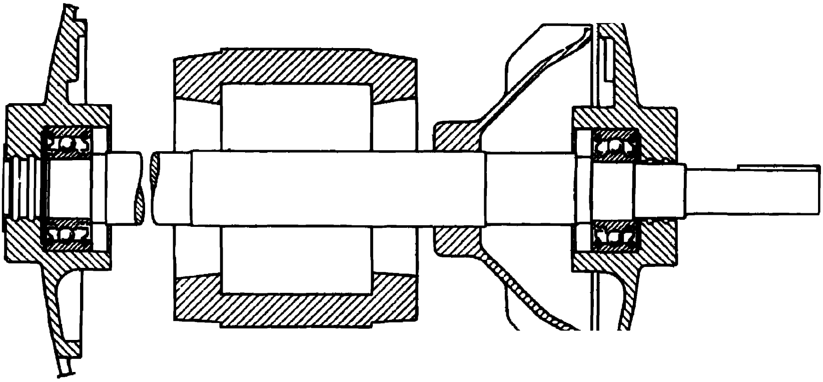

FIG. 14-131. Ball-bearing assembly. Slight end play lets either bearing take thrust load, depending upon the direction of thrust.

If the end play in a particular motor must be less than the minimum obtained by satisfying the conditions outlined, any end play in excess of what is necessary for unrestricted expansion of the shaft may be taken out by installing shimming washers behind the outer race of the one or both bearings.

If no end play is permitted outside the internal looseness in the bearings, the bearing at the shaft-extension end is locked on the shaft by a nut clamping the inner race against a shoulder of the shaft, while the outer race of the same bearing is clamped between a shoulder in the bore of the bearing housing and a suitable step cap over the housing (see Fig. 14-132). The bearing at the opposite end must be made to float in its housing to allow free expansion of the shaft.

In applications where no end play can be tolerated, special bearings with no internal looseness are furnished by most manufacturers. The standard double-row bearing is, for practical purposes, free of internal looseness in the axial direction. Figures 14-133 and 14-134 show the two methods used for taking out looseness in the double-row bearing assembly. In Fig. 14-133, the grooves for the balls are spaced slightly closer in the outer race, and the resultant load components will then converge inside the bearing. In Fig. 14-134 the inner race is grooved with slightly less spacing than the outer race, and the resultant load components will be directed to converge outside the bearing. The former method is preferred in the electric motor as this bearing has some angular flexibility, permitting slight deflection in the shaft. The mounting of preloaded bearings requires very accurate machining and line-up of the parts, as even slight misalignment will impose excessive loads on the bearings.

Wear in a ball bearing is of such small magnitude that the air gap in the motor can be considered permanently uniform, provided that the shaft is made rigid enough not to deflect too much under the combined weight of the rotor and the bending moments of all external shaft loads, plus any unbalanced magnetic loads acting on the rotor. As the efficiency of most types of motors is adversely affected by nonuniformity of the air gap, the shaft is designed to withstand these forces with little deflection, and consequently there is no need for self-aligning bearings in the standard electric motor. The self-aligning ball bearing has very low thrust capacity as compared to the rigid ball bearing and is for this reason not a suitable bearing for the general-purpose motor.

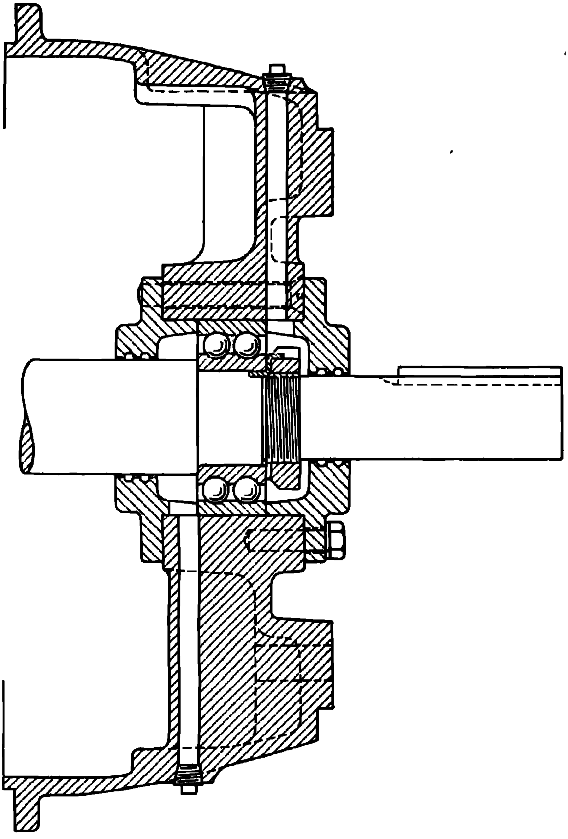

FIG. 14-132. Ball-bearing assembly. Axial locking of the bearing at the driving end permits accurate line-up of rotating load.

Motors with vertical mounting and hollow shafts, primarily for pump service, are manufactured in standard series by the leading electric-motor manufacturers. These motors are designed for heavy thrust loads. The top bearing usually takes the thrust and may be required to take thrust in both directions; the lower bearing is a guide bearing.

Since the shaft is made hollow to admit the pump shaft to the clutch on the top of the motor, the bearings become quite large, and in the larger ratings the relatively high speed attained by the balls and their retainers introduces a problem of lubrication. It has been found that, where the speed is very high, grease, particularly if there is an excess, will exert a drag on the fast-moving balls, therefore generating considerable heat which is harmful to both bearing and motor.

In one particular case of failure of a large 3,600-rpm horizontal machine, it was found that the heat generated in the grease-lubricated bearing had expanded the outer race until the bearing could not float and yield to the axial expansion of the shaft. The thrust load resulted in failure of both bearings.

The problem of heat generated in the grease can be minimized considerably by not using more grease than necessary and by providing a grease overflow so that repeated refillings cannot build up grease under pressure.

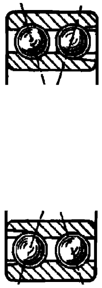

Fig. 14-133.

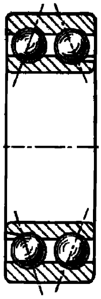

Fig. 14-134.

Figs. 14-133 and 14-134. Direction of load components in double-row bearings with no end play.

The speed limits for grease lubrication are discussed by R. K. Allen in his book, "Rolling Bearings."[1] The speed of grease lubrication in ball bearings is limited by the formula:

$$d \times N \leqq \frac{300{,}000}{}$$

where d = the diameter of the bore in the ball bearing in millimeters, and N = rpm of the shaft.

Example: What is the maximum rpm for a grease-lubricated ball bearing of the 320 size? d = 100 mm.

$$100 \times N \leqq \frac{300{,}000}{\sqrt{100/50}}$$

Hence

$$N \leqq \frac{300{,}000}{100\sqrt{2}} \leqq 2{,}100 \text{ rpm}$$

This formula is useful only as a guide, and it may have to be revised from time to time because of new and improved materials and designs. The factors likely to cause changes in this formula are improvements or developments in the greases and possibly new types of ball retainers. Other factors that may affect present concepts of maximum speed of the bearings are superfinish of the surfaces and demands for higher speeds, even at the cost of shorter life.

Regardless of how the limit of speed for grease-lubricated bearings is established, experience has shown that, for very high speeds, oil lubrication is more satisfactory. No doubt there is a wide region at moderate speeds where either system may be used successfully, so the designer may have his choice, but as the oil-lubricated bearing is very important, some of the designs that have been used in the electric motor will be discussed.

The problem of supplying the right amount of oil at all times to the bearing without risk of oil leakage has led to a number of more or less complicated designs or systems. The quantity of oil required in the ball bearing is very small; an oil mist is sufficient, but if such mist is generated in the bearing housing by agitation, the quantity is difficult to control. The mist formation depends partly upon the viscosity of the oil, and as the viscosity is a function of temperature, which varies considerably

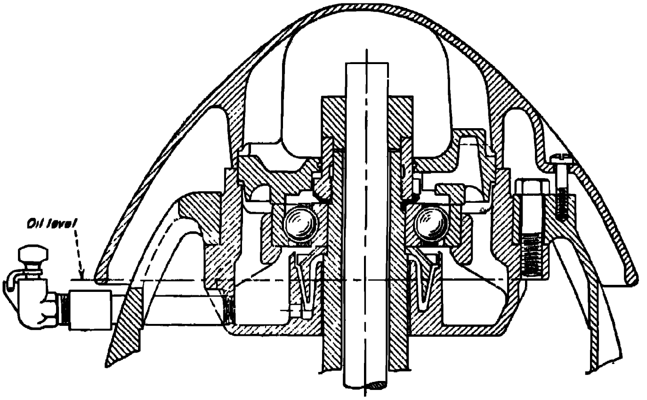

Fig. 14-135. Downward-thrust bearing in vertical hollow-shaft pump motor.

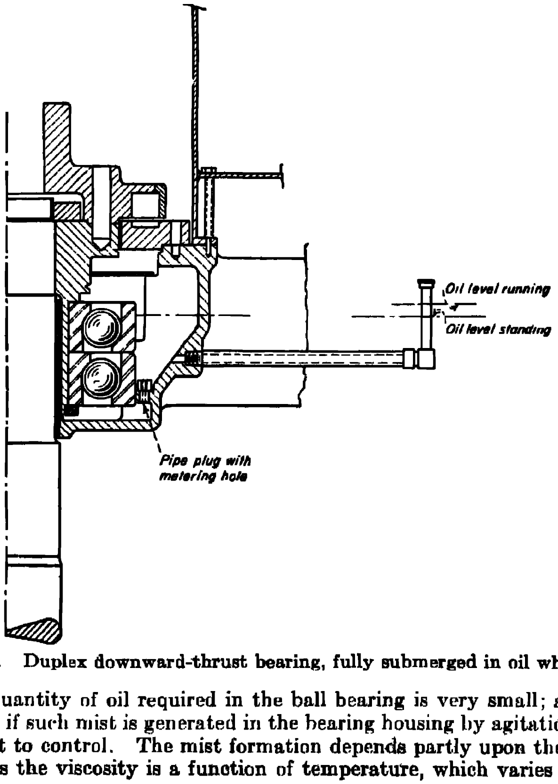

Fig. 14-136. Duplex downward-thrust bearing, fully submerged in oil while at rest.

from the condition at start and until the maximum temperature is reached, the bearing might be starved for oil or may be flooded as the temperature changes.

With this type of oil lubrication the bearing is started up from rest in a more or less dry condition, and if the rest periods are prolonged, as in the case of stand-by machines, corrosion may become a problem in the dry bearing. Atomization of the oil by means of compressed air is more positive and easier to control, but such systems become complicated and expensive for individual machines. Condensation of vapors in the air may be difficult to extract, and water is detrimental to proper lubrication.

A design that has proved reliable and satisfactory in large vertical hollow-shaft motors and at the same time is simple enough to be practical is shown by Fig. 14-135. The objection raised above with regard to danger of corrosion is possibly its greatest fault. Another objection from the manufacturing standpoint is that, if the same frame and bearing assembly is used for machines of different speeds, the tapered collar which pumps the oil to the bearing, as well as the surrounding walls of the oil reservoir, must be changed to correspond with the change in speed.

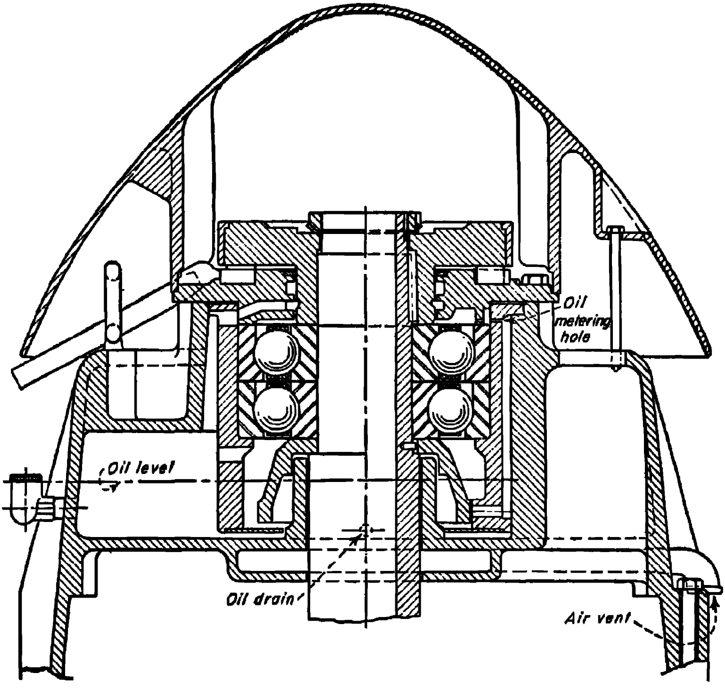

FIG. 14-137. Duplex downward-thrust bearing, lubricated during running by trickle feed.

A better method, which has found extensive use in large vertical machines but also lends itself to industrial motors, is shown by Fig. 14-136. The bearing during rest is completely or partly submerged in the oil and is thus protected from corrosion. When the motor is started, the oil is thrown out of the bearing into the annular oil reservoir surrounding the bearing. A small metering hole in a partition near the bottom of the bearing regulates the return flow to the bearing to the proper amount needed. When the motor is stopped, the metering hole equalizes the oil level, and the bearing again becomes submerged in the protective oil. This design has more flexibility for different speeds of the motor, as the size of the metering hole is the only detail that requires change. A similar method can also be used for horizontal mountings, but as the highest oil level must be well below the shaft to prevent oil leakage along the shaft, only the lowest part of the bearing, including the balls at the bottom, can be submerged in oil during rest.

A variation of a nonsubmerged bearing is shown by Fig. 14-137. The oil is pumped by a revolving collar through vertical ducts to the top of the bearing, where small metering holes admit a small quantity of oil to trickle down through the bearing. This design will also permit different speeds with simple alterations of the parts.

Figure 14-138 shows a horizontally mounted bearing with oil lubrication, where the oil is delivered to the bearing by an oil ring. This method works satisfactorily where the speed of the shaft is not extremely high.

Ball-bearing Assembly. The following points should be observed in mounting prelubricated and sealed ball bearings in electric motors:

The bearings are mounted by pressing the bearing onto the shaft against an interference fit of about 0.0005 in. per in. of shaft diameter. The mounting is preferably made by pressing on the inner race, using a tubular square-ended tool, which is guided so that it cannot exert pressure on the ball retainer or outer race. The prelubricated bearing does not lend itself to mounting by a shrinking-on method, as the heat required for expansion may injure the sealed-in lubricant or the seals of the bearing. The final location of the bearing is determined by a shoulder of less height than the radial thickness of the inner race. The shoulder height is limited to be less than the height of the inner race in order to provide a grip or a hold for a bearing puller by which the bearing may be pulled off without injury.

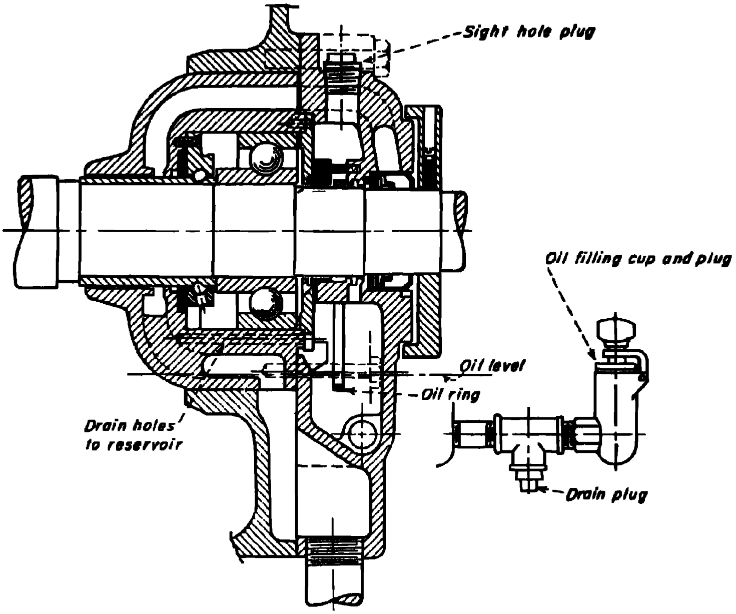

FIG. 14-138. Oil-ring-lubricated ball bearing in horizontal machine.

The bore in the bracket housing is made to give a close fit around the bearing but without actual interference. Owing to the close fit, there is a tendency for the bracket to cock and bind if it is not entered and pushed on absolutely square. Such binding should be overcome by tapping the bracket lightly near the center of the bearing housing. Repeated cocking by hard blows at the rim must be avoided, as this may permanently distort the bearing races by brinelling.

The safest way to remove the bracket is to use a suitable puller, which engages in two diametrically located boltholes, and a jackscrew against the end of the shaft.

For similar reasons of protecting the bearings from injurious shocks prior to putting the motor in service, care should be taken when mounting pulleys or other power take-offs on the shaft extension that no heavy blows are struck in the axial direction. Any such mounting should be made by light tapping with a mallet or with suitable press tools. If heavy pounding is necessary, the fit is too tight and should be relieved to where light tapping is sufficient.

Sleeve Bearings. The sleeve bearing is still a very important part of present-day machine design and deserves full consideration in any discussion of bearings and choice of bearings for electric motors.

The one outstanding advantage of the sleeve bearing, which is entirely lacking in the ball bearing and which is of great value in the assembly of, for example, motor-generator sets, is the fact that it may be made split, permitting assemblies which otherwise could not be made or would involve complicated design. Another advantage of the sleeve bearing in electric-motor applications is that the cushioning and damping effect of the oil film tends to make the motor run very smoothly and quietly, which, at high speed in particular, usually makes the sleeve-bearing motor the better performer. Some users prefer the sleeve bearing because it is comparatively easy to repair after a burnout, and thus it may not be necessary to carry spare bearings.

In case of a complete bearing failure, however, extensive damage may be done to the rest of the machine. In the electric motor, the windings and the shaft are almost certain to become damaged in case of a sleeve-bearing burnout, and therefore not too much weight should be placed on easy repair of the bearing. It is better to choose a bearing which will not cause extensive damage in case of a failure.

The sleeve bearing, because of its loose fit over the shaft journal (the looseness is usually 0.001 per in. of diameter and is never less than 0.002 for journals over 1 in. in diameter), permits easy assembling of the end bells to the frame. In this connection the end bell should be turned upside down while pushing it over the shaft so that the oil ring does not interfere with the entry of the bearing over the shaft. Even mild attempts to force the bearing on the shaft if the oil ring is interfering may distort the oil ring so that it fails to revolve freely with the shaft. The result will generally be a bearing failure.

The thrust capacity for continuous load is very small in the average sleeve bearing, yet the ability to take axial shocks at standstill is ordinarily greater than in corresponding size ball bearings. Therefore there is less danger of doing injury to the bearing in mounting pulleys or couplings on motors equipped with sleeve bearings.

Foreign matter is destructive to all bearings, and provisions must be made in the mounting to exclude gritty dust as completely as possible. If foreign matter, regardless of all precautions, enters the interior of a sleeve bearing, the danger of damage is usually less than if it enters a ball bearing. In the sleeve bearing, foreign matter is often carried away with the oil and may settle out in the bottom of the oil reservoir where it is harmless. If abrasive material lodges in the babbitt, it may score the hardest journal to destruction with no indication other than a rise in the temperature of the bearing. This may not be detected until after the damage is complete.

Usually the end play of the rotor is made considerably larger in sleeve-bearing motors than in those equipped with ball bearings. The reason for this is that the thrust capacity of the sleeve bearing is so small that even the thrust caused by imperfect alignment of the magnetic and mechanical centers may be too much for the bearings. Liberal end play will permit these centers to line up automatically, and the end thrust is thus reduced. It is claimed that end play is occasionally beneficial because of less tendency for brushes on commutators or collectors to wear grooves which may cause binding and sparking. It is probably also beneficial for the distribution of oil in the bearing. There is, however, danger of pumping action which may cause leakage of oil from the bearing, and it can increase the wear of the seals.

The most common means for supplying oil to the sleeve bearing is by an oil ring which revolves with the journal, lifting oil from the oil reservoir below and depositing it on the journal at the top of the bearing through a suitable slot. The oil is distributed over the full length of the bearing by grooves provided in the lining, and the rotation of the journal builds up a high-pressure oil film on which the journal floats with very little frictional resistance. This is the ideal condition which should prevail in all bearings having full fluid lubrication, and when it exists, good performance results. Unfortunately, the ideal is not always attained, and in the electric motor there are many instances or reasons why the ideal cannot be obtained. The most obvious of these reasons is that, in case of an external load in form of a belt or gear drive, the direction of the load may be anywhere around the 360 deg of the bearing. Therefore the oil grooves, which should not extend through the high-pressure area (which is approximately the loaded side), may be at the wrong location and thus oppose the build-up of a high-pressure film in that area.

Another reason why the sleeve bearing in the electric motor may not always work under the best conditions is the fact that the electric motor, because of its versatility, often is operating on a cycle with frequent starts and stops. At every start it takes a few revolutions to build up the oil film. In the meantime, the friction and wear are high. If the side load is high and remains during shutdown, as, for example, the load from initial belt tension, the oil is rapidly squeezed out from the loaded side, and even a short shutdown period will cause the bearing to start more or less dry. The ideal application for a sleeve bearing is where the machine is running for long periods of time without shutdowns. For frequent stops and starts the ball bearing will prove more satisfactory, not only because of less bearing troubles, but the lower bearing friction tends to reduce the starting time. As mentioned, the sleeve bearings of the electric motor are usually lined with soft babbitt to accommodate shaft deflection inside the bearing. The longer the bearing is, the more the shaft will deflect, and the more there is need for such flexibility of the bearing lining. The length of the sleeve bearing used in electric motors in the past has been rather great: 2 to 2½ times and even 3 times the shaft diameter. Usually the bearing load, when figured as evenly distributed over the projected area or length times diameter, has been limited not to exceed 100 psi, depending upon the velocity of the journal. There has been a tendency recently to shorten the length and consequently, of necessity, to increase the unit loading, but no radical changes, as, for example, attempts to meet the dimensions of the ball bearings, have as yet been made in sleeve bearings for electric motors.

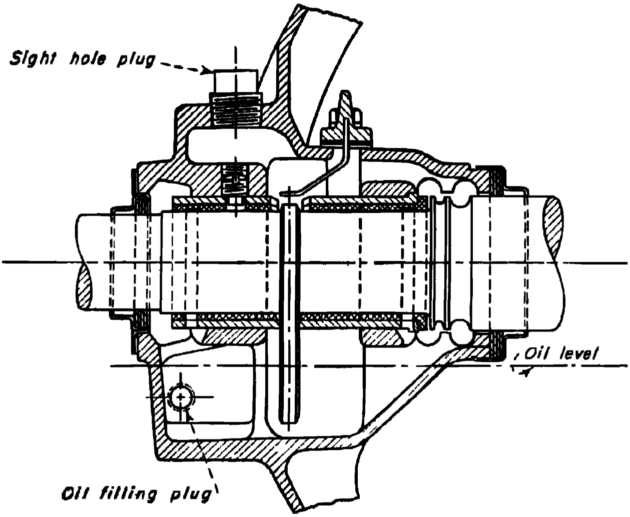

FIG. 14-139. Sleeve bearing with oil-ring lubrication and felt seals around the shaft.

Figures 14-139 and 14-140 show two typical sleeve bearings with oil-ring lubrication. Figure 14-140 is made with an air by-pass which equalizes the air pressure inside and outside the motor and thus prevents air from passing through the bearing, cutting down the oil leakage.

Oil-ring-lubricated bearings should not be used where the journal velocity exceeds 1,800 fpm. For higher speeds, forced lubrication, preferably with cooling of the oil, should be used. For very low journal speeds, a chain will carry more oil to the bearing than an oil ring and is therefore often used for such cases.

Waste-packed bearings are used in many places where speeds are not too high and the load is light, but this type of oiling is not used extensively in general-purpose motors.

It is possible that future developments in the field of superfinish of both journals and bearings will permit very thin oil films (boundary lubrication) to separate effectively the metallic surfaces at very much higher pressures than have been considered safe so far and thus bring about a revival of the sleeve bearing better to fit the present needs.

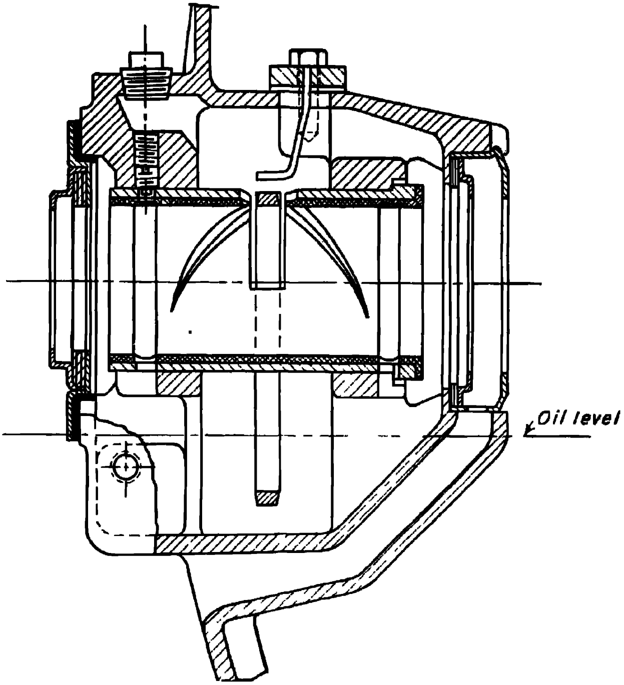

FIG. 14-140. Sealed-sleeve bearing with air by-pass.

MOTOR BEARING MAINTENANCE

The time is past when the oiler's work was classified or rated with the floor sweepers. With the high cost of shutdowns for repairs, the oiler's work has become very important and requires a high-class man properly trained for the work. The oiler should be trained to recognize signs of weakness in the bearings, such as small but unjustified temperature rise, different noises, vibrations, and appearance of the oil which may indicate oxidation or rise in acidity, and should make reports regarding these symptoms. In this way, bearings can be prevented from failing at the most critical moments of production, thus saving considerable expense. In general, the recommendations for oil and grease as given by the motor manufacturer should be followed. For special conditions (usually atmospheric), the user can, if careful performance records are kept, find more suitable lubricants for his particular conditions. The oiler should be capable of keeping such records.

The most important characteristics required of the oil for the lubrication of sleeve bearings are

1. Correct viscosity to assure best distribution of fluid-film formation for the existing operating conditions of speed, load, and temperature
2. High chemical stability in order to resist oxidation and the formation of sludge or soapy deposits
3. High film strength to minimize friction and film breakdown where boundary conditions exist
4. Ability to resist mixing with water and forming emulsions

Good grease for antifriction bearings should have

1. High chemical stability, resisting oxidation and formation of nonlubricating or corrosive products
2. Maximum resistance to separation of the oil from the soap base
3. Correct consistency at operating temperature to form protective films, minimize leakage, and reduce power loss and heat generation to a minimum

The most common cause for failure of motor bearings is the unduly tightened belt. The belt man, knowing that the new belt will stretch, installs it extremely tight. The operators of the driven machine will usually keep the belt tighter than the horsepower rating of the motor requires, particularly if heavy peak loads are a characteristic of the drive. In such cases, an oiler can prevent trouble if he is capable of judging needed belt tension.

Reference

1. ALLEN, R. K.: "Rolling Bearings," Sir Isaac Pitman & Sons, Ltd., London, 1945.

SECTION 15

ANTIFRICTION BEARINGS

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EDITOR'S NOTE

Antifriction bearings are now universally applied to every sort of plant and production equipment. It is important, therefore, that plant engineers and supervisors be reasonably well acquainted with the various types and their functional characteristics in order that they may use them intelligently. Inasmuch as plant supervision is seldom concerned with the initial selection of bearings for an original application, the procedure involved in such selection is omitted. Where such guidance is required, however, it may be obtained from bearing manufacturers' catalogues or by consultation with bearing engineers.

The concern of plant supervision is mainly with bearing maintenance. This is treated in detail under the heading of Antifriction Bearing Fundamentals. A secondary consideration is the understanding of the construction and functioning of the various types of bearings, which are covered in subsections Cylindrical and Needle Roller Bearings, The Tapered Roller Bearing, Self-aligning Roller Bearings, and Ball Bearings.

For a comprehensive treatment of the subject of antifriction bearings see the bibliography at the end of this section. For application data see manufacturers' catalogues and application handbooks.

ANTIFRICTION BEARING FUNDAMENTALS

BY BRYCE RULEY AND JOHN C. MORGAN

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Antifriction or rolling bearings are used primarily to reduce the friction between stationary and moving machine parts. Fundamentally they are superior to plain or sliding bearings because they have a much lower coefficient of friction, and their starting friction is not much greater than the running friction.

Antifriction bearings are either of the ball or roller type. Functionally they are designed to carry either pure radial or pure thrust loads or a combination of both.

Most ball and roller bearings are made up of an outer ring, an inner ring, and a set of rolling elements, balls or rollers, spaced by a cage or separator. The rolling elements are the most important since they transmit the loads from the moving parts of the machine to the stationary supports. Balls are, of course, uniformly spherical in form, but rollers may be straight cylinders, or they may be barrel- or cone- (tapered) shaped, or other forms, depending on the purpose of the design and the inventiveness of the bearing designer. The rings supply smooth, hard, accurate surfaces for the balls or rollers to roll on, while the cage or separator keeps them properly spaced circumferentially. Some types of ball and roller bearings are made without separators, and some types of roller bearings are made with only the inner or outer ring, the rollers operating directly upon a suitably hardened and ground shaft or housing.

BEARING LIFE

The individual requirements of an application and the character of the load dictate the choice of bearing type; the size of the bearing depends on the magnitude of the load. Bearings are rated according to the loads they are able to carry and the length of time they can carry such loads at certain speeds. A unique feature of antifriction bearings is that their useful life is not determined by wear but by fatigue of the operating surfaces under the repeated stresses of normal operation. Fatigue failure, which occurs as a progressive flaking or pitting of the surfaces of the raceways and rolling elements, is accepted as the basic reason for the termination of the useful life of a bearing, since all other types of failures are usually regarded as preventable.

Since the load on a bearing determines the severity of the stress on the bearing parts, it follows that, if the load is increased, the life of the bearing will be decreased, and, conversely, if the load is decreased, the life of the bearing will be increased. This relation is expressed by the equation

$$\text{Life} \propto \frac{1}{\text{load}^3}$$

Thus, if the load on a given bearing is doubled, the life expectancy is reduced to one-eighth its former value.

The life of a bearing under a given load will, therefore, be a certain number of revolutions. If this number of revolutions is used up at a relatively high rate, the life of the bearing will be correspondingly short. If the total number of revolutions is used up at a low rate, the life of the bearing will be correspondingly longer. The relation between life and speed is, therefore, expressed by the equation

$$\text{Life} \propto \frac{1}{\text{speed}}$$

All bearing-rating tables are based on extensive laboratory tests. When a number of apparently identical bearings are run to fatigue failure under a given load, a considerable dispersion is found in the individual bearing lives. This dispersion is shown by Fig. 15-1. Since the *median life* (life at which 50 per cent of the bearings have failed) or the *average life* (usually considered the life at which 60 per cent of the bearings have failed) are the most easily and accurately found, many bearing companies give their load ratings for some median or average life. This is usually given as so many hours median (or average) life at a certain speed in rpm. Other companies give their ratings in terms of the 10 *per cent failure* point, accentuating the importance of reliability in bearing applications. A rough comparison can be made between bearings rated by different methods by the rule that average life is four to five times 10 per cent failure life.

Since laboratory tests of bearings are made under good operating conditions, some allowance must be made when operating conditions in applications are unusual or unfavorable. The original designer has usually provided for this by the use of a "service factor" based on his own practical experience or on the experience and advice of bearing engineers.

Many types of bearings can carry both radial and thrust loads. For a radial bearing that can do this, the combined load is usually transformed into an equivalent radial load, *i.e.*, a pure radial load, for which the bearing would have the same life as under the actual combined load imposed. Similarly, for those few types of thrust bearings which can carry combined loads, the actual load is transformed into an equivalent thrust load. Since the method of combining loads depends to a considerable extent on the internal design of the bearings and the method of mounting them, the reader is referred to manufacturers' catalogues for specific formulas.

The capacity of a bearing as indicated by its catalogue rating in terms of life and load can be realized only if it is properly applied and properly protected by adequate seals and good lubrication.

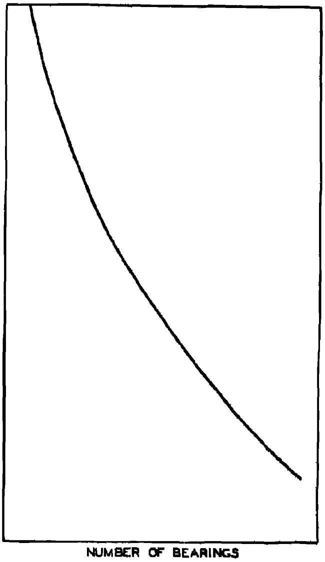


Fig. 15-1. Life dispersion.

LIMITING SPEEDS

One feature of bearing-rating tables is that they indicate limiting speeds above which the bearing will not operate satisfactorily. The designer is aware of this, of course, and gives it proper weight in the original selection of a bearing for a given application. The speed limits cannot be sharply defined since they depend on the type of bearing, the type of cage, and on the type and effectiveness of the lubrication. In general, catalogue limits should not be exceeded without consultation with bearing engineers.

LUBRICATION

The lubricant has four important functions in an antifriction bearing:

1. Lubrication of the surfaces of the bearing where sliding may occur. These are the contacting areas between the rolling elements and the cages or separators, between rollers and guide flanges, and between the rolling elements and raceways.
2. Prevent rust and corrosion by protecting the surfaces of the rolling elements and raceways.
3. Conduct heat away from the bearing and from one bearing part to another.
4. Seal out dirt and other corrosive or abrasive materials.

In low-speed applications, the lubricant may be either grease or oil. If grease is used, it should be chosen for proper consistency at the operating temperature of the bearing, and the bearing housing should be filled about half full, *never more than two-thirds full.* Oil-bath lubrication can be used for low and moderate speeds. The oil should have a viscosity of at least 70 sec Saybolt at the operating temperature, and *the oil level should be approximately even with the center of the lowermost ball or roller when the bearing is stationary.*

At high speeds circulating oil is usually necessary. This involves removing the oil from the bearing, circulating it through a cooling section, and returning it to the bearing.

In general, for any given application, the best practice is to follow the manufacturer's instructions both as to the type of lubricant and means and frequency of application.

SEALS AND CLOSURES

Since there is relative motion between the shaft and housing, some method must be used to seal in the lubricant and keep out dirt and moisture. This is the function of the seal. The seal may be an integral part of the bearing where the bearing can carry a sufficient lubricant for its operating life, or it may be a part of the housing itself. Although seals come in a wide variety of types, they may be divided into three general classes: plate seals, friction seals, and labyrinth seals.

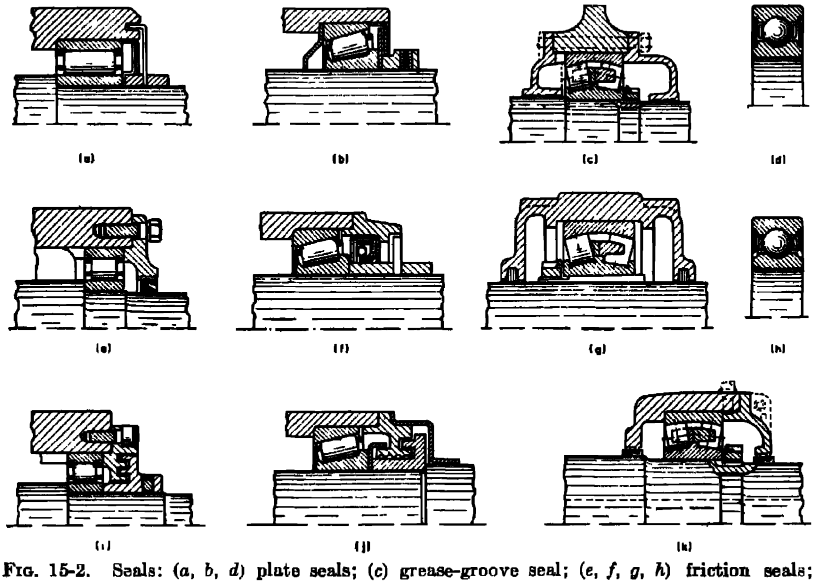


Fig. 15-2. Seals: (*a*, *b*, *d*) plate seals; (*c*) grease-groove seal; (*e*, *f*, *g*, *h*) friction seals; (*i*, *j*, *k*) labyrinth seals.

Plate seals are used with small ball bearings and grease lubrication when the operating conditions are not very severe. The dust plate is usually pressed tightly into a groove in the outer ring and comes close to but does not contact the inner race. The plate tends to keep the grease in the bearing and gives some protection against the entrance of foreign material. This seal is frictionless.

When the operating conditions are more severe, *e.g.*, when dust or abrasive materials are present, a more effective seal is required. One such seal consists of a felt ring held between two plates and pressing against a groove on the inner ring. This is, of course, a friction seal. It is much more effective in excluding dust and in retaining the lubricant but tends to increase the total friction of the bearing installation. Variations of this seal are made with rubber and synthetic materials used in place of felt.

A type of friction seal in common use consists of a felt ring which is mounted in a groove in the bore of the housing end cover and presses against the shaft surface. This is a rather effective seal when new and properly made but tends to lose some of its effectiveness after running some time since no elastic force acts to keep the felt pressed against the shaft.

Other friction seals have been designed which remedy this defect to a greater or lesser degree. A leather cuff has some elasticity so that the efficiency of the seal does not decrease very rapidly. This may be further increased by inserting a circular spring between two plates so that it gives an elastic pressure which holds the leather against the plate. Various other materials may be used in place of leather.

Labyrinth seals consist of two closely fitting rings which have a tortuous circular passage between them. One ring is mounted on the shaft and turns with it, while the other ring is integral with the housing. This seal is frictionless but is somewhat more expensive than some other types of seals.

Plate seals are used for grease lubrication only. Felt and other types of friction seals are used for both grease and oil lubrication, although felt seals for oil lubrication are usually used in conjunction with some other type of seal. Labyrinth seals are used for both grease and oil and usually are of special design to suit the operating conditions (see Fig. 15-2).

SHAFT AND HOUSING FITS

Ball and roller bearings are usually installed with either the inner ring or the outer ring a tight or interference fit. Practices vary so widely, however, depending on the type of application, the type of bearing, and the character of the load, that new application should be made only with the advice and direction of bearing engineers. Existing applications are generally supplied with instruction manuals which cover this phase of bearing practice in specific detail.

BEARING REMOVAL

Inasmuch as bearings may be applied as nonseparable units, or they may be separable so that either the outer ring or inner ring can be handled independently, it is important that the application be studied carefully to make sure that the force required to remove a bearing is applied to the proper, *i.e.*, the *tightly fitted*, part. Cleanliness is essential, of course, and the housing and machine parts adjacent to the bearing should be thoroughly clean. Proper care should also be exercised to avoid damage to the bearing in the process of removal.

The arbor press is perhaps the best means of applying the force necessary to remove bearings or individual bearing rings from shafts or housings. Regardless of the means used, however, it is absolutely essential that the driving force be directed through the tightly fitted ring.

Generally, when a piece of equipment is taken apart, the bearings stay with the member to which they are tightly fitted. Consequently, to remove a bearing, the pull or pressure should be exerted only on the tight member and so applied that it comes off straight and square, thus avoiding possible damage to both the bearing and the shaft or housing. It is important also that the force be applied only to the rings so that bearing seals or shields and separators or cages will not be damaged in the process. In the case of bearings that are made with all three major components (outer ring, inner ring, and rolling elements) separable, both rings may often be a tight fit. Then each tightly fitted ring must be removed separately, taking the usual care to avoid loading other bearing parts.

Bearing pullers are also a useful and convenient means of removing bearings and bearing rings. Driving bearings off with a hammer is the least desirable method although perhaps the one most commonly used. It is extremely hazardous, but where there is no alternative, the following rules should be observed:

1. Mount the shaft or housing in a vise wherever possible. Use wood blocks, leather, or metal guards to protect the parts from injury by the vise jaws.

2. Avoid using wood or soft metals for driving as splinters or chips may enter the bearing.

3. Do not drive directly on rings, shields or seals, separators, or on the rolling element itself.

4. In driving use smart, quick taps rather than heavy blows. This will prevent cocking.

5. Tap alternately on opposite sides to make sure the bearing or ring will move uniformly off its seat (see Fig. 15-3).

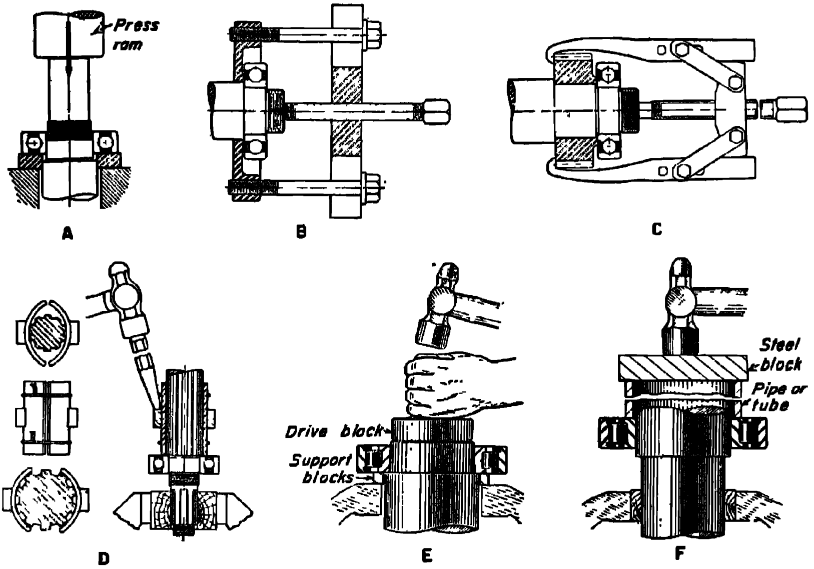


FIG. 15-3. Bearing-removal methods.

CLEANING AND INSPECTION

After the bearings are removed, they should be thoroughly cleaned and inspected to determine whether they are in proper condition for reuse.

Bearings should be placed in a metal basket and suspended in a container filled with light oil for thorough soaking. If the bearings are filled with lubricant in badly oxidized condition, they should be soaked in light oil heated to 180 to 200°F with some agitation in order to hasten the process. In case of extreme carbonization and sludging, the bearings should be soaked in a commercial solvent or in a mixture of alcohol and kerosene or safety naphtha. Only clean rags should be used for wiping and handling. Cotton waste should never be used as the short threads may get into the bearings.

A short clean bristle brush is helpful in dislodging dirt or scale from the bearing parts. Compressed air, free from condensed moisture and dirt, may also be used to blow out the bearings, but the air should never be permitted to spin the bearings as the dirt or chips present may damage the operating surfaces.

After bearings have been cleaned, they should be covered with rust-preventive compound or light oil or grease and wrapped in clean oilproof paper. If the cleaned bearings are to be reinstalled within a period up to several days, they may be left in a closed container or pan completely immersed in oil.

Before used bearings are reapplied they should be inspected carefully for damage to determine their suitability for reuse. Almost all the damage occurring to roller bearings is the result of improper handling or improper operating conditions. Only pitting or flaking may be due to natural causes as they occur when the bearing has reached the limit of its expected life.

Pitting or flaking generally appear on bearing operating surfaces that have reached the limit of their expected life, although they may also appear much earlier on surfaces stressed beyond the compressive endurance limit of the material. This phenomenon starts with the development of tiny fatigue cracks from which material fragments break loose, eventually developing craters and spreading over the entire surface of the loaded zone.

Cavities and indentations are forms of damage which are caused by the presence of dirt, chips, and even carbonized lubricants. These substances may be pressed into the roller and race operating surfaces and cause early pitting. A form of damage similar to the above but resulting from an entirely different cause and of more regular pattern occurs when bearings are subjected to vibrations or other very small movements while the bearing is stationary. This is actually a wear condition which results in markings approximating the contact area of the ball or roller and called *false brinelling*. True *brinelling* is a series of nicks or indentations on the rings corresponding to the spacing of the balls and rollers and is usually caused either during the application or removal if the driving force is permitted to travel through the rolling elements. Brinelling may also be caused by hammer blows or by the repeated application of shock loads far in excess of the bearing capacity. *Smearing* or *scoring* may be caused by careless assembly and by sliding contact between the rings and balls or rollers. *Cratering* is generally caused by electrical arcing resulting in annealing and due to rapid cooling, a rehardening of those points at which the spark occurs. There are other types of damage, of course, for which the causes are more obvious, such as *corrosion*, *cracking*, and *splitting* of rings and balls or rollers, *seizing*, and the results of the failure of cages or separators either due to wear or careless handling.

REMOUNTING BEARINGS

Before remounting the bearing in its original position, the bearing seal should be thoroughly cleaned in order to avoid scoring of the bearing rings and improper seating against shoulders, counterbores, etc. Housings should also be free from dust, dirt, and metal chips. The bearing seat and the bearing bore should be coated with lubricant to aid assembly and prevent formation of rust at the press fit. Any type of oil or grease is helpful, but best results have been obtained with lubricants containing graphite, zinc, or lead.

The pressure required to drive bearings on shafts or into housings must be applied only against the rings, never against the balls, rollers, retainers, or snap rings. As in the case of bearing removal, the pressure should be applied to the race which rotates and therefore takes the tightest fit. The pressure should be evenly distributed over the end area of the race and applied straight and square to avoid cocking which may cause scoring and possible splitting of the end ring.

Bearings may be assembled by the use of certain types of push-pullers, by arbor-press methods, and manually by the use of hammers and mallets used in conjunction with tubes and driving blocks (see Fig. 15-4).

Hammers or mallets should never be used directly on the bearing parts. The driving should be done through drive blocks or pipes or tubing faced off square and capped with a steel block or cap.

Large inner rings are usually heated in hot oil or in an electric oven to expand the bore sufficiently so that they may be shrunk in place on the shafts. Care should be taken with whatever method is used to heat the races uniformly to a temperature not exceeding 250°F. Temperatures above that may draw the temper of the races and cause soft spots rendering the race useless for further service.

In case the ring has not expanded sufficiently to slide over the bearing seat, the shaft or journal may be shrunk somewhat by packing it in dry ice. Obviously the race must be slid into position smartly as any contact with the bearing seat will result in heat transfer and consequent expansion of the shaft itself which may be rapid enough to prevent the race from being located properly.

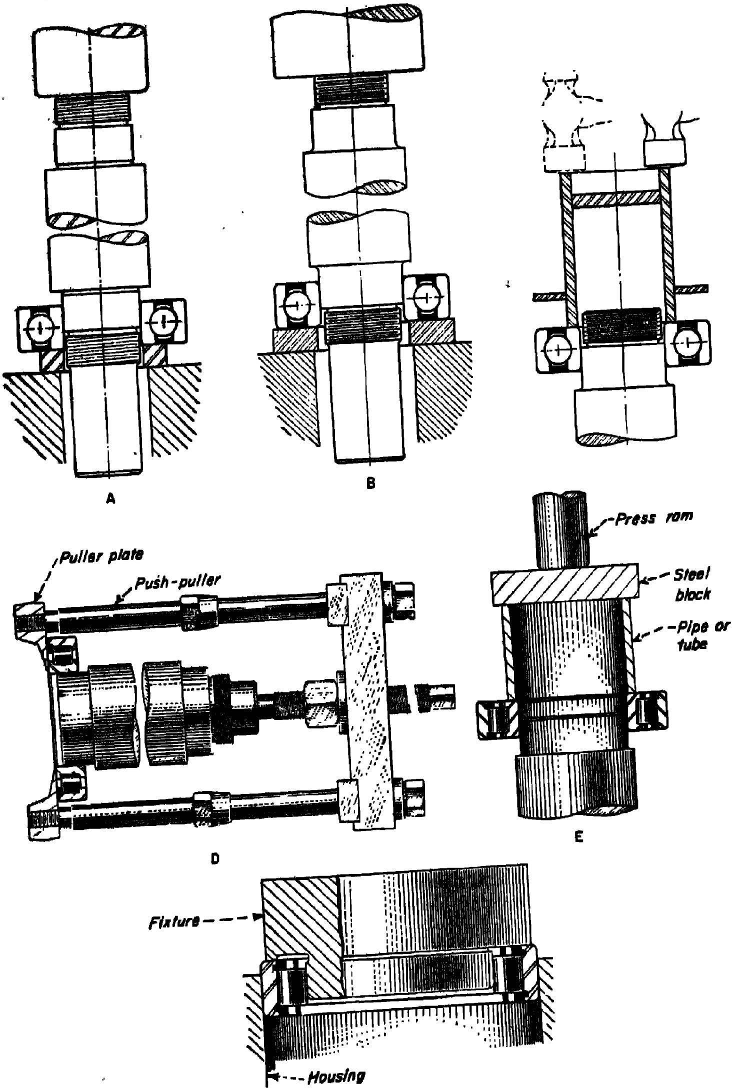


Fig. 15-4. Bearing-mounting methods.

CARE OF OTHER MACHINE ACCESSORIES ASSOCIATED WITH BEARINGS

The closure or seal used at the shaft opening should be inspected each time a bearing is serviced or replaced. Seals are designed to serve two purposes: to keep the lubricant in and to keep dirt out. They should be replaced by new seals when an examination shows that they are failing to function properly in either respect.

There are a great many types of seals, each designed for a different kind of application. One of the most common is a simple felt ring. Because they are cheap, these rings can be replaced each time the machine is overhauled. They should be replaced oftener if they show evidence of wear, scuffing, or of becoming brittle. The one-piece stamped-ring type is preferable to the split-rim type. If the method of installation requires that a split-rim washer be used, a single cut, on the bias, should be made, and care should be taken that the joint is closed properly and the washer assembled with distortion. Felt seals should be soaked in oil until thoroughly impregnated before they are installed.

Friction-type seals are used where a more positive type of closure is desired. They have as the sealing element leather, asbestos, composition, or other suitable material, usually shaped to form a wiping lip. This lip is held against the shaft at a constant pressure by means of a spring, and the whole is assembled in a steel stamping. There are many variations of this basic design, such as double seals, felt and leather, or felt and composition combinations.

The mounting of friction-type seals should be done with the greatest of care, as they are easily injured, and their effectiveness can be destroyed through mishandling. In most cases, the seal is mounted with the wiping lip toward the bearing, and since the inside diameter of the seal is less than the shaft diameter, some difficulty may be experienced in starting the seal over the end of the shaft. All sharp corners and burrs over which the seal has to travel should be removed. A tapered or bullet-shaped adaptor, recessed to fit the shaft chamfer, should be used to carry the seal over the shaft end. If keyways or splines are present, the adaptor should be extended into a thin-walled thimble which will cover the undercut section of the shaft and carry the seal to the smooth section. The seal is then mounted in the housing in the same manner and with the same care as a roller bearing. A little oil or grease should be placed on the sealing element and the shaft lubricated to assist in mounting the seal. The seal should be moved along the shaft with a twisting motion to overcome friction. A thin coat of shellac applied to the outer surface of the steel stamping will assist in making the press fit more resistant to leakage.

Gaskets are used to prevent leakage between mating flat surfaces and may also serve to adjust bearings. As they will become brittle with age and high temperature, gaskets should be renewed each time they are exposed for bearing inspection. The gasket should be fixed to one of the surfaces by a thin, even coat of shellac and permitted to dry before the parts are fastened together. Loose gasket material or frayed edges should be trimmed before assembly to prevent particles from getting into the bearing or into the lubricating system. Shims, either metallic or fibrous, should be examined with each bearing inspection. They should be cleaned and, if damaged in any manner that might affect bearing adjustment, should be replaced or repaired.

Lubrication plays such an important part in prolonging the life of roller bearings that every precaution should be taken to ensure perfect functioning of all lubricating devices. This means that all channels for the flow of oil or grease must be cleaned thoroughly, oiling devices checked, and filters renewed if necessary. With grease lubrication, the vent holes should be checked, as pressure-gun greasing may cause overlubrication. Where felts or wicks are used in lubrication, they should be cleaned or renewed as often as conditions require. Any arrangement designed to maintain a constant oil level should be checked from time to time to make sure it is functioning properly.

Keys, splines, and grinding reliefs should be cleaned carefully during any bearing inspection or replacement. Dirt becomes trapped in these recesses and corners and, when washed free by the lubricant, can become a source of bearing damage. New machine parts having keys, splines or grinding reliefs should be thoroughly washed with a petroleum solvent to remove any grinding or lapping compound that may still be present.

Threaded sections should also be carefully cleaned before assembly. The nut used on the thread should be run up before assembling the bearing on the shaft. Thus, if the threads have been improperly cut or are damaged, they can be repaired without the necessity of removing the bearing from the shaft.

Snap rings are very often used to retain bearings and races that are not secured by a heavy press fit. The snap-ring groove should be cleaned to remove all possible dirt and to ensure a firm seat for the ring. The snap ring should be ground or turned flat and be free from burrs and sharp edges. It should also be shaped so that it fits the groove snugly, not being of such diameter that it bulges or twists so as to present an uneven surface to the bearing face. The snap-ring ends should be of such design as to permit the ring to be removed easily from the application.

Washers should be flat and true to prevent distortion of the bearing when adjusting the assembly. They should be free from burrs to avoid scoring the shaft or indenting the bearing by breaking off and circulating with the lubricant. Cotter-pin holes should be cleaned with a stiff piece of wire to remove old grease and dirt. The cotter pin should be replaced after having been used once because repeated bending will weaken the legs and allow them to break while in operation.

STANDARD DIMENSIONS

Most types of bearings are made to standard dimensions so that bearings of different manufacturers have the same external dimensions and tolerances. This is true even of bearings of different types, *i.e.*, radial ball, angular-contact, self-aligning ball, and cylindrical roller types.

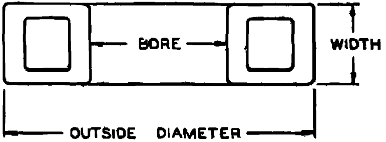
It must not be assumed, however, that bearings of the same type and size are necessarily interchangeable. Various other factors affect functional interchangeability. The requirement of each application dictates bearing choice, and substitutes for original equipment should be made only after careful and detailed comparison between the original and the prospective substitute.

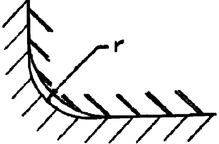
Bearing-standardization activity is now centered in the Anti-friction Bearing Manufacturers Association (AFBMA). Needless to say, the use of ball and roller bearings conforming with standardized dimensions is beneficial to the user since it provides the advantage of alternate sources of supply, generally better delivery, and lower cost.

Tables 15-1 through 15-7*c* show the dimensions of AFBMA standard ball and roller bearings.

Table 15-1. Dimensions Metric Radial Bearings

Bearing bore, d		Diameter series 0											Diameter series 1					
		Outside diam, D		Max fillet radius, in.	Dimension series 10			Dimension series 30			Outside diam, D		Max fillet radius, in.	Dimension series 31				
					Width, B		Type	Width, B		Type				Width, B		Type		
Mm	In.	Mm	In.		Mm	In.		Mm	In.		Mm	In.		Mm	In.			
4	.1575																	
5	.1969																	
6	.2362																	
7	.2756	19	.7480	.012	6	.2362	1											
8	.3150	22	.8661	.012	7	.2756	1											
9	.3543	24	.9449	.012	7	.2756	1											
10	.3937	26	1.0236	.012	8	.3150	1											
12	.4724	28	1.1024	.012	8	.3150	1											
15	.5906	32	1.2598	.012	9	.3543	1											
17	.6693	35	1.3780	.012	10	.3937	1, 5											
20	.7874	42	1.6535	.024	12	.4724	1, 5											
25	.9843	47	1.8504	.024	12	.4724	1, 5	16	.6299	8								
30	1.1811	55	2.1654	.039	13	.5118	1, 5	19	.7480	8								
35	1.3780	62	2.4409	.039	14	.5512	1, 3, 5	20	.7874	8								
40	1.5748	68	2.6772	.039	15	.5906	1, 3, 5	21	.8268	8								
45	1.7717	75	2.9528	.039	16	.6299	1, 3, 5	23	.9055	8								
50	1.9685	80	3.1496	.039	16	.6299	1, 3, 5	23	.9055	8								
55	2.1654	90	3.5433	.039	18	.7087	1, 3, 5	26	1.0236	8								
60	2.3622	95	3.7402	.039	18	.7087	1, 3, 5	26	1.0236	8								
65	2.5591	100	3.9370	.039	18	.7087	1, 3, 5	26	1.0236	8								
70	2.7559	110	4.3307	.039	20	.7874	1, 3, 5	30	1.1811	8								
75	2.9528	115	4.5276	.039	20	.7874	1, 3, 5	30	1.1811	8								
80	3.1496	125	4.9213	.039	22	.8661	1, 3, 5	34	1.3386	8								
85	3.3465	130	5.1181	.039	22	.8661	1, 3, 5	34	1.3386	8								
90	3.5433	140	5.5118	.059	24	.9449	1, 3, 5	37	1.4567	8								
95	3.7402	145	5.7089	.059	24	.9449	1, 3, 5	37	1.4567	8								
100	3.9370	150	5.9055	.059	24	.9449	1, 3, 5	37	1.4567	8								
105	4.1339	160	6.2992	.079	26	1.0236	1, 3, 5	41	1.6142	8								
110	4.3307	170	6.6929	.079	28	1.1024	1, 3, 5	45	1.7717	8	180	7.0866	.079	56	2.2047	6		
120	4.7244	180	7.0866	.079	28	1.1024	1, 3, 5	46	1.8110	8, 6	200	7.8740	.079	62	2.4409	6		
130	5.1181	200	7.8740	.079	33	1.2992	1, 3, 5	52	2.0472	8, 6	210	8.2677	.079	64	2.5197	6		
140	5.5118	210	8.2677	.079	33	1.2992	1, 3, 5	53	2.0866	8, 6	225	8.8583	.079	68	2.6772	6		
150	5.9055	225	8.8583	.079	35	1.3780	1, 3, 5	56	2.2047	8, 6	250	9.8425	.079	80	3.1496	6		
160	6.2992	240	9.4488	.079	38	1.4961	3, 5	60	2.3622	8, 6	270	10.6299	.079	86	3.3858	6		
170	6.6929	260	10.2362	.079	42	1.6535	3, 5	67	2.6378	8, 6	280	11.0236	.079	88	3.4646	6		
180	7.0866	280	11.0236	.079	46	1.8110	3, 5	74	2.9134	8, 6	300	11.8110	.098	96	3.7795	6		
190	7.4803	290	11.4173	.079	46	1.8110	3, 5	75	2.9528	8, 6	320	12.5984	.098	104	4.0945	6		
200	7.8740	310	12.2047	.079	51	2.0079	3, 5	82	3.2283	8, 6	340	13.3858	.098	112	4.4094	6		
220	8.6614	340	13.3858	.098	56	2.2047	3, 5	90	3.5433	8, 6	370	14.5669	.118	120	4.7244	6		
240	9.4488	360	14.1732	.098	56	2.2047	3, 5	92	3.6220	8, 6	400	15.7480	.118	128	5.0394	6		
260	10.2362	400	15.7480	.118	65	2.5591	5	104	4.0945	6	440	17.3228	.118	144	5.6693	6		
280	11.0236	420	16.5354	.118	65	2.5591	5	106	4.1732	6	460	18.1102	.157	146	5.7480	6		
300	11.8110	460	18.1102	.118	74	2.9134	5	118	4.6457	6	500	19.6850	.157	160	6.2992	6		
320	12.5984	480	18.8976	.118	74	2.9134	5	121	4.7638	6	540	21.2598	.157	176	6.9291	6		
340	13.3858	520	20.4724	.157	..	......	....	133	5.2362	6	580	22.8346	.157	190	7.4803	6		
360	14.1732	540	21.2598	.157	..	......	....	134	5.2756	6	600	23.6220	.157	192	7.5591	6		
380	14.9606	560	22.0472	.157	..	......	....	135	5.3150	6	620	24.4094	.157	194	7.6378	6		
400	15.7480	600	23.6220	.157	..	......	....	148	5.8268	6	650	25.5906	.197	200	7.8740	6		
420	16.5354	620	24.4094	.157	..	......	....	150	5.9055	6	700	27.5591	.197	224	8.8189	6		
440	17.3228	650	25.5906	.197	..	......	....	157	6.1811	6	720	28.3465	.197	226	8.8976	6		
460	18.1102	680	26.7717	.197	..	......	....	163	6.4173	6	760	29.9213	.236	240	9.4488	6		
480	18.8976	700	27.5591	.197	..	......	....	165	6.4961	6	790	31.1024	.236	248	9.7638	6		
500	19.6850	720	28.3465	.197	..	......	....	167	6.5748	6	830	32.6772	.236	264	10.3937	6		





NOTE: 1, radial ball bearings; 2, self-aligning ball bearings; 3, angular-contact ball bearings; 4, double-row ball bearings; 5, cylindrical roller bearings; 6, spherical roller bearings; or spherangular bearings; 8, double-row cylindrical roller bearings.

Table 15-1. Dimensions Metric Radial Bearings (*Cont.*)

Bearing bore, *d*		Diameter series 2														
		Outside diam, *D*		Max fillet radius, in.	Dimension series 02			Dimension series 22			Dimension series 32			Dimension series 92		
					Width, *B*		Type	Width, *B*		Type	Width, *B*		Type	Width, *B*		Type
Mm	In.	Mm	In.		Mm	In.		Mm	In.		Mm	In.		Mm	In.	
4	.1575	13	.5118	.012	5	.1969	1									
5	.1969	16	.6299	.012	5	.1969	1									
6	.2362	19	.7480	.012	6	.2362	1,2									
7	.2756	22	.8661	.012	7	.2756	1,2									
8	.3150															
9	.3543	26	1.0236	.024	8	.3150	1,2									
10	.3937	30	1.1811	.024	9	.3543	1,2									
12	.4724	32	1.2598	.024	10	.3937	1,2,3	..	. .	..	15.9	.6250	4, 5			
15	.5906	35	1.3780	.024	11	.4331	1,2,3	.	.		15.9	.6250	4, 5			
17	.6693	40	1.5748	.039	12	.4724	1,2,3	..	..	..	17.5	.6875	4, 5			
20	.7874	47	1.8504	.039	14	.5512	1,2,3	.	.. ..	.	20.6	.8125	4, 5			
25	.9843	52	2.0472	.039	15	.5906	1,2,3,5	..	......	..	20.6	.8125	4, 5			
30	1.1811	62	2.4409	.039	16	.6299	1,2,3,5	...	.....	..	23.8	.9375	4, 5			
35	1.3780	72	2.8346	.039	17	.6693	1,2,3,5	...	.....	..	27.0	1.0625	4, 5			
40	1.5748	80	3.1496	.039	18	.7087	1,2,3,5	..	......	..	30.2	1.1875	4, 5			
45	1.7717	85	3.3465	.039	19	.7480	1,2,3,5	...	......	..	30.2	1.1875	4, 5			
50	1.9685	90	3.5433	030	20	.7874	1,2,3,5	...	......	..	30.2	1.1875	4, 5			
55	2.1654	100	3.9370	.059	21	.8268	1,2,3,5	...	......	..	33.3	1.3125	4, 5			
60	2.3622	110	4.3307	.059	22	.8661	1,2,3,5	...	......	..	36.5	1.4375	4, 5			
65	2.5591	120	4.7244	.059	23	.9055	1,2,3,4	...	.....	..	38.1	1.5000	4, 5			
70	2.7559	125	4.9213	.059	24	.9449	1,2,3,5	...	......	..	39.7	1.5625	4, 5			
75	2.9528	130	5.1181	.059	25	.9843	1,2,3,5	...	...	.	41.3	1.6250	4, 5			
80	3.1496	140	5.5118	.079	26	1.0236	1,2,3,5	33	1.2992	6	44.4	1.7500	4, 5			
85	3.3465	150	5.9055	.079	28	1.1024	1,2,3,5	36	1.4173	6	49.2	1.9375	4, 5			
90	3.5433	160	6.2992	.079	30	1.1811	1,2,3,5	40	1.5748	6	52.4	2.0625	4, 5			
95	3.7402	170	6.6929	.079	32	1.2598	1,2,3,5	43	1.6929	6	55.6	2.1875	4, 5			
100	3.9370	180	7.0866	.079	34	1.3386	1,2,3,5	46	1.8110	6	60.3	2.3750	4, 5, 6			
105	4.1339	190	7.4803	.079	36	1.4173	1,3,5	50	1.9685	6	65.1	2.5625	4, 5			
110	4.3307	200	7.8740	.079	38	1.4961	1,2,3,5	53	2.0866	6	69.8	2.7500	4, 5, 6			
120	4.7244	215	8.4646	.079	40	1.5748	1,3,5	58	2.2835	6	76	2.9921	6	...	3.0000	5, 6
130	5.1181	230	9.0551	.098	40	1.5748	1,3	64	2.5197	6	80	3.1496	6	...	3.1250	5, 6
140	5.5118	250	9.8425	.098	42	1.6535	1,3	68	2.6772	6	88	3.4646	6	...	3.2500	5, 6
150	5.9055	270	10.6299	.098	45	1.7717	1,3	73	2.8740	6	96	3.7795	6	...	3.5000	5, 6
160	6.2992	290	11.4173	.098	48	1.8898	1,3	80	3.1496	6	104	4.0945	6	...	3.8750	5, 6
170	6.6929	310	12.2047	.118	52	2.0472	1,3	86	3.3858	6	110	4.3307	6	...	4.1250	5, 6
180	7.0866	320	12.5984	.118	52	2.0472	1	86	3.3858	6	112	4.4094	6	...	4.2500	5, 6
190	7.4803	340	13.3858	.118	55	2.1654	1	92	3.6220	6	120	4.7244	6	...	4.5000	5, 6
200	7.8740	360	14.1732	.118	58	2.2835	1	98	3.8583	6	128	5.0394	6	...	4.7500	5, 6
220	8.6614	400	15.7480	.118	..	.....	.....	108	4.2520	6	144	5.6693	6	..	5.2500	5, 6
240	9.4488	440	17.3228	.118	..	.....	......	120	4.7244	6	160	6.2992	6	..	5.7500	5, 6
260	10.2362	480	18.8976	.157	..	. ...	.	130	5.1181	6	174	6.8504	6	. .	6.2500	5, 6
280	11.0236	500	19.6850	.157	..		. ..	.	.	..	176	6.9291	6	...	6.5000	5, 6
300	11.8110	540	21.2598	.157	...	.....	...	..	....	..	192	7.5591	6	..	7.0000	5, 6
320	12.5984	580	22.8346	.157	...	.		.	.... .	.	208	8.1890	6	...	7.5000	5, 6
340	13.3858	620	24.4094	.197	.. ..	. .	..	. ..	.	..	224	8.8189	6			
360	14.1732	650	25.5906	.197	..	.	. .	.	. .	..	232	9.1339	6			
380	14.9606	680	26.7717	.197	..	. . .	. .	.	. . .	..	240	9.4488	6			
400	15.7480	720	28.3465	197	..	. . .		.		..	256	10.0787	6			
420	16.5354	760	29.9273	.236	..	. ..	. .	.	.	..	272	10.7087	6			
440	17.3228	790	31.1024	.236	..	. .	.	.	.	.	280	11.0236	6			
460	18.1102	830	32.6772	.236	..	. ...	.... .	...	. ..	..	296	11.6535	6			
480	18.8976	870	34.2520	.236	..	.	. .	.		..	310	12.2047	6			
500	19.6850	920	36.2205	.236	..	... ..	....	.	. ..	..	336	13.2283	6			

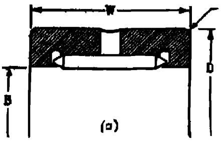
NOTE: 1, radial ball bearings; 2, self-aligning ball bearings; 3, angular-contact ball bearings; 4, double-row ball bearings; , cylindrical roller bearings; 6, spherical roller bearings; or spherangular bearings; 8, double-row cylindrical roller bearings.

Table 15-1. Dimensions Metric Radial Bearings (*Cont.*)

Bearing bore, *d*		Diameter series 3														
		Outside diam, *D*		Max fillet radius, in	Dimension series 03			Dimension series 23			Dimension series 33			Dimension series 93		
					Width, *B*		Type	Width, *B*		Type	Width, *B*		Type	Width, *B*		Type
Mm	In.	Mm	In		Mm	In		Mm	In		Mm	In		Mm	In.	
4	1575	16	6299	012	5	1969	1									
5	1969	19	7480	012	6	2362	1									
6	2362															
7	2756															
8	3150															
9	3543															
10	3937	35	1 3780	024	11	4331	1									
12	4724	37	1 4567	039	12	4724	1,2									
15	5906	42	1 6535	039	13	5118	1,2									
17	6693	47	1 8504	039	14	5512	1,2	19	7480	2	22 2	8750	4, 5			
20	7874	52	2 0472	039	15	5906	1,2,3	21	8268	2	22 2	8750	4, 5			
25	9843	62	2 4409	039	17	6693	1,2,3,5	24	9449	2	25 4	1 0000	4, 5			
30	1 1811	72	2 8346	039	19	7480	1,2,3,5	27	1 0630	2	30 2	1 1875	4, 5			
35	1 3780	80	3 1496	059	21	8268	1,2,3,5	31	1 2205	2	34 9	1 3750	4, 5			
40	1 5748	90	3 5433	059	23	9055	1,2,3,5	33	1 2992	2, 6	36 5	1 4375	4, 5			
45	1 7717	100	3 9370	059	25	9843	1,2,3,5	36	1 4173	2, 6	39 7	1 5625	4, 5			
50	1 9685	110	4 3307	079	27	1 0630	1,2,3,5	40	1 5748	2, 6	44 4	1 7500	4, 5			
55	2 1654	120	4 7244	079	29	1 1417	1,2,3,5	43	1 6929	2, 6	49 2	1 9375	4, 5			
60	2 3622	130	5 1181	079	31	1 2205	1,2,3,5	46	1 8110	2, 6	54 0	2 1250	4, 5			
65	2 5591	140	5 5118	079	33	1 2992	1,2,3,5	48	1 8898	2, 6	58 7	2 3125	4, 5			
70	2 7559	150	5 9055	079	35	1 3780	1,2,3,5	51	2 0079	2, 6	63 5	2 5000	4, 5			
75	2 9528	160	6 2992	079	37	1 4567	1,2,3,5	55	2 1654	2, 6	68 3	2 6875	4, 5			
80	3 1496	170	6 6929	079	39	1 5354	1,2,3,5	58	2 2835	2, 6	68 3	2 6875	4, 5			
85	3 3465	180	7 0866	098	41	1 6142	1,2,3,5	60	2 3622	2, 6	73 0	2 8750	4, 5			
90	3 5433	190	7 4803	098	43	1 6929	1,2,3,5	64	2 5197	6	73 0	2 8750	4, 5			
95	3 7402	200	7 8740	098	45	1 7717	1,2,3,5	67	2 6378	6	77 8	3 0625	5			
100	3 9370	215	8 4646	098	47	1 8504	1,2,3,5	73	2 8740	6	82 6	3 2500	5			
105	4 1339	225	8 8583	098	49	1 9291	1,2,3,5	77	3 0315		87 3	3 4375	5			
110	4 3307	240	9 4488	098	50	1 9685	1,2,3,5	80	3 1496	6	92 1	3 6250	5			
120	4 7244	260	10 2362	098	55	2 1654	1,3,5	86	3 3858	6					4 1250	5, 6
130	5 1181	280	11 0236	118	58	2 2835	1,3,5	93	3 6614	6					4 3750	5, 6
140	5 5118	300	11 8110	118	62	2 4409	1,3,5	102	4 0157	6					4 5000	5, 6
150	5 9055	320	12 5984	118	65	2 5591	1,3,5	108	4 2520	6					4 8750	5, 6
160	6 2992	340	13 3858	118	68	2 6772	1	114	4 4882	6					5 2500	5, 6
170	6 6929	360	14 1732	118	72	2 8346	1	120	4 7244	6					5 5000	5, 6
180	7 0866	380	14 9606	118	75	2 9528	1	126	4 9606	6					5 7500	5, 6
190	7 4803	400	15 7480	157	78	3 0709	1	132	5 1969	6					6 0000	5, 6
200	7 8740	420	16 5354	157				138	5 4331	6					6 5000	5, 6
220	8 6614	460	18 1102	157				145	5 7087	6						
240	9 4488	500	19 6850	157				155	6 1024	6						
260	10 2362	540	21 2598	197				165	6 4961	6						
280	11 0236															
300	11 8110															
320	12 5984															
340	13 3858															
360	14 1732															
380	14 9606															
400	15 7480															
420	16 5354															
440	17 3228															
460	18 1102															
480	18 8976															
500	19 6850															

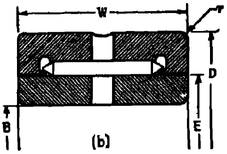
NOTE 1, radial ball bearings, 2, self-aligning ball bearings, 3, angular-contact ball bearings, 4, double-row ball bearings; 5, cylindrical roller bearings, 6, spherical roller bearings, or spherangular bearings, 8, double-row cylindrical roller bearings.

Table 15-2a. Needle Bearings
(No inner race, oil holes)



Bore, *B*	Outside diam, *D*	Width, *W*	Fillet* radius, *r*
0.3750	0.8125	0.750	0.025
0.5000	1.0000	0.750	0.025
0.6250	1.1250	0.750	0.025
0.7500	1.2500	0.750	0.040
0.8750	1.3750	0.750	0.040
1.0000	1.5000	1.000	0.040
1.1250	1.6250	1.000	0.040
1.2500	1.7500	1.000	0.040
1.3750	1.8750	1.000	0.040
1.5000	2.0625	1.250	0.060
1.6250	2.1875	1.250	0.060
1.7500	2.3125	1.250	0.060
1.8750	2.4375	1.250	0.060
2.0000	2.5625	1.250	0.060
2.2500	3.0000	1.500	0.060
2.5000	3.2500	1.500	0.080
2.7500	3.5000	1.500	0.080
3.0000	3.7500	1.500	0.080
3.2500	4.2500	2.000	0.080
3.5000	4.5000	2.000	0.080
3.7500	4.7500	2.000	0.100
4.0000	5.0000	2.000	0.100
4.2500	5.2500	2.000	0.100
4 5000	6.0000	2.500	0.100
5.0000	6.5000	2.500	0.100
5.5000	7.0000	2.500	0.100
6.0000	7.5000	2.500	0.120
6.5000	8.0000	2.500	0.120
7.2500	9.1250	3.000	0.120
7.7500	9.6250	3.000	0.120
8.2500	10.1250	3.000	0.120
8.7500	10.6250	3.000	0.160
9.2500	11.1250	3.000	0.160

Table 15-2b. Needle Bearings
(Oil holes)



Bore, *B*	Outside diam, *D*	Inner ring outside diam, *E*	Width, *W*	Fillet* radius, *r*
0.3750	1.1250	0.625	0.750	0.025
0.5000	1.2500	0.750	0.750	0.040
0.6250	1.3750	0.875	0.750	0.040
0.7500	1.5000	1.000	1.000	0.040
0.8750	1.6250	1.125	1.000	0.040
1.0000	1.7500	1.250	1.000	0.040
1.1250	1.8750	1.375	1.000	0.040
1.2500	2.0625	1.500	1.250	0.060
1.3750	2.1875	1.625	1.250	0.060
1.5000	2.3125	1.750	1.250	0.060
1.6250	2.5625	2.000	1.250	0.060
1.7500	3.0000	2.250	1.500	0.060
2.0000	3.2500	2.500	1.500	0.080
2.2500	3.5000	2.750	1.500	0.080
2.5000	3.7500	3.000	1.500	0.080
2.7500	4.2500	3.250	2.000	0.080
3.0000	4.5000	3.500	2.000	0.080
3.2500	5.0000	4.000	2.000	0.100
3.5000	5.2500	4.250	2.000	0.100
3.7500	6.0000	4.500	2.500	0.100
4.0000	6.5000	5.000	2.500	0.100
4.5000	7.0000	5.500	2.500	0.100
5.0000	7.5000	6.000	2.500	0.120
5.5000	8.0000	6.500	2.500	0.120
6.0000	9.1250	7.250	3.000	0.120
6.5000	9.6250	7.750	3.000	0.120
7.0000	10.1250	8.250	3.000	0.120
7.5000	10.6250	8.750	3.000	0.160
8.0000	11.1250	9.250	3.000	0.160

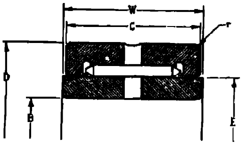


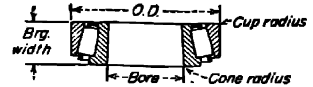
Table 15-2c. Needle Bearings
(Wide type)

Bore, *B*	Outside diam, *D*	Overall width, *W*	Outer ring width +0.000 −0.005, *C*	Min. radius or 45° bevel, *r*	Inner race outside diam, *E*
5.5000	6.5000	1.515	1.500	0.080	$5\frac{7}{8}$
6.0000	7.0000	1.515	1.500	0.080	$6\frac{3}{8}$
6.5000	7.5000	1.515	1.500	0.080	$6\frac{7}{8}$
7.0000	8.1250	1.765	1.750	0.080	$7\frac{3}{8}$
7.5000	8.6250	1.765	1 750	0.080	$7\frac{7}{8}$
8.0000	9.1250	1.765	1.750	0.100	$8\frac{3}{8}$
8.5000	9.7500	2.015	2.000	0.100	$8\frac{15}{16}$
9.0000	10.2500	2.015	2.000	0.100	$9\frac{7}{16}$
9.5000	10.7500	2.015	2.000	0.100	$9\frac{15}{16}$
10.000	11.3750	2.265	2.250	0.100	$10\frac{7}{16}$
10.5000	11.8750	2 265	2.250	0.100	$10\frac{15}{16}$
11.0000	12.5000	2.265	2.250	0.120	$11\frac{1}{2}$
11.5000	13.0000	2.265	2.250	0.120	12
12.0000	13.5000	2.265	2.250	0.120	$12\frac{1}{2}$
12.5000	14.1250	2.515	2.500	0.120	13
13.0000	14.6250	2.515	2.500	0.120	$13\frac{1}{2}$
13.5000	15.2500	2.515	2.500	0.120	$14\frac{1}{16}$

* The corner radius or chamfer on bearings must clear the maximum fillet radius given in table. All races to have oil holes centrally located.

Table 15-3. Tapered Roller Bearings

Type TS



Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius	Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius	Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius	Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius
A2037-A2126	.3750	1.2595	.3940	3/64	3/64	28158-28315	1.5748	3.1495	.8270	1/16	1/16	5584-5535	2.5000	4.8125	1.7188	9/64	1/8	73562-73875	5.6250	8.7500	1.3750	9/64	1/8
A2047-A2126	.4720	1.2595	.3940	1/32	3/64	344-332	1.5748	3.1496	.8268	9/64	3/64	559-552A	2.5000	4.8750	1.5000	9/64	1/8	82562-82950	5.6250	9.5000	2.2500	9/64	1/8
A4050-A4138	.5000	1.3775	.4330	3/64	3/64	420-414	1.5748	3.4843	1 0625	9/64	1/32	565-563	2.5000	5.0000	1.4375	9/64	1/8	82576-82950	5.7500	9.5000	2.2500	9/64	1/8
A4059-A4138	.5900	1.3775	.4330	1/32	3/64	24780-24720	1.6250	3.0000	.8750	9/64	1/32	639-632	2.5000	5.3750	1.6250	9/64	1/8	81575-81962	5.7500	9.6250	1.8750	9/64	1/8
A6062-A6157	.6250	1.5745	.4730	3/64	3/64	11162-11315	1.6250	3.1495	.7090	1/16	1/16	6379-6320	2.5625	5.3447	2.1250	9/64	1/8	99575-99100	5.7500	10.0000	2.6250	9/32	1/8
17580-17520	.6250	1.6875	.6563	1/16	1/16	342-332	1.6250	3.1496	.8268	9/64	3/64	3958-394A	2.6250	4.3307	.8661	9/64	3/64	EE107057-107105	5.7500	10.5625	2.9375	1/4	1/4
05062-05185	.6250	1.8504	.5662	1/16	3/64	26882-26820	1.6250	3.1562	1.0000	9/64	1/8	3984-3920	2 6250	4.4375	1.1875	9/64	1/8	EE217056-217112	5.7500	11.2500	3.0000	1/4	1/4
09062-09196	.6250	1.9380	.9063	1/32	1/16	3877-3820	1.6250	3 3750	1.1875	9/64	1/8	33262-33462	2.6250	4.6250	1.1875	9/64	1/8	EE750576-751200	5.7500	12.0000	2.3750	1/8	1/4
A6067-A6157	.6690	1.5745	.4730	1/32	3/64	3577-3525	1.6250	3.4375	1.1875	9/64	1/8	479-472A	2.6250	4.7244	1.1418	3/32	1/8	EE450577-451212	5.7500	12.1250	3.5000	3/8	17/64
A6075-A6157	.7500	1.5745	.4730	.040	3/64	419-414	1.6250	3.4843	1.0625	9/64	3/32	560-552A	2.6250	4.9750	1.5000	9/64	1/8	82587-82950	5.8750	9.5000	2.2500	9/64	1/8
05075-05185	.7500	1.8504	.5662	3/64	3/64	4388-4335	1.6250	3.5625	1.5625	9/64	1/8	641-632	2 6250	5.3750	1.6250	9/64	1/8	99587-99100	5.8750	10.0000	2.6250	9/32	1/8
09078-09196	.7500	1.9380	.9063	3/64	1/16	49162-49368	1.6250	3.6875	1.2500	9/64	1/8	399A-394A	2.6875	4.3307	.8661	3/32	3/64	EE560590-561275	5.9000	12.7500	3.0625	17/32	5/16
1775-1729	.7500	2.2400	.7625	1/16	3/64	46162-46368	1.6250	3.6875	1.2500	1/32	1/8	480-472A	2.6875	4.7244	1.1418	9/64	1/8	81600-81962	6.0000	9.6250	1.8750	9/64	1/8
05079-05185	.7870	1 8504	.5662	1/16	3/64	447-432	1 6250	3.7500	1.0938	9/64	3/32	560S-552A	2.6875	4.8750	1.5000	9/64	1/8	99600-99100	6.0000	10.0000	2.6250	9/32	1/8
07079-07204	.7874	2.0470	.5910	1/16	3/64	526-522	1 6250	4.0000	1.3750	9/64	1/8	570-563	2.6875	5.0000	1.4375	9/64	1/8	EE107060-107105	6.0000	10.5625	2.9375	1/4	1/4
12580-12520	.8125	1.9360	.7813	1/16	1/16	25578-25520	1.6875	3.2650	.9375	.090	1/32	33275-33462	2 7500	4.6250	1.1875	9/64	1/8	EE217060-217112	6.0000	11.2500	3.0000	5/16	1/4
07087-07204	.8750	2 0470	.5910	3/64	3/64	3579-3525	1.6875	3.4375	1.1875	9/64	1/8	482-472A	2.7500	4.7244	1.1418	9/64	1/8	EE450601-451212	6.0000	12.1250	3.5000	3/8	17/64
1380-1329	.8750	2.1250	.7625	1/16	1/16	3428-332	1.6880	3.1496	.8268	9/64	3/64	566-563	2.7500	5.0000	1.4375	9/64	1/8	EE560600-561275	6.0000	12.7500	3.0625	11/16	5/16
1755-1729	.8750	2.2400	.7625	3/64	3/64	26884-26820	1.6880	3.1562	1.0000	9/64	1/8	643-632	2.7500	5.3750	1.6250	9/64	1/8	EE560620-561275	6.2960	12.7500	3.0625	15/32	5/16
1280-1220	.8750	2.2500	.8750	1/32	1/16	25577-25520	1.6880	3.2650	.9375	9/64	1/32	655-652	2.7500	6.0000	1.6250	9/64	1/8	EE590638-591350	6.3750	13.5000	3.1250	1/4	1/4
1779-1729	.9375	2 2400	.7625	1/32	3/64	12175-12303	1.7500	3.0312	.6875	.060	.060	835-832	2.7500	6.6250	2.1250	9/64	1/8	86650-86100	6.5000	10.0000	1.8125	3/16	1/8
3659-3620	.9375	2.4375	1.1250	3/32	1/8	13175-13318	1.7500	3.1875	.7500	0	1/16	33281-33462	2.8125	4.6250	1.1875	9/64	1/8	94649-94113	6.5000	11.3750	2.5000	9/32	1/8
2685-2631	.9375	2.6150	.9375	1/32	3/64	35175-35326	1 7500	3.2650	.8750	9/64	1/32	567A-563	2.8125	5.0000	1.4375	9/64	1/8	EE219065-219122	6.5000	12.2500	3.2500	1/4	1/4
26093-26283	.9375	2.8345	.7480	3/32	1/16	25580-25520	1.7500	3.2650	.9375	9/64	1/32	645-632	2 8125	5.3750	1.6250	1/4	1/8	EE590650-591350	6.5000	13.5000	3.1250	11/16	1/4
07098-07204	.9835	2.0470	.5910	1/16	3/64	355-354A	1.7500	3.3464	.8125	3/32	3/64	567-563	2.8750	5.0000	1.4375	9/64	1/8	EE618065-618136	6.5000	13.6875	2.7500	3/8	1/4
07100-07204	1.0000	2.0470	.5910	.040	3/64	3578-3525	1.7500	3.4375	1.1875	9/64	1/8	744-742	2.8750	5.9090	1.7500	9/64	1/8	EE780655-781400	6.5000	14.0000	2.4375	5/16	5/16
15578-15520	1.0000	2.2500	.6875	3/64	1/16	49175-49368	1.7500	3 6875	1.2500	9/64	1/8	657-652	2 8750	6.0000	1.6250	9/64	1/8	EE105065-108142	6.5000	14.2500	4.1875	17/32	1/8
1986-1932	1.0000	2.3125	.7500	3/64	3/64	46176-46368	1.7500	3.6875	1.2500	9/64	1/8	568-563	2.9062	5.0000	1.4375	1/32	1/8	86669-86100	6.6929	10.0000	1.8125	3/16	1/8
15100-15250	1.0000	2.5000	.8125	9/64	3/64	438-432	1.7500	3.7500	1.0938	9/64	3/32	34300-34478	3.0000	4.7812	.9688	.080	.080	EE590675-591350	6.7500	13.5000	3.1250	1/4	1/4
2687-2631	1.0000	2.6150	.9375	3/64	3/64	527-522	1.7500	4.0000	1.3750	9/64	1/8	47680-47620	3.0000	5.2500	1.3125	1/32	1/8	EE780676-781400	6.7500	14.0000	2.4375	5/16	5/16
26100-26283	1.0000	2.8345	.7480	1/16	1/16	460-453X	1.7500	4.1250	1.1875	9/64	1/8	495A-493	3.0000	5.3750	1.1875	9/64	1/8	NA67787-67720	6.8750	9.7500	1.8750	9/64	1/8
15580-15520	1.0625	2.2500	.6875	9/64	1/16	59175-59412	1.7500	4.1250	1.4375	9/64	1/8	575-572	3.0000	5.5115	1.4375	9/64	1/8	94687-94113	6.8750	11.3750	2.5000	9/32	1/8

Table 15-8. Tapered Roller Bearings (*Cont.*)

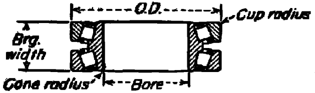
Type TS

Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius	Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius	Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius	Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius
1985-1932	1.1250	2.3125	.7500	1/32	3/64	535-532X	1.7500	4.2500	1.4375	9/64	1/8	6461-6420	3.0000	5.8750	2.1250	9/64	1/8	EE219068-219122	6.8750	12.2500	3.2500	1/4	1/4
15112-15250	1.1250	2.5000	.8125	9/64	3/64	65385-65321	1.7500	4.5000	1.7500	9/64	1/32	748S-742	3.0000	5.9090	1.7500	9/64	1/8	EE780688-781400	6.8750	14.000	2.4375	3/16	3/16
2689-2631	1.1250	2.6150	.9375	3/64	3/64	615-612	1.7500	4.7500	1.6250	9/64	1/8	659-652	3.0000	6.0000	1.6250	9/64	1/8	NA67790-67720	7.0000	9.7500	1.8750	9/64	1/8
02474-02420	1.1250	2.6875	.8750	1/32	1/16	25584-25520	1.7710	3.2650	.9375	1/16	1/32	6576-6535	3.0000	6.3750	2.1250	9/64	1/8	EE91702-91112	7.0000	11.2500	2.5000	1/4	1/8
2578-2523	1.1250	2.7500	.9375	3/32	3/64	3776-3720	1.7710	3.6718	1.1875	9/64	1/8	843-832	3.0000	6.6250	2.1250	1/4	1/8	94700-94113	7.0000	11.3750	2.5000	9/32	1/8
26112-26283	1.1250	2.8345	.7480	1/16	1/16	358-354A	1.7716	3.3464	.8125	1/16	3/64	34306-34478	3.0625	4.7812	.9688	9/64	.080	EE280702-281200	7.0000	12.0000	2.6250	1/4	1/8
3198-3120	1.1250	2.8593	1.1875	3/64	1/8	376-372A	1.7716	3.8125	.8750	1/32	1/16	47681-47620	3.1875	5.2500	1.3125	9/64	1/8	EE470078-470132	7.0000	13.2500	3.5625	1/4	1/4
02872-02820	1.1250	2.8750	.8750	1/32	1/8	18690-18620	1.8125	3.1250	.6875	7/64	1/16	496-493	3.1875	5.3750	1.1875	9/64	1/8	EE780705-781400	7.0000	14.0000	2.4375	3/16	3/16
2690-2631	1.1562	2.6150	.9375	9/64	3/64	13181-13318	1.8125	3.1875	.7500	1/32	1/16	581-572	3.1875	5.5115	1.4375	9/64	1/8	EE420701-421437	7.0000	14.3720	3.6250	1/2	1/8
17118-17244	1.1805	2.4410	.6300	1/16	1/16	359S-354A	1.8125	3.3464	.8125	3/32	3/64	740-742	3.1875	5.9090	1.7500	13/64	1/8	87737-87111	7.3750	11.1250	2.0000	9/64	1/8
15117-15250	1.1805	2.5000	.8125	3/64	3/64	2984-2924	1.8125	3.3464	1 0000	9/64	3/64	838-832	3 1875	6.6250	2.1250	1/32	1/8	NA222075-222126	7.3750	12.5970	3.5000	7/32	3/16
14117A-14276	1.1810	2 7170	.7813	9/64	3/64	436-432	1.8125	3.7500	1 0938	9/64	3/32	47686-47620	3.2500	5 2500	1.3125	9/64	1/8	67885-67820	7.5000	10.5000	1.8750	9/64	1/8
17119-17244	1.1875	2.4410	.6300	1/16	1/16	369A-362A	1.8750	3.5000	.8125	9/64	3/64	495-493	3 2500	5.3750	1.1875	9/64	1/8	87750-87111	7.5000	11.1250	2.0000	9/64	1/8
2558-2523	1.1875	2.7500	.9375	3/32	3/64	3778-3720	1.8750	3.6718	1.1875	1/4	1/8	580-572	3.2500	5.5115	1.4375	9/64	1/8	93750-93125	7.5000	12.5000	2.5000	11/64	1/8
3191-3120	1.1875	2.8593	1.1875	9/64	1/8	526-522	1.8750	4.0000	1.3750	9/64	1/8	663-652	3.2500	6.0000	1.6250	9/64	1/8	EE210753-211300	7.5000	13.0000	2.5000	9/32	1/8
15118-15250	1.1895	2.5000	.8125	9/64	3/64	5358-5335	1.8750	4.0625	1.7188	3/64	1/8	842-832	3.2500	6.6250	2.1250	9/64	1/8	EE420751-421437	7.5000	14.3720	3.6250	1/4	1/8
14119A-14276	1.1900	2.7170	.7813	1/32	3/64	463-453X	1 8750	4.1250	1.1875	3/16	1/8	498-493	3.3125	5.3750	1.1875	9/64	1/8	93787-93125	7.8750	12.5000	2 5000	11/64	1/8
1674-1620	1.2450	2.6250	.8125	1/16	1/16	536-532X	1.8750	4.2500	1.4375	9/64	1/8	749-742	3.3475	5.9090	1.7500	9/64	1/8	EE132083-132125	8.0000	12.5000	2.1250	5/32	1/8
08125-08231	1.2500	2.3125	.5781	.040	.040	617-612	1.8750	4.7500	1.6250	9/64	1/8	596-592A	3.3750	6.0000	1.5625	9/64	1/8	93800-93125	8.0000	12.5000	2.5000	11/64	1/8
02475-02420	1.2500	2.6875	.8750	9/64	1/16	5395-5335	1 9375	4.0625	1 7188	9/64	1/8	665-652	3 3750	6.0000	1.6250	9/64	1/8	EE122080-122126	8 0000	12.5000	2.6250	11/64	1/8
14125A-14276	1.2500	2.7170	.7813	9/64	3/64	366-362A	1.9685	3.5000	8125	3/32	3/64	593-592A	3 5000	6.0000	1.5625	9/64	1/8	EE420801-421437	8.0000	14.3720	2.6250	1/8	1/8
2582-2523	1.2500	2 7500	.9375	9/64	3/64	465-453X	1.9685	4.1250	1.1875	3/32	1/8	759-752	3.5000	6 3750	1.8750	9/64	1/8	EE710806-711600	8.0000	16.0000	2.5000	9/32	1/4
3193-3120	1.2500	2.8593	1.1875	9/64	1/8	18200-18337	2.0000	3.3750	.7500	.060	.060	855-854	3.5000	7.5000	2.2500	5/16	1/8	EE132084-132125	8.1250	12.5000	2.1250	5/32	1/8
02875-02820	1.2500	2.8750	.8750	9/64	1/8	368A-362A	2.0000	3.5000	.8125	9/64	3/64	69354-69630	3 5430	6.3030	1.1860	3/32	1/8	93825-93125	8.2500	12.5000	2.5000	11/64	1/8
2875-2820	1.2500	2.8750	.8750	9/64	1/8	3780-3720	2.0000	3.6718	1.1875	9/64	1/8	598-592A	3.6250	6.0000	1.5625	9/64	1/8	543085-543114	8.5000	11.4177	1.2500	9/64	1/8
3476-3420	1.2500	3.1250	1.1563	3/64	1/8	375-372A	2 0000	3.8125	.8750	3/32	1/16	77362-77675	3.6250	6.7500	1.8750	9/64	1/8	EE130851-131400	8.5000	14.0000	2.7500	17/64	1/16
346-332	1.2500	3.1496	.8268	1/32	3/64	49585-49520	2.0000	4.0000	1.2500	9/64	1/8	42368-42584	3.6875	5.8437	1.1250	.120	.120	543086-543114	8.6602	11.4177	1 2500	9/64	1/8
26126-26283	1.2600	2.8345	.7480	1/16	1/16	529X-522	2 0000	4.0000	1 3750	9/64	1/8	42375-42584	3.7500	5.8437	1.1250	.120	.120	544090-544118	9.0000	11.8125	1.3125	9/64	1/8
14130-14276	1 3125	2.7170	.7813	9/64	3/64	455-453X	2 0000	4.1250	1 1875	1/32	1/8	594-592A	3.7500	6.0000	1.5625	9/64	1/8	88900-88126	9 0000	12 8750	2.0625	1/4	1/8
2585-2523	1.3125	2.7500	.9375	9/64	3/64	59200-59412	2 0000	4.1250	1.4375	9/64	1/8	683-672	3 7500	6 6250	1 6250	9/64	1/8	98900-98140	9.0000	14.0000	2.6875	9/32	1/8
2876-2820	1.3125	2.8750	.8750	9/64	1/8	4580-4535	2.0000	4.1250	1.5625	9/64	1/8	77375-77675	3.7500	6.7500	1.8750	9/64	1/8	EE130902-131400	9.0000	14.0000	2.7500	17/32	1/16

2785-2720	1.3125	3.0000	.9375	9/64	1/8
31590-31520	1.3125	3.0000	1.1563	1/32	1/8
14137A-14276	1.3750	2.7170	.7813	1/16	3/64
02877-02820	1.3750	2.8750	.8750	9/64	1/8
2878-2820	1.3750	2.8750	.8750	1/32	1/8
25878-25820	1.3750	2.8750	.9375	9/64	3/32
2786-2720	1.3750	3.0000	.9375	13/64	1/8
36137-36300	1.3750	3.0000	1.1563	1/16	1/8
31593-31520	1.3750	3.0000	1.1563	9/64	1/8
3478-3420	1.3750	3.1250	1.1563	9/64	1/8
335-332	1.3750	3.1496	.8268	1/32	3/64
3379-3320	1.3750	3.1562	1.1563	9/64	1/8
417-414	1.3750	3.4843	1.0625	1/32	1/32
449-432	1.3750	3.7500	1.0938	1/32	3/32
19138-19283	1.3770	2.8345	.6700	1/16	1/16
339-332	1.3779	3.1496	.8268	1/32	3/64
26883-26820	1.3779	3.1562	1.0000	1/32	1/8
2794-2720	1.4365	3.0000	.9375	9/64	1/8
19143-19283	1.4375	2.8345	.6700	1/16	1/16
16143-16284	1.4375	2.8440	.8125	9/64	3/64
3878-3820	1.4375	3.3750	1.1875	1/32	1/8
13889-13836	1.5000	2.5625	.5000	1/16	1/32
13685-13621	1.5000	2.7170	.7500	9/64	3/32
19150-19283	1.5000	2.8345	.6700	.060	1/16
16150-16284	1.5000	2.8440	.8125	9/64	3/64
2788-2720	1.5000	3.0000	.9375	9/64	1/8
3490-3420	1.5000	3.1250	1.1563	9/64	1/8
28151-28315	1.5000	3.1495	.8270	9/64	1/16
337-332	1.5000	3.1496	.8268	1/32	3/64
3381-3320	1.5000	3.1562	1.1563	9/64	1/8
3876-3820	1.5000	3.3750	1.1875	9/64	1/8
418-414	1.5000	3.4843	1.0625	9/64	1/32
49150-49368	1.5000	3.6875	1.2500	9/64	1/8
444-432	1.5000	3.7500	1.0938	9/64	3/32
542-532X	1.5000	4.2500	1.4375	9/64	1/8
26881-26820	1.5625	3.1562	1.0000	9/64	1/8
3382-3320	1.5625	3.1562	1.1563	9/64	1/8
422-414	1.5625	3.4843	1.0625	9/64	1/32
537-532X	2.0000	4.2500	1.4375	9/64	1/8
3975-3920	2.0000	4.4375	1.1875	9/64	1/8
619-612	2.0000	4.7500	1.6250	9/64	1/8
555-552A	2.0000	4.8750	1.5000	3/32	1/8
6279-6220	2.0000	5.000	2.0000	9/64	1/8
3767-3720	2.0625	3.6718	1.1875	3/32	1/8
33890-33821	2.0625	3.7500	1.0938	1/16	3/32
377-372A	2.0625	3.8125	.8750	3/32	1/16
540-532X	2.0625	4.2500	1.4375	9/64	1/8
389A-382	2.1250	3.8750	.8268	1/32	1/32
456-453X	2.1250	4.1250	1.1875	9/64	1/8
4595-4535	2.1250	4.1250	1.5625	9/64	1/8
539-532X	2.1250	4.2500	1.4375	9/64	1/8
621-612	2.1250	4.7500	1.6250	9/64	1/8
557S-552A	2.1250	4.8750	1.5000	9/64	1/8
6280-6220	2.1250	5.0000	2.0000	9/64	1/8
636-632	2.1250	5.3750	1.6250	9/64	1/8
385-382	2.1653	3.8750	.8268	3/32	1/32
389-382	2.1880	3.8750	.8268	3/32	1/32
28682-28622	2.2500	3.8437	.9688	9/64	1/32
387A-382	2.2500	3.8750	.8268	9/64	1/32
462-453X	2.2500	4.1250	1.1875	3/32	1/8
390-394A	2.2500	4.3307	.8661	3/32	3/64
3979-3920	2.2500	4.4375	1.1875	9/64	1/8
623-612	2.2500	4.7500	1.6250	9/64	1/8
555S-552A	2.2500	4.8750	1.5000	9/64	1/8
65225-65500	2.2500	5.0000	1.7500	9/64	1/8
6375-6320	2.2500	5.3447	2.1250	11/64	1/8
29582-29520	2.3622	4.2500	1.0000	1/32	1/8
397-394A	2.3622	4.3307	.8661	1/32	3/64
28985-28920	2.3750	4.0000	1.0000	9/64	1/8
3980-3920	2.3750	4.4375	1.1875	9/64	1/8
6376-6320	2.3750	5.3447	2.1250	9/64	1/8
39250-39412	2.5000	4.1250	.8438	.080	.080
395-394A	2.5000	4.3307	.8661	9/64	3/64
3982-3920	2.5000	4.4375	1.1875	9/64	1/8
33251-33462	2.5000	4.6250	1.1875	1/32	1/8
477-472A	2.5000	4.7244	1.1418	1/32	1/8
685-672	3.8750	6.6250	1.6250	9/64	1/8
779-772	3.8750	7.1250	1.8750	9/64	1/8
52400-52618	4.0000	6.1875	1.4375	9/64	1/8
687-672	4.0000	6.6250	1.6250	9/64	1/8
780-772	4.0000	7.1250	1.8750	9/64	1/8
861-854	4.0000	7.5000	2.2500	5/16	1/8
782-772	4.1250	7.1250	1.8750	9/64	1/8
37425-37625	4.2500	6.2500	.9063	9/64	1/8
64450-64700	4.5000	7.0000	1.6250	9/64	1/8
71450-71750	4.5000	7.5000	1.8750	9/64	1/8
938-932	4.5000	8.3750	2.6250	9/32	1/8
68462-68712	4.6250	7.1250	1.3750	9/64	1/8
795-792	4.7500	8.1250	1.8750	1/8	1/8
67388-67320	5.0000	8.0000	1.8125	9/64	1/8
74500-74850	5.0000	8.5000	1.8750	9/64	1/8
95500-95925	5.0000	9.2500	2.5000	1/4	1/8
EE116050-116097	5.0000	9.7500	2.5000	1/8	3/16
EE153050-153100	5.0000	10.0000	3.0625	3/8	1/4
EE540502-541162	5.0000	11.6250	3.2500	17/32	1/4
EE455051-455116	5.0000	11.6250	3.3750	1/4	1/4
EE580500-581200	5.0000	12.0000	3.1250	1/4	1/8
EE750502-751200	5.0000	12.0000	2.3750	1/4	1/4
48506-48750	5.0625	7.5000	1.3750	9/64	1/8
799-792	5.0625	8.1250	1.8750	1/8	1/8
797-792	5.1181	8.1250	1.8750	9/64	1/8
67390-67320	5.2500	8.0000	1.8125	9/64	1/8
74525-74850	5.2500	8.5000	1.8750	9/64	1/8
95525-95925	5.2500	9.2500	2.5000	3/8	1/8
NA48390-48320	5.3750	7.5000	1.5625	9/64	1/8
74537-74850	5.3750	8.5000	1.8750	9/64	1/8
EE580537-581200	5.3750	12.0000	3.1250	1/4	1/8
74550-74850	5.5000	8.5000	1.8750	9/64	1/8
73551-73875	5.5000	8.7500	1.3750	9/64	1/8
898-892	5.5000	9.0000	2.2500	9/64	1/8
82550-82950	5.5000	9.5000	2.2500	9/64	1/8
99550-99100	5.5000	10.0000	2.6250	9/32	1/8
EE540550-541162	5.5000	11.6250	3.2500	3/8	1/4
EE750558-751200	5.5000	12.0000	2.3750	1/8	1/4
EE450551-451212	5.5000	12.1250	3.5000	3/8	17/64
EE430900-431575	9.0000	15.7500	3.5000	13/32	1/8
EE710906-711600	9.0000	16.0000	2.5000	9/32	1/4
EE700091-700167	9.0000	16.7500	4.0000	9/32	1/4
88925-88128	9.2500	12.8750	2.0625	1/4	1/8
EE8575-8520	9.2500	12.8750	2.0625	1/4	1/8
96925-96140	9.2500	14.0000	2.6875	9/32	1/8
EE125094-125145	9.4930	14.5000	2.6875	1/4	1/8
EE127095-127140	9.5000	14.0000	2.2500	1/4	1/8
EE170950-171436	9.5000	14.3720	2.0000	1/4	1/8
EE125095-125145	9.5000	14.5000	2.6875	1/4	1/8
EE923095-923175	9.5000	17.5000	4.0000	1/4	3/16
EE295950-295193	9.5000	19.2500	4.7500	1/4	1/4

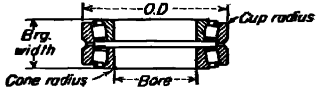
Table 15-3. Tapered Roller Bearings (*Cont.*)

Type TDI



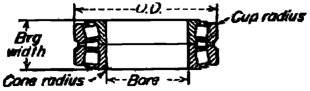
Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius
17116D-17244	1.1875	2.4410	1.3306	1/32	1/16
14126D-14276	1.2500	2.7170	1.5625	1/16	3/64
14134D-14276	1.3125	2.7170	1.5625	1/16	3/64
19145D-19283	1.4375	2.8345	1.5392	1/32	1/16
19152D-19283	1.5000	2.8345	1.5392	1/32	1/16
13169D-13318	1.6875	3.1875	1.3750	1/32	1/16
358D-354A	1.6875	3.3464	1.9790	1/16	3/64
13176D-13318	1.7500	3.1875	1.3750	0	1/16
13182D-13318	1.8125	3.1875	1.3750	1/32	1/16
376DE-372A	1.8750	3.8125	2.0940	1/32	1/16
378DE-372A	1.9375	3.8125	2.0940	1/32	1/16
375D-372A	2.0000	3.8125	2.0940	1/32	1/16
78216D-78551	2.1650	5.5130	2.6020	3/32	3/32
399D-394A	2.4375	4.3307	2.1870	1/32	3/64
78251D-78551	2.5000	5.5130	2.6020	3/32	3/32
765D-752	3.0000	6.3750	4.0000	9/64	1/8
496D-493	3.1875	5.3750	2.3750	1/16	1/8
581D-572	3.1875	5.5115	3.1875	1/16	1/8
498D-493	3.3125	5.3750	3.0000	1/32	1/8
767D-752	3.5000	6.3750	4.0000	1/16	1/8
865D-854	3.5000	7.5000	4.6250	3/8	1/8
42362D-42584	3.6250	5.8437	2.2500	1/16	120
867D-854	3.7500	7.5000	4.6250	1/4	1/8
52388D-52618	3.8750	6.1875	3.1563	1/16	1/8
779D-772	3.8750	7.1250	4.0000	1/16	1/8
52400D-52618	4.0000	6.1875	3.1563	1/16	1/8
868D-854	4.0000	7.5000	4.6250	1/16	1/8
945D-932	4.0000	8.3750	5.6250	13/32	1/8
782D-772	4.1250	7.1250	4.0000	1/16	1/8
71426D-71750	4.2500	7.5000	3.8750	1/16	1/8
946D-932	4.2500	8.3750	5.6250	1/8	1/8
95426D-95925	4.2500	9.2500	5.5000	33/64	1/8
71450D-71750	4.5000	7.5000	3.8750	1/16	1/8
938D-932	4.5000	8.3750	5.6250	1/8	1/8
95451D-95925	4.5000	9.2500	5.5000	33/64	1/8
EE116048D-116098	4.6250	9.8750	6.0000	5/16	3/16
95474D-95925	4.7500	9.2500	5.5000	1/4	1/8
EE153047D-153100	4.7500	10.0000	6.3750	1/2	1/4
EE153048-153100	4.8750	10.0000	6.3750	7/16	1/4
97500D-97900	5.0000	9.0000	6 3125	1/16	1/8

Type TDO



Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius
05075-05185D	.7500	1.8504	1.2500	9/64	1/32
17118-17245D	1.1805	2.4410	1.5625	1/16	1/32
26118-26284D	1.1805	2.8338	1.6835	1/16	1/32
24118-24262D	1.1875	2.6150	1.7500	1/16	1/32
15118-15251D	1.1895	2.5000	1.8125	9/64	1/32
2580-2524D	1.2500	2.7500	2.6250	1/32	1/32
02877-02823D	1.3750	3.0000	1.8750	9/64	1/32
3482-3424D	1.3750	3.2500	2.6250	1/32	1/32
28150-28318D	1.5000	3.1510	1.8125	1/16	1/32
447-432D	1.6250	3.7500	2.4375	9/64	1/32
22168-22325D	1.6875	3.2500	1.7500	090	1/32
355-353D	1.7500	3.5480	2.0000	3/32	1/32
369A-363D	1.8750	3.5433	1.9687	9/64	1/32
3775-3729D	2.0000	3.6718	2.5625	1/32	1/32
455-452D	2.0000	4.2500	2.5625	1/32	1/32
537-533D	2.0000	4.3750	3.1250	9/64	1/16
387-384D	2 2500	3.9370	2.0625	3/32	1/32
395-394D	2 5000	4.3307	2.0625	9/64	1/32
559-552D	2.5000	4.8750	3.1250	9/64	1/16
639-632D	2.5000	5.3750	3.7500	9/64	1/16
480-472D	2.6875	4.7244	2.5625	9/64	1/32
566-563D	2.7500	5.0000	3.1875	9/64	1/16
659-654D	3.0000	6.0000	3.7500	9/64	1/16
7488-742D	3.0000	6.1250	4.0000	9/64	1/16
755-752D	3.0000	6.3750	4.1250	9/64	1/16
496-493D	3.1875	5.3750	2.7500	9/64	1/32
581-572D	3.1875	5.5115	3.2500	9/64	1/32
842-834D	3.2500	6.7500	4.9375	9/64	1/32
598-592D	3.6250	6.0000	3.2500	9/64	1/32
683-672D	3.7500	6.6250	3.6250	9/64	1/32
780-774D	4.0000	7.1250	4.1250	9/64	1/16
861-854D	4.0000	7.5000	5.0000	5/16	1/16
936-932D	4.2500	8.3750	5 6250	5/16	1/16
64450-64700D	4.5000	7.0000	3.6250	9/64	1/32
71450-71751D	4.5000	7.5000	4.1875	9/64	1/16
795-792D	4.7500	8.1250	4.2500	1/8	1/32
67388-67322D	5.0000	7.7500	4.0000	9/64	1/32
74500-74851D	5.0000	8.5000	4.1875	9/64	1/16
95500-95927D	5.0000	9.2500	5.6250	1/4	1/16
82550-82932D	5.5000	9.3125	5.1875	9/64	1/16

Type TNA



Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius
NA05075-05185D	.7500	1.8504	1.2500	9/64	1/32
NA17098-17245D	.9835	2.4410	1.5625	1/16	1/32
NA26118-26284D	1.1805	2.8338	1.6836	1/16	1/32
NA28138-28318D	1.3770	3.1510	1.8125	1/16	1/32
NA357-353D	1.5748	3.5480	2.0000	1/16	1/32
NA22171-22325D	1.7188	3.2500	1.7500	.090	1/32
NA438-432D	1.7500	3.7500	2.4375	9/64	1/32
NA366-363D	1.9685	3.5433	1.9688	9/64	1/32
NA3780-3729D	2.0000	3.6718	2.5625	9/64	1/32
NA455-452D	2.0000	4.2500	2.5625	9/64	1/32
NA539-533D	2.1250	4.3750	3.1250	9/64	1/16
NA385-384D	2.1653	3.9370	2.0625	9/64	1/32
NA397-394D	2.3622	4.3307	2.0625	9/64	1/32
NA476-472D	2.3622	4.7244	2.5625	9/64	1/32
NA558-552D	2.3750	4.8750	3.1250	9/64	1/16
NA569-563D	2.6250	5.0000	3.1875	9/64	1/16
NA643-632D	2 7500	5.3750	3.7500	9/64	1/16
NA484-472D	2.7559	4.7244	2.5625	9/64	1/32
NA495A-493D	3.0000	5.3750	2.7500	9/64	1/32
NA659-654D	3.0000	6.0000	3.7500	9/64	1/16
NA580-572D	3.2500	5.5115	3.2500	9/64	1/32
NA749-742D	3.2500	6.1250	4.0000	9/64	1/16
NA842-834D	3.2500	6.7500	4.9375	9/64	1/32
NA593-592D	3.5000	6.0000	3.2500	9/64	1/32
NA759-752D	3.5000	6.3750	4.1250	9/64	1/16
NA52375-52637D	3.7500	6.3750	3.2500	9/64	1/32
NA776-774D	3.7500	7.1250	4.1250	9/64	1/16
NA691-672D	4.0000	6.6250	3.6250	9/64	1/32
NA861-854D	4.0000	7.5000	5.0000	9/64	1/16
NA782-774D	4.1250	7.1250	4.1250	9/64	1/16
NA71450-71751D	4.5000	7.5000	4.1875	9/64	1/16
NA938-932D	4.5000	8.3750	5.6250	9/64	1/16
NA798-792D	5.0000	8.1250	4.2500	9/64	1/32
NA95500-95927D	5.0000	9.2500	5.6250	9/64	1/16
NA74525-74851D	5.2500	8.5000	4.1875	9/64	1/16
NA48390-48320D	5.3750	7.5000	3.3750	9/64	1/32
NA81550-81963D	5.5000	9.6250	4.2500	9/64	1/16
NA82576-82932D	5.7500	9.3125	5.1875	9/64	1/16
NA82587-82932D	5.8750	9.3125	5.1875	9/64	1/16
NA81600-81963D	6.0000	9.6250	4.2500	9/64	1/16

95499D-95925	5.0000	9.2500	5.5000	19/64	1/8
EE153053D-153100	5.0000	10.0000	6.3750	1/8	1/4
EE540501D-541162	5.0000	11.6250	5.8750	35/64	1/4
EE455050D-455116	5.0000	11.6250	6.5000	17/32	1/4
74512D-74850	5.1250	8.5000	4.0000	1/16	1/8
73512D-73875	5.1250	8.7500	2.3850	9/64	1/8
67390D-67320	5.2500	8.0000	3.6250	1/16	1/8
EE455052D-455116	5.2500	11.6250	6.5000	3/8	1/4
EE450531D-451250	5.3110	12.5000	6.3750	17/32	17/64
EE455053D-455116	5.3750	11.6250	6.5000	11/32	1/4
73550D-73875	5.5000	8.7500	2.3850	9/64	1/8
EE455048-455116	5.5000	11.6250	6.5000	1/8	1/4
EE450530-451250	5.5000	12.5000	6.3750	17/32	17/64
EE92558D-92988	5.5620	9.8750	4.3750	1/16	3/16
EE517056D-517117	5.6230	11.7300	4.2500	1/8	1/8
EE517057-517117	5.6875	11.7500	4.2500	1/8	1/8
81576D-81962	5.7500	9.6250	3.4375	1/16	1/8
EE450575D-451250	5.7500	12.5000	6.3750	17/32	17/64
99587D-99100	5.8750	10.0000	4.7500	1/16	1/8
81601D-81962	6.0000	9.6250	3.4375	1/16	1/8
99603D-99100	6.0000	10.0000	6.2500	1/16	1/8
EE517061D-517117	6.0000	11.7500	4.2500	1/8	1/8
EE450600D-451250	6.0000	12.5000	6.3750	3/8	17/64
EE217063D-217114	6.2500	11.4375	4.9375	1/8	1/4
82680D-82620	7.0000	11.0000	4.4375	1/16	1/8
EE91700D-91112	7.0000	11.2500	4.1875	1/16	1/8
94704D-94113	7.0000	11.3750	6.2500	1/16	1/8
EE280700D-281200	7.0000	12.0000	4.3086	1/8	1/8
EE210700D-211300	7.0000	13.0000	4.3750	1/16	1/8
EE222074D-222126	7.3750	12.5970	6.6250	1/8	3/16
93751D-93125	7.5000	12.5000	5.2500	1/4	1/8
EE210750D-211300	7.5000	13.0000	4.3750	1/8	1/8
EE420750D-421450	7.5000	14.5000	6.2500	1/8	1/8
EE132078D-132125	7.8750	12.5000	3.7500	1/8	1/8
93580D-93520	8.0000	12.5000	4.1563	1/16	1/8
93801D-93125	8.0000	12.5000	5.2500	1/4	1/8
EE132081D-132125	8.0000	12.5000	3.7500	1/8	1/8
EE420800D-421450	8.0000	14.5000	6.2500	1/8	1/8
9975D-9920	8.5000	13.0000	8.0000	1/8	1/8
EE130850-131400	8.5000	14.0000	4.7500	1/16	1/16
96851D-96140	8.5000	14.0000	5.0000	1/4	1/8
EE130888D-131400	8.8750	14.0000	6.5000	5/16	1/16
EE130903D-131400	9.0000	14.0000	6.5000	5/16	1/16
EE529091D-529157	9.0000	15.7500	5.5000	1/8	1/8
EE430901D-431575	9.0000	15.7500	6.2500	1/8	1/8
EE700090D-700167	9.0000	16.7500	7.0000	9/64	1/4
EE8575D-8520	9.2500	12.8750	3.6875	1/16	1/8
EE127094D-127140	9.4970	14.0000	4.2500	1/16	1/8
8880D-8820	9.5000	13.5000	3.6250	1/16	1/8
EE170951D-171450	9.5000	14.5000	3.6500	1/16	1/8
EE821096D-821165	9.5000	16.5000	7.0000	1/4	1/4
EE750558-751204D	5.5000	12.0000	5.3438	1/8	1/16
81575-81963D	5.7500	9.6250	4.2500	9/64	1/16
99575-99102D	5.7500	10.0000	5.8750	9/32	1/16
EE450601-451215D	6.0000	12.1250	7.8750	3/8	3/32
94649-94114D	6.5000	11.3750	5.6250	9/32	1/16
EE780688-781376D	6.8750	13.7500	5.5000	3/16	1/16
EE91702-91113D	7.0000	11.2500	5.3750	1/4	1/16
EE280702-281201D	7.0000	12.0000	5.8204	1/4	1/16
87750-87112D	7.5000	11.1250	4.2500	9/64	1/16
93750-93127D	7.5000	12.5000	5.7500	11/64	1/16
EE210753-211301D	7.5000	13.0000	5.5000	9/32	1/16
EE420793-421451D	7.9375	14.5000	7.6250	1/8	1/16
EE132083-132126D	8.0000	12.5000	5.0000	5/32	1/16
543085-543115D	8.5000	11.3125	2.7500	9/64	1/32
EE820085-820161D	8.5000	16.0000	7.6875	1/4	1/16
EE130899-131401D	8.8750	14.0000	6.0000	17/64	1/16
96900-96140D	9.0000	14.0000	6.0000	9/32	1/32
EE430900-431576D	9.0000	15.7500	7.3750	13/32	1/16
EE700091-700168D	9.0000	16.7500	8.2500	9/32	1/16
EE8575-8520D	9.2500	12.8750	4.5000	1/4	1/16
EE127095-127136D	9.5000	13.7460	5.0000	1/4	1/16
EE170950-171451D	9.5000	14.5000	4.7500	1/4	1/16
EE275095-275156D	9.5000	15.5000	6.1875	1/4	1/16
EL923095-923176D	9.5000	17.5000	8.2500	1/4	1/16
EE295950-295192D	9.5000	19.2500	10.0000	1/4	1/16
NA99600-99102D	6.0000	10.0000	5.6250	9/64	1/16
NA94650-94114D	6.5000	11.3750	5.6250	9/64	1/16
NA67787-67720D	6.8750	9.7500	4.0625	9/64	1/32
NA67790-67720D	7.0000	9.7500	4.0625	9/64	1/32
NA87700-87112D	7.0000	11.1250	4.2500	9/64	1/16
NA94700-94114D	7.0000	11.3750	5.6250	7/32	1/16
NA222075-222127D	7.3750	12.6250	7.3125	7/32	1/16
NA537075-537103	7.5000	10.2500	2.6250	9/64	1/32
NA93800-93127D	8.0000	12.5000	5.7500	7/32	1/16
NA132083-132126D	8.0000	12.5000	4.7500	1/4	1/16
NA130902-131401D	9.0000	14.0000	5.7500	1/4	1/16
NA170950-171451D	9.5000	14.5000	4.7500	1/4	1/16

Type TNAS (see illustration for Type TNA)

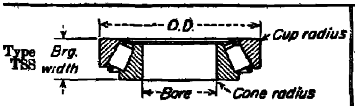
Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius
NA21075-21226D	.7500	2.2500	1.9375	1/16	1/32
NA41125-41294D	1.1250	2.9375	2.1875	3/32	1/16
NA43131-43319D	1.3125	3.1875	2.1875	3/32	1/16
NA44156-44363D	1.5625	3.6250	2.1875	3/32	1/16
NA44163-44363D	1.6250	3.6250	2.1875	1/32	1/16
NA53176-53376D	1.7500	3.7500	2.5625	3/32	1/32
NA55200-55444D	2.0000	4.4375	2.5625	3/32	1/16
NA66212-66462D	2.1250	4.6250	2.8750	9/64	1/32
NA72212-72488D	2.1250	4.8750	3.0625	3/32	1/16
NA78250-78549D	2.5000	5.5000	3.0625	3/32	1/16
NA9378-9320D	3.0000	7.0000	4.3125	9/64	3/32
NA98350-98789D	3.5000	7.8750	4.5625	9/64	3/32
NA97450-97901D	4.5000	9.0000	4.5625	9/64	3/32

Type TDOS (see illustration for Type TDO)

Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius
21075-21226D	.7500	2.2500	1.9375	1/16	1/32
41100-41294D	1.0000	2.9375	2.1875	3/32	1/16
43125-43319D	1.2500	3.1875	2.1875	1/16	1/16
44150-44363D	1.5000	3.6250	2.1875	3/32	1/16
53176-53376D	1.7500	3.7500	2.5625	3/64	1/32
55200-55444D	2.0000	4.4375	2.5625	9/64	1/16
72200-72488D	2.0000	4.8750	3.0625	9/64	1/16
66212-66462D	2.1250	4.6250	2.8750	9/64	1/32
78250-78549D	2.5000	5.5000	3.0625	3/32	1/16
9385-9320D	3.3125	7.0000	4.3125	9/64	3/32
98335-98789D	3.3465	7.8750	4.5625	9/64	3/32
97500-97901D	5.0000	9.0000	4.5625	9/64	3/32
EE114080-114161D	8.0000	16.0000	7.7500	1/4	1/8
EE113091-113171D	8.9045	17.0000	7.7500	1/4	1/8

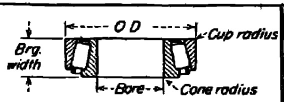
Table 15-3. Tapered Roller Bearings (*Cont.*)

Type TSS



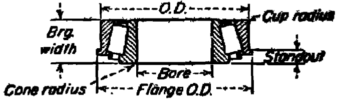
Bearing number	Bore	Outside diameter	Bearing width	Cone radius	Cup radius
11590-11520	.6250	1.6875	.5625	1/16	1/16
21075-21212	.7500	2.1250	.8750	1/16	3/32
23100-23256	1.0000	2.5625	.8750	1/16	1/16
41100-41286	1.0000	2.8593	.9688	3/32	1/16
41125-41286	1.1250	2.8593	.9688	3/16	1/16
43112-43312	1.1250	3.1250	1.0000	1/32	1/16
43125-43312	1.2500	3.1250	1.0000	1/16	1/16
44150-44348	1.5000	3.4843	1.0000	3/32	1/16
44168-44348	1.5625	3.4843	1.0000	9/64	1/16
53162-53375	1.6250	3.7500	1.2188	1/16	1/32
53177-53375	1.7500	3.7500	1.2188	9/64	1/32
55200-55443	2.0000	4 4375	1.1875	9/64	1/8
72200-72487	2.0000	4 8750	1.4375	9/64	1/8
72212-72487	2.1250	4.8750	1.4375	9/64	1/8
78215-78551	2.1250	5.5130	1.4375	9/64	3/32
66585-66520	2.3622	4.8125	1.3125	9/64	1/8
9180-9121	2.4375	6.0000	1.8750	9/64	1/8
78250-78551	2.5000	5.5130	1.4375	3/32	3/32
9185-9121	2.6875	6.0000	1.8750	9/64	1/8
9285-9220	3.0000	6.3750	1.9375	9/64	1/8
9380-9321	3.0000	6.7500	1.9375	9/64	1/8
98316-98788	3.1496	7.8740	2.0772	9/64	1/8
9385—9321	3.3125	6.7500	1.9375	9/64	1/8
98335-98788	3.3465	7.8740	2.0772	9/64	1/8
98350-98788	3.5000	7.8740	2.0772	9/64	1/8
90381-90744	3.6125	7.4375	2.0000	9/64	1/8
98400-98788	4.0000	7 8740	2.0772	9/64	1/8
EE215040-215096	4.0000	9.8750	3.0000	1/4	1/8
97450-97900	4.5000	9.0000	2.1250	9/64	1/8
EE514045-514110	4.5000	11.0000	3.2500	1/4	1/4
97493-97900	4.9330	9.0000	2.1250	9/64	1/8
97500-97930	5 0000	9.0000	2.1250	9/64	1/8
EE516050-516120	5.0000	12.0000	3.5000	1/4	1/4
97503-97900	5.0312	9.0000	2.1250	9/64	1/8
EE516057-516120	5.7500	12.0000	3.5000	1/4	1/4
EE518061-518135	6.1250	13 5000	3.3750	1/4	1/4
EE117063-117148	6 3750	14 7500	3 4375	1/4	1/16
EE607070-607140	7.0000	14 0000	3 1250	1/4	1/4

Type TST



Bearing number	Bore	Taper per foot	Outside diameter	Bearing width	Cone radius	Cup radius
15575T-15520	.9375	1.00	2 2500	.6875	1/16	1/16
07098T-07204	.9835	1 00	2.0470	.5910	1/16	3/64
1757T-1729	1.0000	1.00	2.2400	.7625	1/32	3/64
2688T-2631	1.0625	1 00	2.6150	.9375	1/16	3/64
26112T-26283	1.1250	1.00	2.8345	.7480	1/16	1/16
26117T-26283	1.1770	1.00	2.8345	.7480	1/16	1/16
2583T-2523	1.1875	1.00	2.7500	.9375	1/16	3/64
14123T-14276	1 2500	1.00	2.7170	.7813	1/16	3/64
2580T-2523	1.2500	1.00	2.7500	.9375	1/16	3/64
14582T-14525	1.3125	1.00	2.6875	.8125	1/32	3/32
2581T-2523	1 3125	1.00	2.7500	.9375	1/32	3/64
14132T-14276	1.3125	1.00	2 7170	.7813	1/16	3/64
2790T-2720	1.3125	1.00	3.0000	.9375	1/32	1/8
25877T-25820	1.3750	1.00	2.8750	.9375	1/16	3/32
2787T-2720	1.3750	1.00	3.0000	.9375	1/16	1/8
3379T-3320	1.3750	1 00	3.1562	1.1563	1/32	1/8
2791T-2720	1.4062	1 00	3 0000	.9375	9/64	1/8
25879T-25820	1.4375	1.00	2 8750	.9375	1/16	3/32
3380T-3320	1.4375	1.00	3.1562	1.1563	1/16	1/8
2798T-2720	1.4855	1.00	3.0000	.9375	1/16	1/8
26876T-26820	1 4855	1.00	3.1562	1.0000	1/16	1/8
19150T-19283	1 5000	1 00	2 8345	.6700	1/16	1/16
2882T-2820	1 5000	1 00	2 8750	.8750	1/32	1/8
2788T-2720	1 5000	1.00	3.0000	.9375	1/16	1/8
3490T-3420	1.5000	1.00	3.1250	1.1563	9/64	1/8
3391T-3320	1.5000	1.00	3.1562	1.1563	9/64	1/8
415T-414	1.5000	1.00	3.4843	1 0625	1/32	1/32
26879T-26820	1 5495	1.00	3.1562	1.0000	1/16	1/8
11156T-11315	1 5625	1.00	3.1495	.7090	1/16	1/16
3382T-3320	1 5625	1.00	3.1562	1.1563	9/64	1/8
3575T-3525	1 5625	1.00	3.4375	1.1875	1/16	1/8
422T-414	1.5625	1.00	3.4843	1.0625	3/32	1/32
3384T-3320	1.6250	1.00	3.1562	1 1563	1/16	1/8
419T-414	1.6250	1.00	3.4843	1.0625	9/64	1/32
439T-432	1.6250	1.00	3.7500	1.0938	1/32	3/32
3578T-3525	1 7500	1.00	3.4375	1.1875	9/64	1/8
435T-432	1 7500	1 00	3.7500	1.0938	1/32	3/32
349T-332	1.7688	9°0′	3.1496	.8268	1/32	3/64

Type TSF



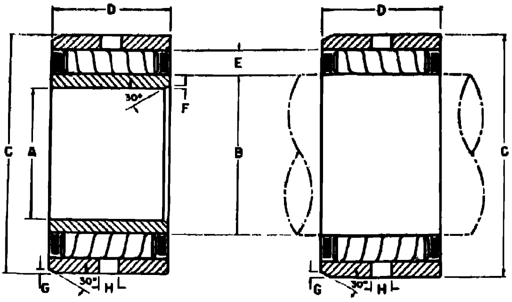
Bearing number	Bore	Outside diameter	Flange outer diameter	Stand out	Bearing width	Cone radius	Cup radius
A2047-A2126B	4720	1 2595	1.3845	.1753	.3940	1/32	3/64
A4050-A4138B	.5000	1.3775	1 5025	.1631	.4330	3/64	3/64
A6062-A6157B	.6250	1.5745	1.6995	.1918	.4730	3/64	3/64
05075-05185B	.7500	1.8504	2 0064	.2381	.5662	3/64	1/16
21075-21212B	.7500	2.1250	2.2812	.4063	.8750	1/16	3/32
1380-1328B	.8750	2.0625	2.2187	.3563	.7625	1/16	1/16
1755-1729B	.8750	2.2400	2.3960	.2938	.7625	3/64	1/32
07093-07204B	.9375	2.0470	2.2030	.2004	.5910	1/16	1/16
1986-1931B	1.0000	2.3750	2.5310	.3125	.7813	3/64	1/32
23100-23256B	1.0000	2.5625	2.7187	.4063	.8750	1/16	1/16
41100-41286B	1.0000	2.8593	3.0781	.5000	.9688	3/32	1/16
2689-2631B	1.1250	2.6150	2.7710	.3438	.9375	3/64	1/32
17118-17244B	1.1805	2.4410	2.5970	.2075	.6300	1/16	1/16
26116-26283B	1.1805	2.8345	2.9905	.2793	.7480	1/16	1/16
3191-3120B	1.1875	2.8593	3.0473	.4375	1.1875	9/64	1/8
08125-08231B	1.2500	2 3125	2.4685	.2653	.5781	.040	.040
02475-02420B	1.2500	2.6875	2.8437	.3438	.8750	9/64	1/16
14125A-14276B	1.2500	2.7170	2.8730	.3125	.7813	9/64	1/32
2582-2523B	1.2500	2.7500	2.9060	.3438	.9375	9/64	1/16
3478-3420B	1.3750	3.1250	3.3130	.4063	1.1563	9/64	1/8
19138-19283B	1.3770	2.8345	2.9905	.2481	.6700	1/16	1/16
16143-16284B	1.4375	2.8440	3 0000	.3438	.8125	9/64	1/32
28151-28315B	1.5000	3.1495	3.3055	.3583	.8270	9/64	1/16
3381-3320B	1.5000	3.1562	3.3436	.4063	1.1563	9/64	1/8
44150-44348B	1.5000	3.4843	3.7023	.5313	1.0000	3/32	1/16
26881-26822B	1.5625	3.1250	3.2812	.3438	.9375	9/64	1/32
344-332B	1.5748	3.1496	3.3370	.3125	.8268	9/64	1/32
3577-3525B	1.6250	3.4375	3.6255	.4375	1.1875	9/64	1/8
447-432B	1.6250	3.7500	3.9680	.4375	1.0938	9/64	3/32
355X-354B	1.7500	3.3464	3.5339	.3125	.8125	9/64	1/16
53177-53387B	1.7500	3.8750	4.1250	.6563	1.2188	9/64	1/32
369A-362B	1 8750	3.5433	3.7308	.3499	.7874	9/64	1/32
18200-18337B	2 0000	3.3750	3.5310	.3906	.7500	.060	.060
3780-3720B	2.0000	3.6718	3.8598	.4375	1.1875	9/64	1/8
529X-522B	2.0000	4.0000	4.2500	.5625	1.3750	9/64	1/8
4558-453B	2 0000	4.2500	4.4680	.4375	1.0938	9/64	1/32
537-532B	2 0000	4.3750	4.6250	.5625	1.5000	9/64	1/8
619-612B	2 0000	4 7500	5.0312	.6563	1.6250	9/64	1/8

EE350701-351687	7.0000	16.8750	4.1875	1/4	1/4
EE350750-351687	7.5000	16.8750	4.1875	1/4	1/4
EE380080-380190	8.0000	19 0000	4 6250	1/4	1/4
EE390090-390200	9.0000	20 0000	4.6250	1/4	1/4
EE390095-390200	9.5000	20.0000	4.6250	1/4	1/4

25584T-25520	1.7708	1.00	3.2650	.9375	1/16	1/32
359T-354A	1.8125	1.00	3.3464	.8125	1/32	3/64
463T-453X	1.8438	1.00	4.1250	1.1875	1/16	1/8
536T-532X	1.8750	1.00	4.2500	1.4375	9/64	1/8
537T-532X	2.0000	1.00	4.2500	1.4375	9/64	1/8
65199T-65500	2.0000	2.00	5.0000	1.7500	9/64	1/8
377T-372A	2.0625	1.00	3.8125	.8750	3/32	1/16
28677T-28622	2.0625	1.00	3.8437	.9683	9/64	1/32
368T-362A	2.0852	9°0′	3 5000	.8125	1/32	3/64
539T-532X	2.1250	1.00	4.2500	1 4375	9/64	1/8
65212T-65500	2.1250	2.00	5.0000	1.7500	9/64	1/8
467T-453X	2.2500	1.00	4.1250	1.1875	9/64	1/8
468T-453X	2.3125	1.00	4.1250	1 1875	9/64	1/8
388T-382	2.3340	9°0′	3.8750	.8268	1/32	1/32
559T-552A	2.5000	1.00	4.8750	1.5000	1/32	1/8
560T-552A	2.6250	1.50	4.8750	1 5000	9/64	1/8
399T-394A	2.6463	9°0′	4.3307	.8661	1/32	3/64
485T-472A	2.8447	1.00	4.7244	1.1418	9/64	1/8
489T-472A	2.8773	9°0′	4.7244	1.1418	1/32	1/8
755T-752	3.0000	1.50	6 3750	1.8750	9/64	1/8
5784T-5735	3.1875	1 00	5 3438	1 7500	9/64	1/8
758T-752	3.3750	1.00	6.3750	1.8750	9/64	1/8
764T-752	3.6250	1 00	6.3750	1.8750	9/64	1/8
52398T-52618	3.9838	9°0′	6.1875	1.4375	1/32	1/8
780T-772	4 0000	1.50	7.1250	1 8750	9/64	1/8
74551T-74850	5 5000	1 00	8 5000	1.8750	1/16	1/8
74563T-74850	5.6250	1.00	8.5000	1.8750	9/64	1/8
EE217061T-217114	6.0000	1.00	11.4375	2 9375	1/4	1/4
99619T-99100	6.1875	1.00	10 0000	2 6250	1/16	1/8
94719T-94113	7.1875	1.00	11 3750	2 5000	1/16	1/8
EE470077T-470133	7.3125	1.00	13.3750	3 5625	1/8	1/4
EE122083T-122125	8.2500	1.00	12.5000	2.6250	1/8	1/8
EE420826T-421450	8 2500	1.00	14 5000	3.6250	1/8	1/8
88919T-88128	9.1875	1.00	12 8750	2 0625	1/16	1/8

387A-382B	2.2500	3.8125	4.0000	.3125	.8268	9/64	1/32
390-394AB	2.2500	4.3307	4.5187	.3125	.8661	3/32	3/64
39250-39412B	2.5000	4.1250	4 3125	.4063	.8438	.080	.080
29585-29520B	2.5000	4.2500	4.4062	.4063	1.0000	9/64	1/8
3982-3920B	2.5000	4.4375	4.6250	.4375	1.1875	9/64	1/8
559-552B	2.5000	4.8750	5.1250	.5625	1.5000	9/64	1/8
6382-6320B	2.5000	5.3447	5.6567	.6875	2.1250	11/64	1/8
639-632B	2.5000	5.3750	5.6560	.6563	1.6250	9/64	1/8
480-472B	2.6875	4.7244	4.9424	.4375	1.1730	9/64	1/64
566-563B	2.7500	5.0000	5.2500	.5625	1.4375	9/64	1/8
6484-6420B	2.7500	5.8750	6.1875	.6875	2.1250	1/4	1/8
29685-29620B	2.8750	4.4375	4.5940	.4063	1.0000	9/64	1/8
34301-34481B	3.0000	4.8125	5.0000	.3125	.9688	9/64	.080
575-572B	3.0000	5.5115	5.7615	.5625	1.4375	9/64	1/8
7488-742B	3.0000	5.9090	6.2210	.6250	1.7500	9/64	1/8
659-652B	3.0000	6.0000	6 2812	.6563	1.6250	9/64	1/8
47681-47620B	3.1875	5.2500	5.4690	.5000	1.3125	9/64	1/8
496-493B	3.1875	5.3750	5.5930	.5313	1.1875	9/64	1/8
757-752B	3 2500	6.3750	6.6870	.6875	1.8750	9/64	1/8
842-832B	3.2500	6.6250	7.0000	.8750	2.1250	9/64	1/8
598-592B	3.6250	6.0000	6.2500	.6250	1.5625	9/64	1/8
683-672B	3.7500	6.6250	6.9070	.7188	1.6250	9/64	1/8
77375-77675B	3.7500	6.7500	7.0625	.6875	1.8750	9/64	1/8
52387-52637B	3.8750	6.3750	6.6250	.6563	1.4375	9/64	1/8
780-772B	4.0000	7.1250	7.4370	.6875	1.8750	9/64	1/8
861-854B	4.0000	7.5000	7.8750	.8750	2.2500	5/16	1/8
56418-56650B	4.1875	6.5000	6.7500	.6250	1.4375	9/64	1/8
936-932B	4.2500	8.3750	8.8125	.9375	2.6250	5/16	1/8
64450-64700B	4.5000	7.0000	7.2812	.7188	1.6250	9/64	1/8
71450-71750B	4.5000	7.5000	7.8120	.8125	1.8750	9/64	1/8
795-792B	4.7500	8.1250	8.4376	.8125	1.8750	1/8	1/8
67388-67320B	5.0000	8.0000	8.2812	.5938	1.8125	9/64	1/8
74500-74850B	5.0000	8.5000	8.8125	.8125	1.8750	9/64	1/8
95500-95925B	5.0000	9.2500	9.6875	1.0000	2.5000	1/4	1/8
NA48390-48320B	5.3750	7.5000	7.7188	.4688	1.5625	9/64	1/8
898-892B	5.5000	9.0000	9.3750	.8750	2.2500	9/64	1/8
82562-82950B	5.6250	9.5000	9.8750	.8750	2.2500	9/64	1/8
99575-99100B	5.7500	10.0000	10.4360	1.1875	2.6250	9/32	1/8
94649-94113B	6.5000	11.3750	11.8125	1.0625	2.5000	9/32	1/8
NA67787-67720B	6.8750	9.7500	10.0312	.6563	1.8750	9/64	1/8
EE470073-470132B	7.0900	13.2500	13.9376	1.7500	3.5625	17/32	1/4
87750-87111B	7.5000	11.1250	11.5000	.9375	2.0000	9/64	1/8
93750-93125B	7.5000	12.5000	12.9380	1.1250	2.5000	11/64	1/8
67985-67920B	8.1250	11.1250	11.4062	.6563	1 8125	9/64	1/8
96900-96140B	9.0000	14 0000	14.4380	1.2500	2.6875	9/32	1/8
EE8575-8520B	9.2500	12.8750	13 2500	1 0000	2.0625	1/4	1/8
EE125098-125145B	9.5000	14.5000	15.0000	1.0625	2.6875	1/4	1/8

Table 15-4. Dimensions of Wound Roller Bearings

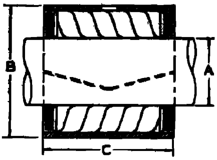


Inner race inside diam, A		Roller assembly inside diam, B, in.	Outer race outside diam, C		Bearing width, D, in.	Roller diam, E, in.	Corner chamfer		Outer race hole diam, H, in.
Mm	In.		Mm	In.			Inner race, F, in.	Outer race, G, in.	
				Light series					
30	1.1811	1 1/2	62	2.4409	13/16	5/16	0.062	0.062	1/4
30	1.1811	1 1/2	62	2.4409	1 1/8	5/16	0.062	0.062	1/4
35	1.3780	1 3/4	72	2.8346	15/16	3/8	0.062	0.062	1/4
35	1.3780	1 3/4	72	2.8346	1 9/16	3/8	0.062	0.062	1/4
40	1.5748	2	80	3.1496	1	3/8	0.078	0.078	1/4
40	1.5748	2	80	3.1496	1 3/8	3/8	0.078	0.078	1/4
45	1.7717	2 3/16	85	3.3465	1 1/8	3/8	0.078	0.078	1/4
45	1.7717	2 3/16	85	3.3465	1 9/16	3/8	0.078	0.078	1/4
50	1.9685	2 3/8	90	3.5433	1 1/4	3/8	0.078	0.078	5/16
50	1.9685	2 3/8	90	3.5433	1 3/4	3/8	0.078	0.078	5/16
50	1.9685	2 3/8	90	3.5433	3 1/2	3/8	0.078	0.078	5/16
55	2.1654	2 5/8	100	3.9370	1 5/16	7/16	0.094	0.078	5/16
55	2.1654	2 5/8	100	3.9370	1 13/16	7/16	0.094	0.078	5/16
55	2.1654	2 5/8	100	3.9370	3 5/8	7/16	0.094	0.078	5/16
60	2.3622	2 7/8	110	4.3307	1 7/16	1/2	0.094	0.094	5/16
60	2.3622	2 7/8	110	4.3307	1 15/16	1/2	0.094	0.094	5/16
60	2.3622	2 7/8	110	4.3307	3 7/8	1/2	0.094	0.094	5/16
65	2.5591	3 1/8	120	4.7244	1 1/2	1/2	0.109	0.109	3/8
65	2.5591	3 1/8	120	4.7244	2 1/16	1/2	0.109	0.109	3/8
65	2.5591	3 1/8	120	4.7244	4 1/8	1/2	0.109	0.109	3/8
70	2.7559	3 5/16	125	4.9213	1 5/8	1/2	0.109	0.109	3/8
70	2.7559	3 5/16	125	4.9213	2 3/8	1/2	0.109	0.109	3/8
70	2.7559	3 5/16	125	4.9213	4 3/4	1/2	0.109	0.109	3/8
75	2.9528	3 1/2	130	5.1181	1 3/4	1/2	0.109	0.109	3/8
75	2.9528	3 1/2	130	5.1181	2 5/8	1/2	0.109	0.109	3/8
75	2.9528	3 1/2	130	5.1181	5 1/4	1/2	0.109	0.109	3/8
80	3.1496	3 3/4	140	5.5118	1 13/16	9/16	0.125	0.125	7/16
80	3.1496	3 3/4	140	5.5118	2 5/8	9/16	0.125	0.125	7/16
80	3.1496	3 3/4	140	5.5118	5 1/4	9/16	0.125	0.125	7/16
85	3.3465	4	150	5.9055	1 15/16	5/8	0.125	0.125	7/16
85	3.3465	4	150	5.9055	2 3/4	5/8	0.125	0.125	7/16
85	3.3465	4	150	5.9055	5 1/2	5/8	0.125	0.125	7/16
90	3.5433	4 1/4	160	6.2992	2 1/16	11/16	0.125	0.125	7/16
90	3.5433	4 1/4	160	6.2992	2 13/16	11/16	0.125	0.125	7/16
90	3.5433	4 1/4	160	6.2992	5 5/8	11/16	0.125	0.125	7/16

Table 15-4. Dimensions of Wound Roller Bearings (*Cont.*)

Inner race inside diam, *A*		Roller assembly inside diam, *B*, in.	Outer race outside diam, *C*		Bearing width, *D*, in.	Roller diam, *E*, in.	Corner chamfer		Outer race hole diam, *H*, in.
Mm	In.		Mm	In.			Inner race, *F*, in.	Outer race, *G*, in.	
Light series (*Cont.*)									
95	3.7402	$4\frac{1}{2}$	170	6.6929	$2\frac{3}{16}$	$\frac{3}{4}$	0.156	0.125	$\frac{7}{16}$
95	3.7402	$4\frac{1}{2}$	170	6.6929	3	$\frac{3}{4}$	0.156	0.125	$\frac{7}{16}$
95	3 7402	$4\frac{1}{2}$	170	6.6929	6	$\frac{3}{4}$	0.156	0.125	$\frac{7}{16}$
100	3.9370	$4\frac{3}{4}$	180	7.0866	$2\frac{5}{16}$	$\frac{3}{4}$	0.156	0.156	$\frac{9}{16}$
100	3.9370	$4\frac{3}{4}$	180	7.0866	$3\frac{1}{4}$	$\frac{3}{4}$	0.156	0.156	$\frac{9}{16}$
100	3.9370	$4\frac{3}{4}$	180	7.0866	$6\frac{1}{2}$	$\frac{3}{4}$	0.156	0.156	$\frac{9}{16}$
110	4 3307	$5\frac{1}{4}$	200	7.8740	$2\frac{9}{16}$	$\frac{7}{8}$	0.156	0.156	$\frac{9}{16}$
110	4.3307	$5\frac{1}{4}$	200	7.8740	$3\frac{1}{2}$	$\frac{7}{8}$	0.156	0.156	$\frac{9}{16}$
110	4.3307	$5\frac{1}{4}$	200	7.8740	7	$\frac{7}{8}$	0.156	0.156	$\frac{9}{16}$
120	4.7244	$5\frac{5}{8}$	215	8 4646	$2\frac{13}{16}$	$\frac{15}{16}$	0.156	0.187	$\frac{9}{16}$
120	4.7244	$5\frac{5}{8}$	215	8.4646	$3\frac{7}{8}$	$\frac{15}{16}$	0.156	0.187	$\frac{9}{16}$
120	4.7244	$5\frac{5}{8}$	215	8.4646	$7\frac{3}{4}$	$\frac{15}{16}$	0.156	0.187	$\frac{9}{16}$
130	5 1181	$6\frac{1}{16}$	230	9.0551	$3\frac{1}{8}$	1	0.156	0.187	$\frac{9}{16}$
130	5.1181	$6\frac{1}{16}$	230	9.0551	$4\frac{1}{4}$	1	0.156	0.187	$\frac{9}{16}$
130	5.1181	$6\frac{1}{16}$	230	9.0551	$8\frac{1}{2}$	1	0.156	0.187	$\frac{9}{16}$
130	5 1181	6	215	8.4646	$4\frac{7}{8}$	$\frac{3}{4}$	0 156	0.187	$\frac{9}{16}$
130	5 1181	6	215	8.4646	$9\frac{3}{4}$	$\frac{3}{4}$	0.156	0.187	$\frac{9}{16}$
140	5 5118	$6\frac{5}{8}$	250	9.8425	$3\frac{1}{4}$	$1\frac{1}{16}$	0 219	0.219	$\frac{9}{16}$
140	5 5118	$6\frac{5}{8}$	250	9.8425	$4\frac{3}{4}$	$1\frac{1}{16}$	0 219	0.219	$\frac{9}{16}$
140	5 5118	$6\frac{5}{8}$	250	9.8425	$9\frac{1}{2}$	$1\frac{1}{16}$	0.219	0 219	$\frac{9}{16}$
150	5 9055	$7\frac{1}{16}$	270	10.6299	$3\frac{1}{2}$	$1\frac{3}{16}$	0.219	0 219	$\frac{5}{8}$
150	5.9055	$7\frac{1}{16}$	270	10.6299	$4\frac{3}{4}$	$1\frac{5}{16}$	0 219	0.219	$\frac{5}{8}$
160	6.2992	$7\frac{5}{8}$	290	11.4173	$3\frac{7}{8}$	$1\frac{1}{4}$	0.250	0.250	$\frac{5}{8}$
160	6 2992	$7\frac{5}{8}$	290	11.4173	$4\frac{7}{8}$	$1\frac{1}{4}$	0.250	0.250	$\frac{5}{8}$
160	6.2992	$7\frac{5}{8}$	290	11.4173	$9\frac{3}{4}$	$1\frac{1}{4}$	0.250	0.250	$\frac{5}{8}$
170	6 6929	$8\frac{1}{16}$	310	12.2047	$5\frac{3}{8}$	$1\frac{3}{8}$	0.250	0 250	$\frac{11}{16}$
170	6.6929	$8\frac{1}{16}$	310	12.2047	$10\frac{3}{4}$	$1\frac{3}{8}$	0.250	0.250	$\frac{11}{16}$
180	7.0866	$8\frac{15}{32}$	320	12.5984	$5\frac{7}{8}$	$1\frac{3}{8}$	0.250	0.250	$\frac{11}{16}$
200	7.8740	$9\frac{1}{4}$	340	13.3858	$4\frac{3}{4}$	$1\frac{3}{8}$	0.250	0.250	$\frac{11}{16}$
200	7.8740	$9\frac{1}{4}$	340	13.3858	$6\frac{7}{8}$	$1\frac{3}{8}$	0.250	0 250	$\frac{11}{16}$
220	8.6614	$10\frac{7}{16}$	380	14 9606	$6\frac{7}{8}$	$1\frac{3}{8}$	0.250	0 250	$\frac{11}{16}$
Medium series									
25	0.9843	$1\frac{1}{4}$	62	2.4409	$1\frac{1}{8}$	$\frac{7}{16}$	0 062	0.062	$\frac{1}{4}$
30	1.1811	$1\frac{1}{2}$	72	2.8346	$1\frac{3}{16}$	$\frac{1}{2}$	0 062	0 062	$\frac{1}{4}$
30	1.1811	$1\frac{1}{2}$	72	2 8346	$\frac{15}{16}$	$\frac{1}{2}$	0 062	0.062	$\frac{1}{4}$
35	1.3780	$1\frac{3}{4}$	80	3.1496	$1\frac{3}{8}$	$\frac{1}{2}$	0.062	0 078	$\frac{1}{4}$
40	1.5748	2	90	3 5433	$1\frac{7}{16}$	$\frac{9}{16}$	0.078	0.078	$\frac{5}{16}$
45	1 7717	$2\frac{1}{4}$	100	3.9370	$1\frac{9}{16}$	$\frac{5}{8}$	0.094	0 078	$\frac{5}{16}$
50	1.9685	$2\frac{1}{2}$	110	4.3307	$1\frac{3}{4}$	$\frac{11}{16}$	0.094	0.094	$\frac{5}{16}$
55	2 1654	$2\frac{3}{4}$	120	4 7244	$1\frac{15}{16}$	$\frac{11}{16}$	0.109	0.109	$\frac{3}{8}$
60	2 3622	3	130	5.1181	$2\frac{1}{8}$	$\frac{3}{4}$	0.125	0 109	$\frac{3}{8}$
65	2.5591	$3\frac{1}{4}$	140	5.5118	$2\frac{5}{16}$	$\frac{13}{16}$	0.125	0.125	$\frac{7}{16}$
70	2.7559	$3\frac{1}{2}$	150	5.9055	$2\frac{1}{2}$	$\frac{7}{8}$	0.125	0.125	$\frac{7}{16}$
75	2.9528	$3\frac{3}{4}$	160	6.2992	$2\frac{11}{16}$	$\frac{15}{16}$	0.156	0.125	$\frac{7}{16}$
80	3.1496	4	170	6.6929	$2\frac{11}{16}$	1	0.156	0.125	$\frac{7}{16}$
85	3.3465	$4\frac{1}{4}$	180	7.0866	$2\frac{7}{8}$	1	0.156	0.156	$\frac{9}{16}$
90	3.5433	$4\frac{1}{2}$	190	7.4803	3	$1\frac{1}{16}$	0.187	0.156	$\frac{9}{16}$
95	3.7402	$4\frac{3}{4}$	200	7.8740	$3\frac{1}{16}$	$1\frac{1}{8}$	0.187	0.156	$\frac{9}{16}$
100	3.9370	5	215	8.4646	$3\frac{1}{4}$	$1\frac{1}{4}$	0.187	0.187	$\frac{9}{16}$
110	4.3307	$5\frac{1}{2}$	240	9.4488	$3\frac{5}{8}$	$1\frac{3}{8}$	0.219	0.219	$\frac{9}{16}$
120	4.7244	$6\frac{1}{16}$	260	10.2362	$4\frac{1}{8}$	$1\frac{3}{8}$	0.250	0.250	$\frac{9}{16}$
130	5.1181	$6\frac{9}{16}$	280	11.0236	$4\frac{3}{8}$	$1\frac{1}{2}$	0.250	0.250	$\frac{9}{16}$

Table 15-5. Dimensions of Wound Roller Split-race Bearings

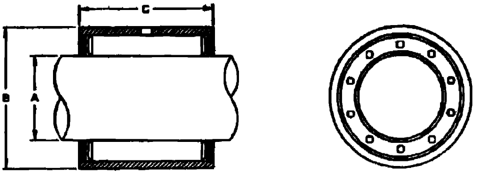


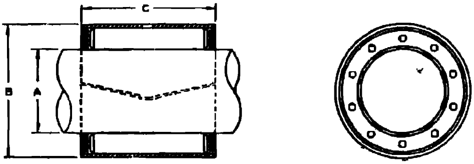
Shaft diam, *A*, in.	Outside diam of bearing, *B*, in.	Bearing width, *C*, in.	Shaft diameter limits		Housing bore limits		Rollers	
			High limit, in.	Low limit, in.	High limit, in.	Low limit, in.	No.	Diam
1	1 15/16	1 1½ 2 2½ 3 4	1.000	0.997	1.940	1.938	7	⅜
1⅛	2 3/16	2 2½ 3 4	1.125	1.122	2.190	2.188	8	7/16
1 3/16	2⅜	2 2½ 3 4	1.188	1.185	2.377	2.375	7	½
1¼	2 7/16	2 2½ 3 4	1.250	1.247	2.440	2.438	8	½
1⅜	2 9/16	2 2½ 3 4	1.375	1.372	2.565	2.563	8	½
1 7/16	2¾	2 2½ 3 4 5	1.438	1.435	2.752	2.750	8	9/16
1½	2 13/16	2 2½ 3 4 5	1.500	1.497	2.815	2.813	8	9/16
1⅝	2 15/16	2 2½ 3 4	1.625	1.622	2.940	2.938	8	9/16
1 11/16	3	2 3 4	1.688	1.685	3.003	3.000	9	9/16
1¾	3 1/16	2 4 5	1.750	1.747	3.066	3.063	9	9/16

Table 15-5. Dimensions of Wound Roller Split-race Bearings (*Cont.*)

Shaft diam, *A*, in.	Outside diam of bearing, *B*, in.	Bearing width, *C*, in.	Shaft diameter limits		Housing bore limits		Rollers	
			High limit, in.	Low limit, in.	High limit, in.	Low limit, in.	No.	Diam
1 7/8	3 3/16	4 5	1.875	1.872	3.191	3.188	10	9/16
1 15/16	3 7/16	2 3 4 5 6	1.938	1.935	3.441	3.438	10	5/8
2	3 1/2	3 4 5	2.000	1.997	3.503	3.500	9	5/8
2 3/16	4 3/16	3 5	2.188	2.185	4.191	4.188	8	7/8
2 1/4	4 1/4	3 4 5 6	2.250	2.247	4.253	4.250	8	7/8
2 7/16	4 9/16	5	2.438	2.435	4.566	4.563	8	15/16
2 1/2	4 5/8	3 5 6 7	2.500	2.497	4.628	4.625	8	15/16
2 3/4	4 7/8	4 5 6	2.750	2.747	4.878	4.875	9	15/16
3	5 1/4	4 5 6 7	3.000	2.996	5.254	5.250	9	1
3 1/2	6 1/8	4 5 7	3.500	3.496	6.129	6.125	12	1 1/8
4	6 5/8	4 5 6 7	4.000	3.996	6.629	6.625	14	1 1/8
4 1/2	7 3/8	4 5 6 7	4.500	4.496	7.380	7.375	14	1 1/4
5	7 7/8	4 5 6 7	5.000	4.995	7.880	7.875	15	1 1/4
6	8 7/8	5 6 7	6.000	5.995	8.880	8.875	18	1 1/4
7	10 1/8	7	7 000	6.995	10.130	10 125	18	1 3/8

Table 15-6. Dimensions of Solid Roller Bearings





Shaft diam, *A*	Outside diam, *B*	Width, *C*	Roller diam
Split races			
1/2	15/16	1 1/2	3/16
1/2	15/16	1 3/4	3/16
1/2	1 1/16	1	1/4
1/2	1 1/16	1 1/2	1/4
1/2	1 1/16	1 3/4	1/4
1/2	1 1/16	2	1/4
5/8	1 1/16	3/4	3/16
5/8	1 1/16	1	3/16
5/8	1 1/16	1 1/4	3/16
5/8	1 1/16	1 1/2	3/16
5/8	1 1/16	1 3/4	3/16
5/8	1 1/16	2	3/16
5/8	1 3/16	1	1/4
5/8	1 3/16	1 1/4	1/4
5/8	1 3/16	1 1/2	1/4
5/8	1 3/16	1 3/4	1/4
5/8	1 3/16	2	1/4
3/4	1 3/16	3/4	3/16
3/4	1 3/16	1	3/16
3/4	1 3/16	1 1/4	3/16
3/4	1 3/16	1 1/2	3/16
3/4	1 3/16	1 3/4	3/16
3/4	1 3/16	2	3/16
3/4	1 5/16	1 1/2	1/4
3/4	1 5/16	1 3/4	1/4
3/4	1 5/16	2	1/4
3/4	1 5/16	2 1/2	1/4
3/4	1 7/16	1	5/16
3/4	1 7/16	1 1/2	5/16
3/4	1 7/16	2	5/16
3/4	1 7/16	2 1/2	5/16
7/8	1 5/16	1 1/2	3/16
7/8	1 5/16	1 3/4	3/16
7/8	1 5/16	2	3/16
7/8	1 7/16	1	1/4
7/8	1 7/16	1 1/2	1/4
7/8	1 7/16	1 3/4	1/4
7/8	1 7/16	2	1/4
7/8	1 7/16	2 1/4	1/4
7/8	1 7/16	2 1/2	1/4
7/8	1 9/16	1	5/16
7/8	1 9/16	1 1/2	5/16
7/8	1 9/16	2	5/16
7/8	1 9/16	2 1/2	5/16
1	1 7/16	1	3/16
1	1 7/16	1 1/2	3/16
1	1 7/16	1 3/4	3/16
1	1 7/16	2	3/16
1	1 7/16	2 1/2	3/16
1	1 9/16	1	1/4
1	1 9/16	1 1/8	1/4
1	1 9/16	1 1/2	1/4
1	1 9/16	2	1/4
1	1 9/16	2 1/2	1/4
1	1 3/4	2	5/16
1	1 3/4	2 1/2	5/16
1 1/8	1 9/16	1	3/16
1 1/8	1 9/16	1 1/2	3/16
1 1/8	1 9/16	2	3/16
1 1/8	1 3/4	1 3/4	1/4
1 1/8	1 3/4	2	1/4
1 1/8	1 3/4	2 1/2	1/4
1 1/8	1 7/8	2	5/16
1 1/4	1 3/4	1	3/16
1 1/4	1 3/4	1 3/4	3/16
1 1/4	1 7/8	2	1/4
1 1/4	1 7/8	2 1/2	1/4
1 1/4	2	1 1/2	5/16
1 1/4	2	2	5/16
1 1/4	2	2 1/2	5/16
1 3/8	2 1/8	2 1/2	5/16
1 1/2	2 1/4	2	5/16
1 1/2	2 1/4	2 1/2	5/16
Solid races			
1/2	1 3/16	1	1/4
5/8	1 3/16	1	3/16
5/8	1 5/16	1	1/4
5/8	1 5/16	1 1/2	1/4
3/4	1 5/16	7/8	3/16
3/4	1 5/16	1	3/16
3/4	1 5/16	1 1/2	3/16
3/4	1 9/16	1	5/16
3/4	1 9/16	1 1/2	5/16
3/4	1 9/16	2	5/16
3/4	1 9/16	2 1/2	5/16
7/8	1 9/16	1	1/4
7/8	1 9/16	1 1/2	1/4
7/8	1 9/16	2	1/4
7/8	1 9/16	2 1/2	1/4
1	1 9/16	1	3/16
1	1 9/16	1 1/2	3/16
1	1 9/16	2	3/16
1 1/8	1 7/8	1	1/4
1 1/4	1 7/8	1	3/16
1 1/4	2	1 5/8	1/4
1 1/4	2	2 1/2	1/4
1 3/8	2 1/8	1 5/8	1/4
1 1/2	2 1/4	1 1/4	1/4
1 1/2	2 3/8	1 3/4	5/16

Note: Any roller assembly from the tables above may be used without the outer race.

Table 15-7*a*. Ball Thrust Bearings*

Bearing bore, *d*		ISA Diameter series 1					SAE light					SAE medium				
		Outside diam, *D*		Max fillet radius, in.	Dimension series 11, height, *H*		Outside diam, *D*		Max fillet radius, in.	Height, *H*		Outside diam, *D*		Max fillet radius, in.	Height, *H*	
Mm	In.	Mm	In.		Mm	In.	Mm	In.		Mm	In.	Mm	In.		Mm	In.
15	.5906											35	1.3780	.024	14	.551
18	.7087						35	1.3780	.024	12	.472					
20	.7874						37	1.4567	.024	12	.472	40	1.5748	.024	14	.551
25	.9843						45	1.7717	.024	14	.551	48	1.8898	.024	15.5	.610
30	1.1811						50	1.9685	.024	14	.551	53	2.0866	.024	15.5	.610
35	1.3780	53	2.0866	.024	12	.472	55	2.1654	.024	16	.630	62	2.4409	.024	18	.709
40	1.5748	60	2.3622	.024	13	.512	60	2.3622	.024	16	.630	64	2.5197	.024	18	.709
45	1.7717	65	2.5591	.024	14	.551	68	2.6772	.024	16	.630	73	2.8740	.024	22	.866
50	1.9685	70	2.7559	.024	14	.551	74	2.9134	.024	18	.709	78	3.0709	.024	22	.866
55	2.1654	78	3.0709	.024	16	.630	78	3.0709	.024	18	.709	88	3.4646	.024	24.5	.965
60	2.3622	85	3.3465	.039	17	.669	82	3.2283	.024	18	.709	90	3.5433	.024	24.5	.965
65	2.5591	90	3.5433	.039	18	.709	90	3.5433	.024	20	.787	100	3.9370	.039	27	1.063
70	2.7559	95	3.7402	.039	18	.709	95	3.7402	.024	20	.787	103	4.0551	.039	27	1.063
75	2.9528	100	3.9370	.039	19	.748	100	3.9370	.024	20	.787	110	4.3307	.039	27	1.063
80	3.1496	105	4.1339	.039	19	.748	110	4.3307	.024	22	.866	115	4.5276	.039	29	1.142
85	3.3465	110	4.3307	.039	19	.748	115	4.5276	.024	22	.866	125	4.9213	.039	30.5	1.201
90	3.5433	120	4.7244	.039	22	.866						135	5.3150	.039	30.5	1.201
95	3.7402						130	5.1181	.024	25	.984	140	5.5118	.039	32.5	1.280
100	3.9370	135	5.3150	.039	25	.984						150	5.9055	.039	32.5	1.280
105	4.1339						140	5.5118	.024	25	.984	155	6.1024	.039	40	1.575
110	4.3307	145	5.7087	.039	25	.984										
115	4.5276						150	5.9055	.024	25	.984	165	6.4961	.039	43	1.693
120	4.7244	155	6.1024	.039	25	.984										
125	4.9213											175	6.8898	.039	46.5	1.831
130	5.1181	170	6.6929	.039	30	1.181										
140	5.5118	180	7.0866	.039	31	1.220						200	7.8740	.039	52	2.047

* Outside diameters listed show approximately the sizes in current production.

Table 15-7b. Spherical Roller Thrust Bearings*

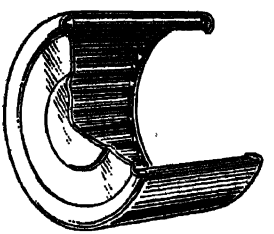
Bearing bore, *d*		Diameter series 3					Diameter series 4				
		Outside diam, *D*		Max fillet radius, in.	Dimension series 93		Outside diam, *D*		Max fillet radius, in.	Dimension series 94	
					Height, *H*					Height, *H*	
Mm	In.	Mm	In.		Mm	In.	Mm	In.		Mm	In.
90	3.5433						190	7.4803	.079	60	2.362
100	3.9370						210	8 2677	.098	67	2.638
110	4.3307	190	7.4803	.079	48	1.890	230	9.0551	.098	73	2.874
120	4.7244						250	9 8425	.118	78	3 071
130	5.1181	225	8.8583	.079	58	2.283	270	10.6299	.118	85	3.346
150	5.9055	250	9.8425	.079	60	2.362	300	11.8110	.118	90	3.543
170	6.6929	280	11.0236	.098	67	2.638	340	13.3858	.157	103	4.055
190	7.4803	320	12.5984	.118	78	3.071	380	14.9606	.157	115	4.528
220	8.6614	360	14.1732	.118	85	3.346	420	16.5354	.197	122	4.803
260	10.2362	420	16.5354	.157	95	3.740	480	18.8976	.197	132	5.197
300	11.8110	480	18.8976	.157	109	4.291	540	21.2598	.197	145	5.709
360	14.1732	560	22 0472	.157	122	4.803					
400	15.7480	620	24.4094	.197	132	5.197					
460	18.1102	710	27.9528	.197	150	5.906					

* Outside diameters listed show approximately the sizes in current production.

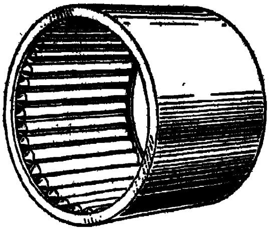
NEEDLE ROLLER BEARINGS

Partial-revolution and oscillating-type, or relatively lightly loaded applications make wide use of the so-called "needle" bearings, with small-diameter rollers and a high ratio of roller length to roller diameter. The term "needle" was originally associated with the needlelike proportions of the roller itself (see Fig. 15-10), and bearings made up with such rollers were called "needle bearings" or "package bearings." These are also available in many types and with and without separable inner or outer rings and with and without separators. They are particularly adaptable where diametral space is limited and the advantages of an antifriction bearing are desired (see Tables 15-2*a*, 15-2*b*, and 15-2*c* for AFBMA Standard Needle Bearing sizes).

A common type of needle bearing consists of the required number of ground rollers enclosed in a hardened shell which may be open at both ends or closed on one end only. Commonly used sizes are given in Table 15-7*c*.



Closed end



Open end

Table 15-7c. Thin-shell Needle Bearings

Nominal shaft diam	Shaft limits		Nominal OD	Housing bore limits		Length +.000 −.010
	Max	Min		Min	Max	
5/32	.1563	.1560	9/32	.2812	.2817	1/4
3/16	.1875	.1872	11/32	.3432	.3437	1/4
3/16	.1875	.1872	11/32	.3432	.3437	3/8
1/4	.2500	.2495	7/16	.4370	.4380	1/4
1/4	.2500	.2495	7/16	.4370	.4380	5/16
1/4	.2500	.2495	7/16	.4370	.4380	7/16
5/16	.3125	.3120	1/2	.4995	.5005	5/16
5/16	.3125	.3120	1/2	.4995	.5005	7/16
5/16	.3125	.3120	1/2	.4995	.5005	9/16
3/8	.3750	.3745	9/16	.5620	.5630	3/8
3/8	.3750	.3745	9/16	.5620	.5630	1/2
3/8	.3750	.3745	9/16	.5620	.5630	5/8
7/16	.4375	.4370	5/8	.6245	.6255	1/2
1/2	.5000	.4995	11/16	.6870	.6880	7/16
1/2	.5000	.4995	11/16	.6870	.6880	1/2
1/2	.5000	.4995	11/16	.6870	.6880	5/8
1/2	.5000	.4995	11/16	.6870	.6880	3/4
9/16	.5625	.5620	3/4	.7495	.7505	5/16
9/16	.5625	.5620	3/4	.7495	.7505	1/2
9/16	.5625	.5620	3/4	.7495	.7505	5/8
9/16	.5625	.5620	3/4	.7495	.7505	3/4
5/8	.6250	.6245	13/16	.8120	.8130	5/16
5/8	.6250	.6245	13/16	.8120	.8130	1/2
5/8	.6250	.6245	13/16	.8120	.8130	3/4
11/16	.6875	.6870	7/8	.8745	.8755	1/2
11/16	.6875	.6870	7/8	.8745	.8755	3/4
3/4	.7500	.7495	1	.9995	1.0005	3/8
3/4	.7500	.7495	1	.9995	1.0005	1/2
3/4	.7500	.7495	1	.9995	1.0005	5/8
3/4	.7500	.7495	1	.9995	1.0005	3/4
13/16	.8125	.8120	1 1/16	1.0620	1.0630	1/2
13/16	.8125	.8120	1 1/16	1.0620	1.0630	7/8
13/16	.8125	.8120	1 1/16	1.0620	1.0630	1
7/8	.8750	.8745	1 1/8	1.1245	1.1255	1/2
7/8	.8750	.8745	1 1/8	1.1245	1.1255	3/4
7/8	.8750	.8745	1 1/8	1.1245	1.1255	1
15/16	.9375	.9370	1 3/16	1.1870	1.1880	1
1	1.0000	.9995	1 1/4	1.2495	1.2505	1/2
1	1.0000	.9995	1 1/4	1.2495	1.2505	3/4
1	1.0000	.9995	1 1/4	1.2495	1.2505	1
1 1/8	1.1250	1.1245	1 3/8	1.3745	1.3755	1/2
1 1/8	1.1250	1.1245	1 3/8	1.3745	1.3755	3/4
1 1/8	1.1250	1.1245	1 3/8	1.3745	1.3755	1
1 1/4	1.2500	1.2495	1 1/2	1.4995	1.5005	1/2
1 1/4	1.2500	1.2495	1 1/2	1.4995	1.5005	3/4
1 1/4	1.2500	1.2495	1 1/2	1.4995	1.5005	1
1 1/4	1.2500	1.2495	1 1/2	1.4995	1.5005	1 1/4
1 3/8	1.3750	1.3745	1 5/8	1.6245	1.6255	1/2
1 3/8	1.3750	1.3745	1 5/8	1.6245	1.6255	3/4
1 3/8	1.3750	1.3745	1 5/8	1.6245	1.6255	1 1/4
1 1/2	1.5000	1.4995	1 7/8	1.8745	1.8755	7/8
1 1/2	1.5000	1.4995	1 7/8	1.8745	1.8755	1
1 1/2	1.5000	1.4995	1 7/8	1.8745	1.8755	1 1/4
1 5/8	1.6250	1.6245	2	1.9995	2.0005	1 1/4
1 3/4	1.7500	1.7495	2 1/8	2.1245	2.1255	1
1 3/4	1.7500	1.7495	2 1/8	2.1245	2.1255	1 1/2
1 7/8	1.8750	1.8745	2 1/4	2.2495	2.2505	1/2
2	2.0000	1.9994	2 3/8	2.3745	2.3755	1 3/4
2 1/8	2.1250	2.1244	2 1/2	2.4995	2.5005	1 1/2
2 5/8	2.6250	2.6244	3	2.9995	3.0005	1
2 3/4	2.7500	2.7494	3 1/8	3.1245	3.1255	1
2 3/4	2.7500	2.7494	3 1/8	3.1245	3.1255	1 1/4

Courtesy of The Torrington Company.

CYLINDRICAL ROLLER BEARINGS
AND
NEEDLE ROLLER BEARINGS

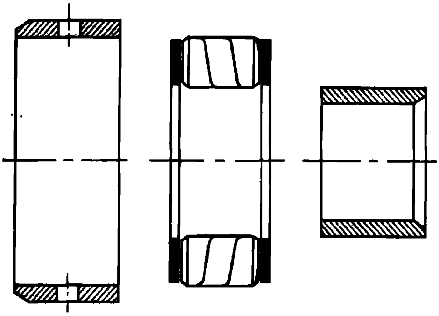
BY JOHN C. MORGAN

Engineer, Hyatt Bearings Division, General Motors Corporation

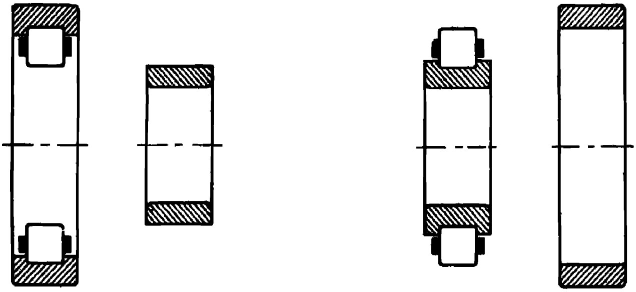
CYLINDRICAL ROLLER BEARINGS

Cylindrical roller bearings are exactly what their name signifies, *i.e.*, antifriction bearings in which the rolling elements are straight cylindrical rollers.

The rollers are usually held together and spaced circumferentially by a cage or separator, and the "roller assembly" thus formed may be entirely free of the inner and outer rings, or it may be assembled with either the inner or outer ring or with both.



Wound-roller separable roller bearing



Solid roller bearing with separable inner ring

Solid roller bearing with separable outer ring

FIG. 15-5a. Types of separable cylindrical roller bearings.

Cylindrical roller bearings are, therefore, classified according to construction, as

1. *Three-part bearings*, in which the roller assembly and both inner and outer ring are separable. This construction permits the use of the roller assembly independently of the rings or with only the inner or outer ring or with both.

2. *Two-part bearings*, in which the inner ring is separable. In this type of construction, the rollers are retained with the outer ring. The use of the inner ring is optional. It is often omitted and the rollers are operated directly upon the surface of a suitably hardened and ground shaft.

3. *Two-part bearings*, in which the outer ring is separable. In this case the rollers are retained with the inner ring. The separable outer ring may be omitted and the rollers run directly upon the surface of the housing, gear, or sheave bore, etc., suitably hardened and ground.

4. *Nonseparable bearings*. In this type both the inner and the outer ring and the rollers are assembled in such a way that the bearing can be handled only as a complete unit (see Figs. 15-5*a* and 15-5*b*).

The rollers in a nonseparable bearing may be spaced by a cage or separator, or the bearing may contain a full complement of rollers, *i.e.*, the greatest number of rollers that can be inserted between the rings with the cage or separator omitted. Usually the proportions of the rings and rollers in a full-complement bearing are such that there is very little circumferential clearance between the rollers (see Fig. 15-5*c*).

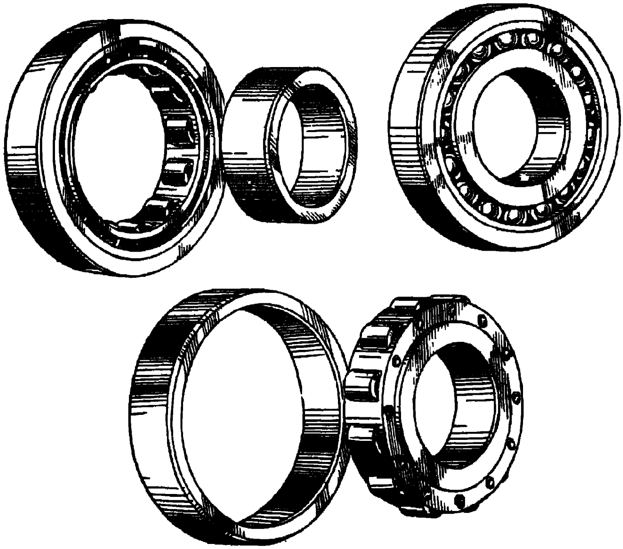


FIG 15-5*b*. Cylindrical roller bearings with separable inner and separable outer ring. Also nonseparable type with full complement of rollers.

Functionally, roller bearings may be classified as "nonlocating," "one-direction locating," and "two-direction locating." This defines their ability to locate or restrain the movement of shafts or other moving parts laterally within certain allowable limits.

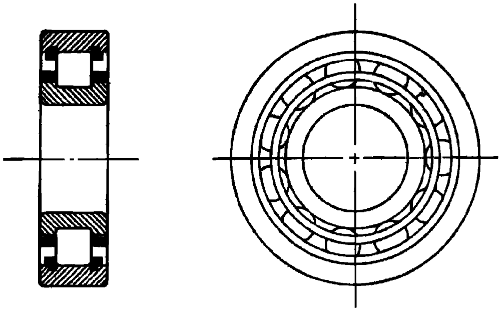
Normally cylindrical roller bearings are used for carrying radial loads only, hence the rings are plain cylinders. Adding shoulders or flanges to either or both rings increases the usefulness of the bearing, however, in that it can then be used to locate shafts or gears and even to sustain light intermittent thrust loads (see Fig. 15-6).

The great majority of applications are fitted with the so-called AFBMA standard metric-dimension bearings. These are so proportioned that the roller diameter is approximately half the bearing-section height, the other half being approximately equally divided between the inner and outer ring. The roller length is usually about the same as its diameter, hence the inappropriate but widely used term "square roller."

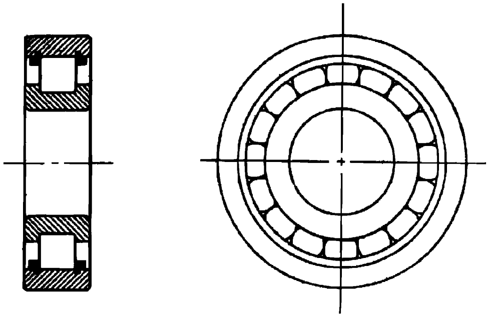
These bearings are made in several series with a wide range of sizes as shown in Table 15-1. The pattern of sizes is so arranged that several bearings, varying in width and outside diameter, are available for every given bore (see Fig. 15-7). A variety of types (Fig. 15-6) is also available in any given size to suit the requirements of any application.

Any given size may be obtained in a standard commercial grade or in a so-called "precision" grade. The former conforms with AFBMA Roller Bearing Engineers' Committee Tolerance Standard RBEC-1 and the latter with Tolerance Standard RBEC-5. The RBEC-5 grade bearings are made to closer tolerances not only on the bore and outside diameter but also on the dimensions that govern running accuracy, *i.e.*, eccentricity of the ring diameters and squareness of ring faces with ring diameters.

The running smoothness of a bearing application depends on the uniformity of the rings and rollers, the character of the finish on the operating surfaces, and on the internal diametral clearance which exists in the assembled bearing. Because of the difficulty of controlling this clearance owing to the dimensional variations of the bearing component parts and the effect of interference fits on the rings, various means have been designed to overcome these difficulties, especially for applications where internal clearance must definitely be controlled or held to a minimum.



Separator type



Full-complement type

Fig. 15-5c. Separator and full-complement-type roller bearings.

The double-row cylindrical roller bearing has been designed specifically for this purpose and is manufactured with a tapered bore (1:12) which allows the bearing to be driven up on the tapered shaft to give any desired internal clearance in the bearing. This type of bearing has the low-friction and high-limiting-speed characteristics of cylindrical roller bearings and great rigidity owing to its double-row width. It is usually made to extra-precision tolerances for machine-tool spindle and similar applications. Double-row cylindrical roller bearings with tapered bores are made with straight cylindrical outer races and can carry only radial loads (see Fig. 15-8).

A popular type of commercial bearing is the so-called "journal bearing." This is almost invariably of three-part construction, with a separable roller assembly and separable straight cylindrical outer and inner rings. These are made up with solid

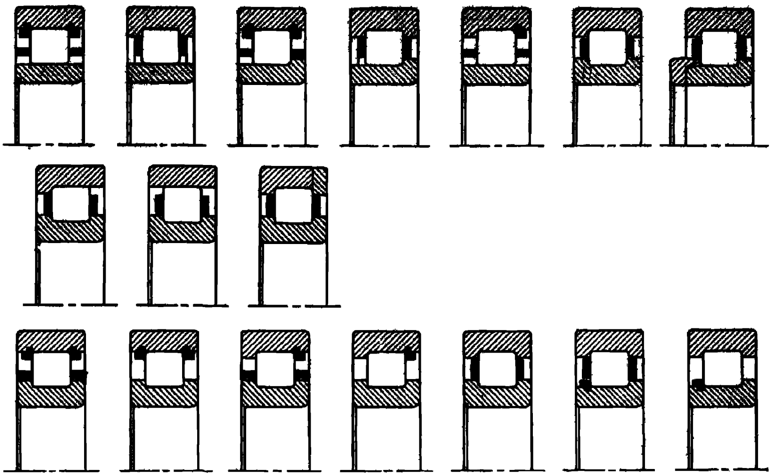


FIG 15-6. Variations in detail design of basic cylindrical roller bearings

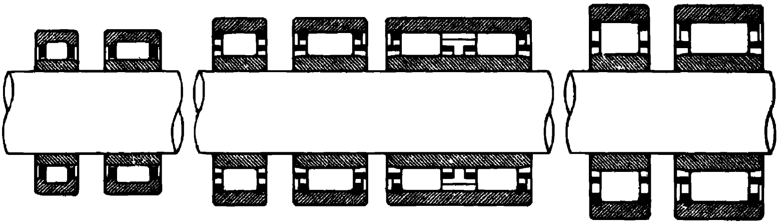


FIG. 15-7. Dimension variation in cylindrical roller bearings.

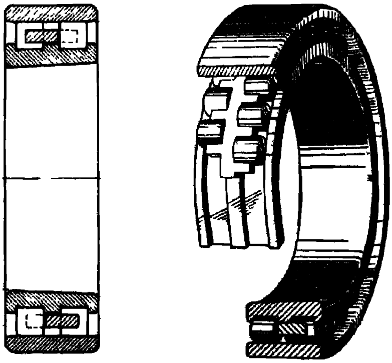


FIG. 15-8. Double-row cylindrical roller bearing with tapered bore.

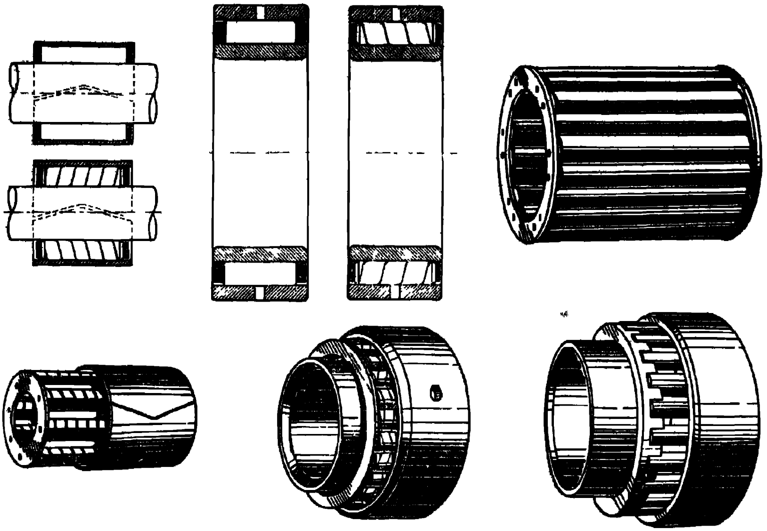


FIG. 15-9. Journal bearings.

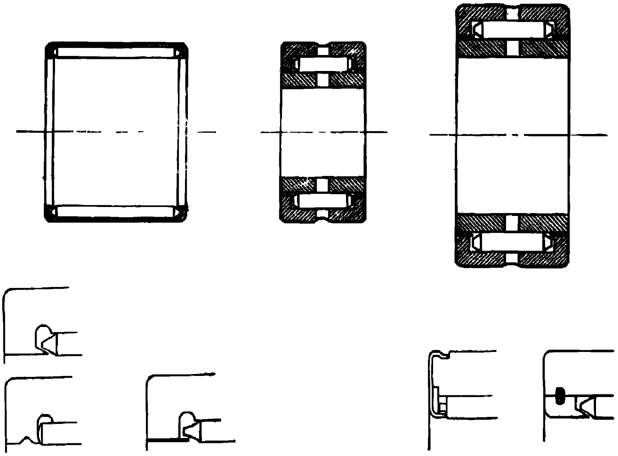


FIG. 15-10. Needle bearings. Different methods of retaining roller ends.

rollers, with "wound" rollers, or with hollow solid rollers. They are often applied with only the outer ring or only with the inner ring, and sometimes the roller assembly is assembled alone, operating directly on the shaft and bore (see Fig. 15-9).

Journal bearings are commonly made to fractional dimensions, although there are a number of standard designs manufactured to metric dimensions on bore and outside diameter, only the widths being in inches.

On a much lower plane of quality and accuracy are the so-called "unground" cylindrical roller bearings, in which the rings and rollers are neither hardened nor finish ground.

THE TAPERED ROLLER BEARING

BY PETER C. POSS

Engineering Editor, The Timken Roller Bearing Company

FUNDAMENTAL PRINCIPLES OF DESIGN

Like many other noteworthy developments in the field of mechanics, the principle of the tapered roller bearing is remarkably simple. Essentially it consists of constructing the rolling elements, as well as the raceways, on the principle of the cone, so that lines drawn coincident with the working surfaces of the rollers and races will all meet at a common point on the axis of the bearing. This is brought out clearly in Fig. 15-11. The significance of this is at once apparent. Initially, true rolling motion is obtained, and secondly, the bearing will handle all loads, radial, thrust, or both, in any combination. Furthermore, no greater number of parts are necessary than in other types of antifriction bearings; *i.e.*, an inner raceway, an outer raceway, the tapered rollers, and the cage or roller retainer. These are designated as the cone, which it resembles from its mechanical profile construction; the cup, which fits over the cone and rollers; the rollers, which roll between the cone and cup; and the cage, which retains the rollers properly spaced about the cone. These are all shown in Fig. 15-12.

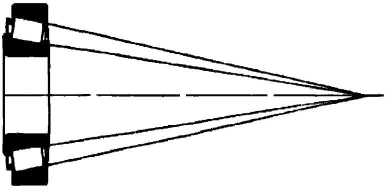


FIG. 15-11. Principle of the tapered roller bearing.

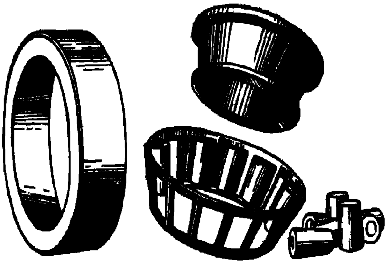


FIG. 15-12. Parts of a tapered bearing: cage, rollers, cone, and cup.

Even in its early stages, this new development in the antifriction bearing field, possessing obvious advantages over other types, met with immediate favor. Manufacturers recognized its merits and soon found occasion to include it in new products, particularly wagons and buggies. That the design is correct is evidenced today by the fact that, although its use has spread to practically every field of industry, it has never been found necessary to depart from the basic principles. Refinements have been made, but the tapered bearing today bears a remarkably close resemblance to the one produced over 50 years ago.

TYPES OF TAPERED BEARINGS

From the foregoing description it will be appreciated that the design of the tapered bearing is such that it is suited particularly well to most positions in machines and will carry all loads regardless of whether they are all radial, all thrust, or in any combination. Even in positions where the load is momentarily changing from one direction to the other, the tapered roller bearing functions perfectly.

There are some locations, however, where certain load or mounting conditions exist which make it desirable to modify somewhat the standard tapered bearing.

These modifications, which do not change the basic principles of the bearing, may be grouped under three natural headings. First, those where the change is made in the raceways or rolling elements, *e.g.*, increasing the angles of the cup, cone, and rollers to obtain greater thrust-carrying capacity; second, by making combinations to secure multiple-row assemblies; finally, bearings developed for certain positions and not suited to general machine applications.

There are occasions also when a designer may wish to adhere to standard dimensions. To meet this condition a light, a medium, and a heavy series of tapered roller bearings have been selected and adopted by the Society of Automotive Engineers as an SAE standard (see Table 15-3).

In the application of the bearings the various factors entering into the operation of the bearing and the machine are considered and the proper tapered bearing is selected. Usually this can be done by using the standard single-row bearing. However, there are occasions when one of the other assemblies is better suited to the design. We are therefore illustrating and describing these various units briefly and indicating the positions where they have found the widest use.

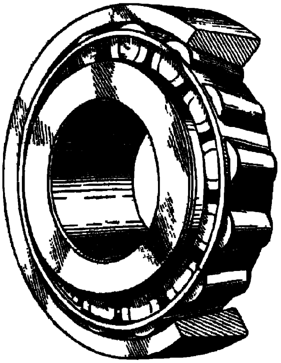


Fig. 15-13. Standard single-row tapered bearing—Type TS.

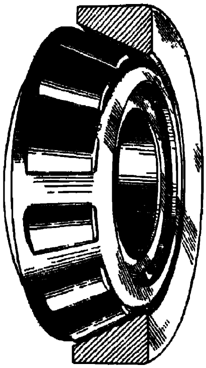


Fig. 15-14. Steep-angle or high-thrust rating bearing—Type TSS.

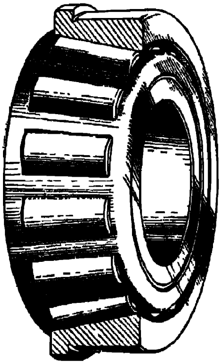


Fig. 15-15. Flanged-cup bearing—Type TSF.

Single-row Bearing—Type ST. In the consideration of the various types of tapered bearings, obviously the standard bearing (Fig. 15-13) is the first to discuss. It is this bearing which is so widely used throughout all industry. The principle of the bearing is understood since the design and construction have been described. It will be recognized, however, that almost endless modifications are possible merely by changing the dimensions or angles of the rolling elements. In actual practice, however, it is essential to make certain basic classifications or "series." These are established by maintaining a fundamental relation between the various components of the bearing. We thus obtain standard or conventional tapered bearings with wide races and long rollers, others with narrow races and short rollers, and a third group representing an approximate mean between the other two. This permits the designer to select the bearing best suited to his needs.

Steep-angle Bearing—Type TSS. In numerous applications, such as power take-offs and worm and bevel-gear shafts, the thrust load predominates, although there is an appreciable radial load. For these positions the high-thrust-capacity or steep-angle bearing has been developed (Fig. 15-14). It differs from the conventional tapered bearing in that the angles of the raceways, as well as the rollers, are quite steep. The fundamental principle of the tapered roller bearing is retained in that lines drawn coincident with the surfaces of the cone, cup, and rollers meet at a common point on the axis of the bearing, thus assuring true rolling motion. The designer therefore obtains all the outstanding advantages of the tapered bearing and in addition a product particularly adapted to heavy thrust loads.

Flanged-cup Bearing—Type TSF. In certain applications, particularly in precision machine-tool spindles, it was found difficult to machine conventional cup seats to a shoulder and maintain the desired accuracy. This was brought about on account of the necessity of removing the housing from the machine after one seat was machined and to set it up in a reverse position to bore the other. Even with the most careful workmanship a variation often occurred so that the cup seats were not in absolute alignment. The required accuracy can now be obtained, however, by boring straight through the housing and facing an outside seat at right angles to the bore. The face thus obtained provides the backing for the flange of the cup (Fig. 15-15).

This bearing is also favored in other machine-tool applications such as gearboxes where there is not sufficient space to use conventional mounting methods. In this case the flange on the cup is set against a face machined on the inside of the gearbox.

In other industrial applications such as sheaves and car wheels, which are frequently cast of manganese steel, it is also used extensively. Manganese steel is exceptionally difficult to machine. In fact, it is necessary to resort to grinding to obtain a finished surface. It would be impractical to attempt to obtain the usual shoulder for cup backing, but here again no difficulty is experienced in grinding a seat on the face of the sheave or wheel hub to locate the flange and thus obtain the required bearing mounting.

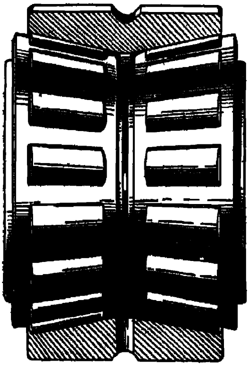


FIG. 15-16. Two-row bearing, double cup, single cones—Type DO.

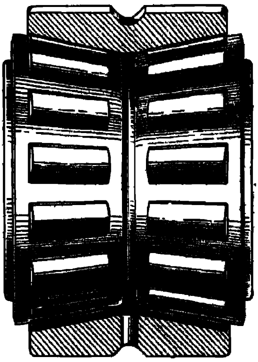


FIG. 15-17. Two-row bearing—Type NA.

The popularity of this bearing has brought about its rapid development so that it is now available in a wide range of sizes.

Multiple-row Bearings. In an ever-widening group of applications, presenting problems not readily solved by the single-row tapered bearing, this type, in one of its many variations, is used extensively. Through the use of these assemblies the designer is able to provide maximum load-carrying capacity in the smallest space, to resist thrust loads from both directions, and to maintain a precise and rigid mounting or a "floating" assembly as desired.

A discussion of multiple-row bearings divides itself naturally into three groups; the two-row cone-adjusted, two-row cup-adjusted, and the four-row cup-adjusted bearings. Within the first two classifications are other types developed to meet specific applications or for certain needs. Thus, in the two-row cone-adjusted assembly will be found the standard two-row cone-adjusted bearing shown in Fig. 15-16 and the "NA" series (Fig. 15-17).

In the two-row cup-adjusted group there has been developed a more extensive range of assemblies. These include the standard two-row cup-adjusted bearing with conventional cone and cups (Fig. 15-19), the tapered-bore bearing, with and without cone puller (Fig. 15-20), and the slotted-cone bearing (Fig. 15-18). At present but one four-row assembly is recommended. That is the four-row cup-adjusted bearing.

Two-row Bearing, Double Cup, Single Cones—Type DO. The two-row bearing, double cup, single cone, employs a double cup with two conventional cones (Fig. 15-16). Two assemblies are possible. Both are identical except that in one case a spacer is introduced between the cones so that the setup or running clearance is established at the time the bearing is assembled. The other, having no cone spacer, requires that the proper setup be obtained by some form of adjusting device. Either form will give satisfactory service. The one having the spacer is usually preferred in those applications where provision must be made for quick removal and replacement of the bearings or where the operating conditions are such that it is unnecessary to provide for take-up after a long period of service.

Numerous applications require that some provision be made to allow for shaft expansion. With this type of bearing it is readily accomplished by securely locking the cones in position and permitting the cup to "float" in the housing. The opposite end of the shaft is usually fixed in position so that the movement is confined to one end.

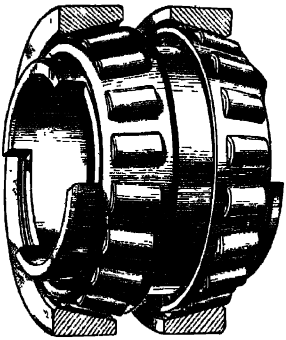


FIG. 15-18. Two-row bearing, slotted double cone—Type DIK.

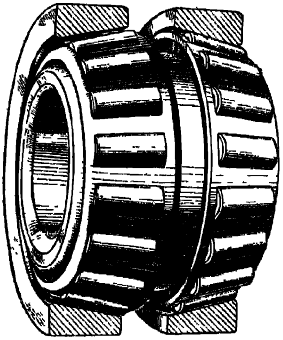


FIG. 15-19. Two-row bearing, double cone, single cups—Type DI.

Two-row Bearing—Type NA. Another construction is shown in Fig. 15-17. Here again we have a double cup and two single cones. In this assembly, however, the cones are butted together so that, when the unit is fixed in place, the proper running clearance is provided. This class is designated as the NA series and is divided into the conventional and steep-angle groups. These bearings, like those already discussed, are suited particularly well to those assemblies where it is not desirable to provide means for take-up in service or where operating conditions require a fixed mounting at one end of an assembly and a certain amount of "float" at the other.

Two-row Bearing, Slotted Double Cone—Type DIK. A bearing developed for use in light-duty pillow blocks is shown in Fig. 15-18. It will be noted that slots are provided in the cone to engage in mating projections on a collar, which in turn is secured to the shaft by means of setscrews. This unit is, of course, not limited to pillow blocks but finds a wide field of usefulness in those applications where cold-finished shafting is used. Since cold-finished shafting may be held to comparatively close limits, it is possible to obtain relatively tight fits on the shaft. This type of bearing, however, is not recommended for heavy-duty service. It should be kept in mind that the bore of these standard bearings is held to the nominal outside diameter of cold-finished shafting. However, tapered-roller-bearing manufacturing practice always provides a plus tolerance, while the standard manufacture tolerance for cold-finished shafting provides for the undersize variation only.

Two-row Bearing, Double Cone, Single Cups—Type DI. The two-row bearing, double cone, single cup, shown in Fig. 15-19, is frequently selected for positions where it is necessary to secure the capacity of a two-row bearing, yet to obtain a simpler mounting than two standard single-row bearings. It is also often used on the fixed end of a shaft when an allowance must be made for shaft expansion. In this case the bearing used on the opposite end is usually one of the double-cup units so set up as to float.

Two-row Bearing, Tapered Bore, Double Cone, Single Cup—Type DIT. The two-row bearing, tapered bore, double cone, single cup, which is identical to the one shown in Fig. 15-19 except that the bore is tapered, was also developed for use on line shafting and in pillow blocks. Like the slotted-cone bearing, however, it is by no means limited to such service but is used extensively in those applications where ease and speed of assembly and disassembly are primary requirements. The usual method of mounting is to use a sleeve tapered to conform to the taper of the bearing. This sleeve should have a saw cut lengthwise to permit it to compress as it advances into the bearing bore, which also provides a rigid seat on the shaft.

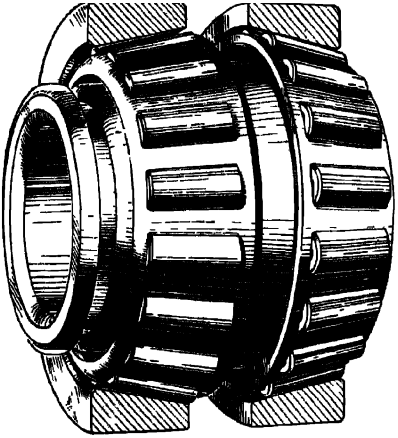


FIG. 15-20. Two-row bearing, tapered bore, double cone with puller—Type DITP.

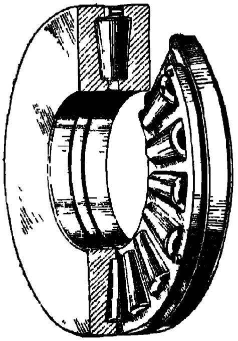


FIG. 15-21. Thrust bearing, heavy duty—Type T.

On line shafting and other positions where it is necessary to be able to slide the complete assembly into position and then fasten it securely, this unit is ideally suited. The field of application is unusually extensive since the tapered sleeve permits a rigid seat on the shaft.

Two-row Bearing, Tapered Bore, Double Cone with Puller—Type DITP. The two-row bearing, tapered bore, double cone with puller, is shown in Fig. 15-20. It differs from the one described above in that a flange is provided on the cone to facilitate removal. The mounting is usually made directly on the shaft. The general field of application covers positions in machines where it is necessary to remove component parts frequently, such as in certain railroad applications and paper-mill machines.

Heavy-duty Bearings. In order to save power, improve performance, and reduce operating costs generally, extensive studies and tests have been made in developing tapered bearings for heavy-duty applications. Numerous installations have been made in various types of rolling mills, cement plants, rubber mills, and other units that develop extreme pressures. Comparative tests have been made, not only to measure the loads which are encountered but likewise to determine the actual savings to be expected. Obviously, this varies with different installations, but in general a saving of from 20 to 50 per cent in power has been obtained.

Power savings, however, are but one of several outstanding advantages since tapered-bearing-equipped rolls may be set to exceptionally close limits, thus accurately maintaining the gauge of the material being rolled. This is frequently an item of major importance, especially in the rubber and metal industries. Likewise the savings in grease and oil are to be considered. In a tapered-bearing installation, carefully designed closures have been developed so that a dual advantage is obtained. First, the lubricant is kept inside the housing so that it performs its primary function of lubricating the moving parts and does not escape to damage the product being handled. Secondly, foreign matter such as water, dust, dirt, or mill scale is excluded.

In addition to the information gained relative to loads and power savings, extensive studies have been made with reference to mountings. This knowledge has been applied as it was acquired so that new applications represent the most advanced thought in tapered-roller-bearing development.

Thrust Bearings—Type T. Although a combination of radial and thrust loads exists much more frequently than either a purely radial or purely thrust load, there are numerous occasions where it is necessary to install a bearing to withstand thrust loads only. To meet this condition, a series of bearings known as the T type is available (Fig. 15-21). These bearings are in regular production in bores ranging from ⅝ in. and a capacity of about 1,000 lb, to massive units having a bore of 11 in. and capable of withstanding over 200,000 lb load.

Special Bearings. Although standard tapered bearings will be found suitable for nearly all applications, at times problems arise requiring special units. The rocker arm of aircraft engines is such an example. The bearing developed for this position is of the two-row double-cup type. In addition to the bearing assembly, *i.e.*, the cone, cup, and rollers, closures are provided so that the unit becomes a complete, self-contained assembly.

MOUNTING TAPERED BEARINGS

In considering the fundamentals of mounting all types of tapered bearings, it must be kept in mind, first of all, that it is impractical to mount a tapered bearing singly. On account of the tapered construction of the bearing, a radial load on one bearing sets up a thrust reaction which must be resisted by the opposed bearing. For this reason tapered bearings must be mounted in pairs. This may be done by using two different types of mountings, referred to usually as direct or indirect, as shown in Fig. 15-22.

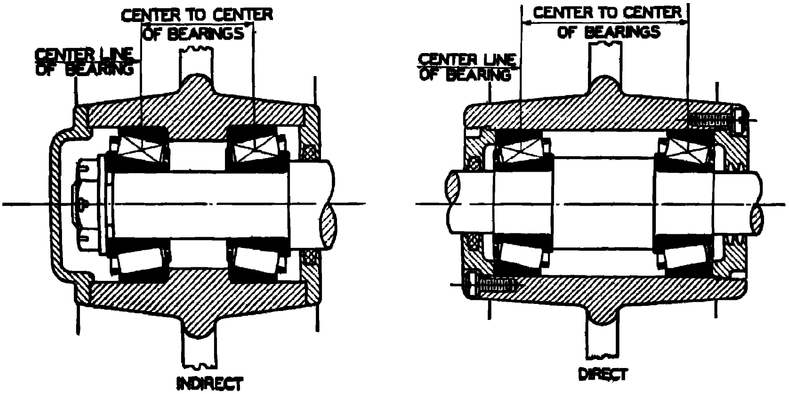


Fig. 15-22. Indirect and direct mounting of tapered roller bearings.

The indirect mounting has the large ends of the rollers of each bearing pointing out or away from the other bearing. The direct mounting has the large ends of the rollers of each bearing pointing toward the mating bearing.

Indirect Mounting. Actual laboratory and field tests have proved the indirect mounting much superior to the direct mounting for wheels, which are usually mounted on a dead shaft. This places the heavy ends of the rollers to the outside, which increases the stability of the application. The distance between bearing centers (Fig. 15-22) should be no less than 15 per cent of wheel outside diameter and preferably 20 per cent.

The indirect mounting is standard in wheels, idler pulleys, sheaves, drums, automotive pinions, and other applications where stability is desired and the distance between bearing centers is relatively short. The increased stability of the indirect over the direct mounting may be better understood by referring to Fig. 15-23. In this illustration the distance between bearing centers *A* is the same. However, the stability of the indirect mounting over the direct is much greater.

Direct Mounting. The direct mounting is used on applications with relatively long bearing centers. It is used on automotive transmission gear shafts differential gear shafts, as well as on machine-tool transmission shafts, general gear reductions, pillow blocks, journal boxes, and other applications that permit wide bearing spreads.

Keeping in mind the two fundamental principles just outlined, each tapered bearing application should be selected to carry the loads to the best advantage. A live shaft application with heavy thrust loads should be designed to carry the thrust on the bearing nearest to the imposed load, *i.e.*, an indirect mounting should be used. Likewise, an overhung bevel-gear arrangement, such as a power take-off, should use a similar mounting since it places the large ends of the rollers adjacent to the heavy radial load. Conversely, where there is a heavy gear or belt-pull load between two bearings, the direct mounting should be used so as to place the large ends of the rollers next to the load.

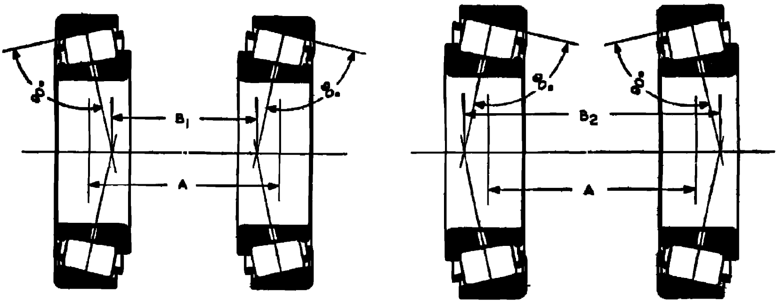


FIG. 15-23. Stability advantage of indirect mounting.

SHAFT AND HOUSING DESIGN

Alternative Methods of Cone Backing—Properly Designed Housing

The setting of the tapered bearing on the shaft and in the housing is important. The usual practice is to seat the cone or cup or both against a shoulder of sufficient height to support whatever thrust and radial loads may be imposed.

Shaft Design. The fillet radius of all shoulder backings should be 1/64 in. less than the radius of the cup or cone.

The conventional method of providing backing is to machine a shoulder on the shaft or in the housing. In some applications it is sometimes necessary to provide alternative backing for the cone by introducing a spacer between the back face of the cone and the shaft shoulder. In this way it may be possible to use a smaller shaft, thereby effecting a production economy.

Another method of providing cone backing for use only with light-duty applications such as conveyor idler rollers or hand trucks is to clamp a split collar tightly to the shaft and lock against end movement by setscrews.

When it is required to maintain maximum shaft strength and also facilitate assembly and disassembly, accurately ground tapered shafts which fit tapered-cone bores are frequently used.

In developing new designs the removal as well as the installation of the bearings should be a matter of primary consideration. The recommended shoulder diameters allow a sufficient portion of the cup and cone to extend so that they may be removed readily. If it is necessary to provide higher shoulders, cored or machined knockout slots should be provided.

Housing Design. In numerous applications it is necessary to provide backing for the small end of the cup with a spacer sleeve supporting the heavy end. This arrangement permits the entire assembly to be clamped firmly in place. When this method is used, some provision must always be made at the small end of the cup for its removal.

Cage clearance must be provided to avoid any rubbing of the rotating bearing cage on the stationary housing parts.

Cylindrical housings and many other applications are chiefly stressed with bending and shearing loads. The designer should consider these loads carefully so that the bearings are not subjected to misalignment, which would bring about early failure.

In split housings, such as generally used with gear reduction units, transmission cases, and similar applications, a cartridge mounting is frequently used, although it is also common practice to mount the bearings directly in the housing. In either case particular attention should be given to the seals so that the lubricant is retained in the housing.

Cleanliness is a matter of the utmost importance, as no bearing will operate successfully in the presence of foreign material, such as core sand, cuttings, scale, or anything of an abrasive nature. When left in the bearing housing, it acts as a lapping agent and leads to rapid wear of the bearings. It is strongly recommended that all inside cast surfaces be cleaned by sand blasting or other means and either painted or coated with a suitable filler to keep the core sand and scale from loosening up in service.

Take-up in Assembly. The take-up feature of the tapered bearing offers many advantages in assembly:

1. Provides wider tolerances for machining bearing seats.
2. Makes possible the adjustment of other dependent parts such as spiral bevel gears and helical gears.
3. Eliminates chatter on machine-tool spindles due to variable bearing clearance.
4. Quieter operation may be obtained by proper bearing adjustment.
5. Elimination of play which may develop after long service.

General applications should have 0.001 to 0.005 in. end play. Special conditions involving elimination of vibration, chatter, unusual quietness, extremely high speed, abnormal heating, or expansion should be taken up with the bearing manufacturer.

The adjustment of the bearing may be made either through the stationary race or the rotating race for speeds up to 1,000 rpm. Above 1,000 rpm, it is usually advisable to make adjustments with the stationary race in order to permit a tight press fit of the rotating race. Adjustment increments of not over 0.003 in. are desired.

In detailing the bearing housing, covers, shaft, etc., provision should be made for machining variations and for future bearing adjustment. When shim adjustment is used, $\frac{3}{32}$ in. should be provided for shims. This will be sufficient for all normal machining variations and in addition will provide a few additional shims for service adjustment. In making assembly, it may be found desirable to place part of the shims at each end of the bearing housing to facilitate adjustment if it becomes necessary. Where threaded adjustment is provided, the $\frac{3}{32}$ in. should be provided for the same reasons. Threads will be used at one end of the assembly only unless it is necessary to vary the position of the entire assembly, in which case threads may be used at each end. Threaded adjusting nuts are used for either cone or cup movement. Light- or medium-duty applications should use nuts with 16 threads per inch and six or more locking castellations. Heavy-duty machinery may require nuts with 12 threads per inch and eight or more castellations. Close-fitting threads are required. Split nuts or housings that can be clamped on mating threads are essential on heavy-duty jobs. Threaded adjustments should not only be positively locked against rotation, but whenever possible the threads should be clamped to ensure tightness against end play.

Whenever possible shim-adjustment devices should be so designed that shims taken out will seat the bearings tighter. This permits field adjustments without requiring new shims. A complete set of shims consists of seven pieces, one of 0.020-in. thickness, three of 0.007-in. thickness each, and three of 0.005-in. thickness each. This permits adjustment increments of 0.001 in. The shims should be smooth sheet metal without rough edges.

Closures—Metal Stampings, Felt, Annular Groove, Stuffing Box, Leather, Cork, Piston Rings, Special, Vertical. In selecting the proper closure design for any tapered-bearing application, it should be kept in mind that the consistency of the lubricant, foreign material to be excluded, speed of the application, and general operating conditions are of unusual importance. Foreign material, such as dust, mill scale, or any hard, gritty substance, will act as a lapping agent and quickly destroy the bearing. Likewise, water, acid, or inferior lubricant will etch the highly finished surfaces and also bring about early failure. A variety of closure designs is therefore available to assist engineers in obtaining the most satisfactory application.

Cone- and Cup-fitting Practice. The adjustable feature of a tapered bearing is an important advantage since it permits the heavy press fits required in numerous applications. Regardless of the fit, the tapered bearing may be set to the correct running clearance after all the other parts have been assembled in position. The proper adjustment of the bearing is therefore independent of variations in mounting conditions.

However, careful study should always be given to the selection of the proper fitting practices, and Tables 15-8 and 15-9 cover in some detail recommendations which should be followed in industrial installations.

Turned Shafts. It is recommended that all cone seats be ground. In those cases where this is impossible a minimum cone seat should be provided equal to the nominal cone bore plus 0.0005 in. per in. of cone bore, to which should be added 0.001 in. for maximum cone seat diameter.

This applies to transmissions, speed reducers, electrical machinery, and all low-speed, heavy-duty, continuous-service applications or applications subject to shock, and using cones with a bore of over 2.5000 in. For all other applications use the fitting practice provided for ground shafts. It is always best to refer any new designs directly to the bearing manufacturer for attention and mounting suggestions.

While there are certain basic rules which should be followed in mounting tapered bearings, some variation is permitted where necessary to meet special conditions of mounting, assembly, and different types of service. For example, on a stationary spindle, a loose fit of the bearing cone can be permitted in some applications to facilitate assembly and disassembly. Such a condition is found in the front wheels of automobiles. In this application, since the spindle itself is usually made of a high-grade alloy steel, there is very little likelihood of damage to either the spindle or the bearings owing to the cone creeping on the spindle. Furthermore, the speed is not high. In applications such as this, however, it is essential to make the outer races or cups a tight press fit in the hub.

When cones are mounted on rotating shafts, especially where the loads are heavy or the speeds high, it is essential that the cones be a tight press fit. Some deviation from this rule may be permitted in exceptional cases, but only where all conditions surrounding the bearing and its operation are known. In mountings where the bearing adjustment is made through the cone, it is possible to fit the cone on the shaft with a light press or even a slip fit provided that the shaft is hard and the parts immediately back of the cone are hard. It should be understood, however, that the ideal mounting is a tight press fit. In many bearing applications incorporating a rotating shaft, adjustment is made through the bearing cup in which case the cone can be a tight press fit and the cup a relatively loose fit in the housing. There should be no tendency for the cup to rotate in the housing under these conditions.

Stationary-shaft Applications. It is possible and quite often desirable to mount cones loosely on stationary-shaft applications where the average speed is not likely to exceed 500 rpm. This permits the cone to creep, thereby utilizing the full working surface. It also facilitates assembly. Examples of such applications are automobile front wheels, tractor front wheels, farm-implement wheels, shop-truck wheels, and similar applications. It is, of course, necessary to provide against excessive looseness

Table 15-8. Industrial Cone-fitting Practice for Ground Shafts

Type of application	Cones	Cone bore								
		Less than ⅝ in.			⅝–2½ in.			Over 2½–12 in.		
		Toler-ance	Recom-mended fit	Cone seat diam equals nominal cone bore	Toler-ance	Recom-mended fit	Cone seat diam equals nominal cone bore	Toler-ance	Recom-mended fit	Cone seat diam equals nominal cone bore
Rotating shafts										
Machine-tool spindles, precision bearings....	Adj. and nonadj.	+.0005 −.0000	.0005 tight 0005 loose	+ 0005 max + 0000 min	+.0005 −.0000	0013 tight .0003 tight	+ 0013 max +.0008 min	+.0005 −.0000	.0013 tight .0003 tight	+.0013 max +.0008 min
Transmissions, speed reducers, electrical equipment, etc.	Adj. and nonadj.	+.0005 −.0000	.0005 tight .0005 loose	+.0005 max +.0000 min	+.0005 −.0000	0015 tight .0005 tight	+.0015 max +.0010 min	+ 0010 −.0000	.0025 tight .0005 tight	+.0025 max +.0015 min
Low speed, heavy duty, continuous service, shock, etc.	Adj. and nonadj.				+.0005 −.0000	.0025 tight 0010 tight	+.0025 max +.0015 min	+.0010 −.0000	Use turned-shaft fitting practice. See p. 1193	
Stationary shafts										
Machine-tool spindles, precision bearings......	Adj. and nonadj.	+.0005 −.0000	.0005 tight 0005 loose	+.0005 max +.0000 min	+.0005 −.0000	.0013 tight .0003 tight	+.0013 max +.0008 min	+.0005 −.0000	.0013 tight .0003 tight	+.0013 max +.0008 min
Transmissions, speed reducers, electrical equipment, etc. (speeds up to 1,000 *rpm*).	Adj. and nonadj.	+ 0005 −.0000	0005 tight 0005 loose	+.0005 max +.0000 min	+.0005 −.0000	0005 tight .0005 loose	+.0005 max +.0000 min	+.0010 −.0000	.0010 tight .0010 loose	+.0010 max +.0000 min
Low speed, heavy duty, continuous service, shock, etc.	Adj. and nonadj.				+.0005 −.0000	.0025 tight 0010 tight	+.0025 max + 0015 min	+.0010 −.0000	Use turned-shaft fitting practice. See p. 1193	
Sheaves, crown blocks, tractor front wheels, implement wheels, etc.	Adj.				+.0005 −.0000	0000 tight .0010 loose	+.0000 max −.0005 min	+.0010 −.0000	.0000 tight .0020 loose	+.0000 max −.0010 min

Table 15-9. Industrial Cup-fitting Practice

Type of application	Cups	Cup outside diam								
		Less than 3 in.			3–5 in.			Over 5–12 in.		
		Tolerance	Recommended fit	Cup seat diam equals nominal cup outside diameter	Tolerance	Recommended fit	Cup seat diam equals nominal cup outside diameter	Tolerance	Recommended fit	Cup seat diam equals nominal cup outside diameter
Machine-tool spindles, precision bearings.	Adj.	+.0005 −.0000	0005 loose 0005 tight	+.0005 max +.0000 min	+.0005 −.0000	.0005 loose .0005 tight	+.0005 max +.0000 min	+.0005 −.0000	.0005 loose .0010 tight	+.0005 max −.0005 min
	Nonadj.	+ 0005 −.0000	0005 tight 0015 tight	−.0005 max −.0010 min	+.0005 −.0000	.0005 tight 0015 tight	−.0005 max −.0010 min	+.0005 −.0000	0005 tight .0020 tight	−.0005 max −.0015 min
	Floating	+.0005 −.0000	0010 loose .0000 loose	+.0010 max +.0005 min	+.0005 −.0000	.0010 loose .0000 loose	+.0010 max +.0005 min	+.0005 −.0000	.0015 loose .0000 loose	+.0015 max +.0005 min
Transmission, speed redurers, electrical machinery, low speed, heavy duty, continuous service, shock loads.	Adj.	+.0010 −.0000	0010 loose 0010 tight	+.0010 max +.0000 min	+.0010 −.0000	.0010 loose 0010 tight	+.0010 max +.0000 min	+.0010 −.0000	.0020 loose .0010 tight	+.0020 max +.0000 min
	Nonadj.	+.0010 −.0000	0005 tight .0025 tight	−.0005 max −.0015 min	+.0010 −.0000	.0010 tight .0030 tight	−.0010 max −.0020 min	+.0010 − 0000	.0010 tight .0030 tight	−.0010 max −.0020 min
	Floating	+.0010 −.0000	.0020 loose 0000 loose	+ 0020 max +.0010 min	+.0010 − 0000	.0020 loose .0000 loose	+.0020 max +.0010 min	+.0010 −.0000	.0030 loose .0010 loose	+.0030 max +.0020 min
Sheaves, crown blocks, tractor front wheels, implement wheels, etc.	Nonadj.	+.0010 −.0000	0005 tight 0025 tight	−.0005 max −.0015 min	+.0010 −.0000	.0010 tight 0040 tight	−.0010 max −.0030 min	+.0010 −.0000	.0010 tight .0040 tight	−.0010 max −.0030 min

of the cones even in these applications, and cone seats and shoulders backing up the cones should be smoothly machined or ground. The actual fit in this type of application where alloy-steel or other high-grade heat-treated steel spindles are used will vary from 0.0002 loose to 0.0012 in. loose in the small-size bearings, and in the larger sizes the fits will vary from 0.0002 loose to 0.0017 in. loose. Where ordinary carbon-steel spindles are used, a fit of 0.0005 tight to 0.0015 in. loose is recommended.

Light press fits of cones are recommended for stationary-shaft applications such as found on loose pulleys, idler gears, and similar applications. In such applications, where speeds may run from 500 to 1,000 rpm, loose-fitting cones should be avoided if possible. Assembly problems may necessitate a loosely fitted cone, but in these cases care should be taken to see that a good steel is used in the shaft and parts adjacent to the bearing. On small bearings these fits will vary from 0.0005 tight to 0.0005 in. loose and on the larger bearings from 0.001 tight to 0.001 in. loose, the desired fit being about a line-to-line condition.

In applications where speeds are above 1,000 rpm or where the loading condition is particularly severe, tight fits are recommended. The tightness of the cone fit should be approximately 0.0005 in. press per in. of shaft diameter. If adjustment is to be made through the tightly fitted cones, it will be necessary to heat the cones during assembly and make the proper setup while the cones are still hot. Where precision bearings are used in which cone bores are held to 0.0005-in. tolerance, a lighter fit may be necessary to maintain utmost precision. In such cases the shaft must be ground, and a press fit of from 0.0005 in. tight to 0.0015 in. tight is satisfactory.

Tight press fits are required on cups in stationary-shaft applications in practically all cases. A loose cup in a rotating housing such as a wheel hub is bound to rotate with respect to the housing itself, and trouble is very likely to be encountered. In some cases where the adjustment is made through the cup, such a fitting practice may be necessary. Before adopting such a design, it is recommended that the bearing manufacturer be consulted so that proper recommendations can be made.

Cups up to 3 in. in outside diameter should be mounted with a press fit of from 0.0005 in. tight to 0.0025 in. tight; above 3 in. in outside diameter, the fit recommended is from 0.001 in. tight to 0.003 in. tight. Where precision bearings are supplied with outside-diameter tolerance of +0.0005 in., a press fit of from 0.0005 in. tight to 0.0015 in. tight is satisfactory up to 5 in. outside diameter, and above that the recommended fit is from 0.0005 in. tight to 0.002 in. tight.

Rotating-shaft Applications. On rotating-shaft applications the cone fit on the shaft is very important. This is particularly true where the speeds are high and loads severe. Where the loads encountered are heavy, it is absolutely essential to have the cones a tight press fit regardless of whether the adjustment is made through the cone or through the cup. In a very few special applications where the loads are relatively low, cones are fitted loosely on the shaft, but in these cases special provision is made to ensure against excessive turning and wearing of the shaft itself. In general, all cones supplied with a standard bore tolerance of +0.001 in. should be fitted with a normal press fit of 0.0005 in. per in. of shaft diameter. With cones that are supplied with a cone-bore tolerance of +0.0005 in. and used in precision mountings such as machine-tool spindles or where the cones are mounted on hardened and ground shafts, a press fit of from 0.0005 in. tight to 0.0015 in. tight is satisfactory.

Where bearing adjustment must be made through the cone and where the loads are light, a looser fit is sometimes used. Such a loose fit can be used satisfactorily only on a hardened shaft and with the shoulder immediately back of the cone also hardened. An example of such an application is the forward pinion bearing used in passenger-car and truck axles. The cone fit used in these applications is from 0.0005 tight to 0.0005 in. loose. Wherever possible, a further safeguard against excessive rotation should be provided by clamping the cone endwise. This construction is quite common where shims are used for initial setup.

In rotating-shaft applications the bearing cups are, of course, stationary, and an adjustable fit is in most cases satisfactory. Cups provided with the standard tolerance of +0.001 in. on the outside diameter up to 5 in. should be fitted from 0.001 in. loose to 0.001 in. tight. Above 5 in. outside diameter, these cups should be fitted from

0.002 in. loose to 0.001 in. tight. Adjustable cups up to 5 in. outside diameter, which are supplied with a tolerance on the outside diameter of +0.0005 in., such as are used in machine-tool and other precision applications, should have a fit of 0.0005 in. loose to 0.0005 in. tight. Cups above 5 in. outside diameter and supplied with the same tolerance should be mounted with a fit from 0.001 in. loose to 0.0005 in. tight.

Where adjustment is made through the bearing cone, the cup should be given a tight press fit in the housing. Cups up to 3 in. outside diameter and furnished with the standard tolerance of +0.001 in. used in a mounting with cone adjustment should be given a press fit of from 0.0005 in. tight to 0.0025 in. tight. All cups above 3 in. outside diameter should be fitted from 0.001 in. tight to 0.003 in. tight. Cups up to 5 in. outside diameter and furnished with an outside-diameter tolerance of +0.0005 in. should be given a press fit of from 0.0005 in. tight to 0.0015 in. tight. Cups with the same tolerance but over 5 in. outside diameter should be fitted from 0.0005 in. tight to 0.002 in. tight.

Where double bearing mountings are used, the bearing cups at one end of the mounting are usually allowed to float in the housing to provide for expansion or machining variations. In these mountings, adjustments are made through the cones, and the cups are mounted back to back either directly against each other or through a spacer. Standard cups with outside-diameter tolerance of 0.001 in. are fitted from 0.000 in. loose to 0.002 in. loose up to 5 in. outside diameter; from 0.001 in. to 0 003 in. loose from 5 in. outside diameter up to 7.500 in. outside diameter; and 0.002 in. to 0.004 in. loose over 7.500 in. outside diameter. In precision bearings up to 5 in. outside diameter, the cups are fitted in the housing from 0.000 in. loose to 0.001 in. loose; above 5 in. outside diameter the fit is 0.000 in. loose to 0.0015 in. loose. All precision-bearing cups have an outside-diameter tolerance of +0.0005 in.

The foregoing information relative to fitting practices to be followed on tapered cups and cones is, of course, general and should serve as a satisfactory guide in the majority of standard bearing applications. In order to facilitate the specifications of proper shaft diameters and housing bores, Tables 15-8 and 15-9 have been prepared. Certain types of applications and mountings have been classified so that the selection of the proper fit is comparatively simple.

In the "NA" series bearing, certain fits must be observed in order to obtain the proper running clearance, which is established at the time of manufacture. These fits should be obtained from the bearing manufacturer.

As stated previously there are numerous special bearing applications for which special fitting practices are required. Fitting practices for such applications should be obtained directly from the bearing manufacturer.

SELF-ALIGNING ROLLER BEARINGS

BY BRYCE RULEY

Senior Engineer, SKF Industries, Inc.

This subsection describes those types of roller bearings which are inherently self-aligning, *i.e.*, able to adjust to misalignment between shaft and housing bore without loss in carrying capacity.

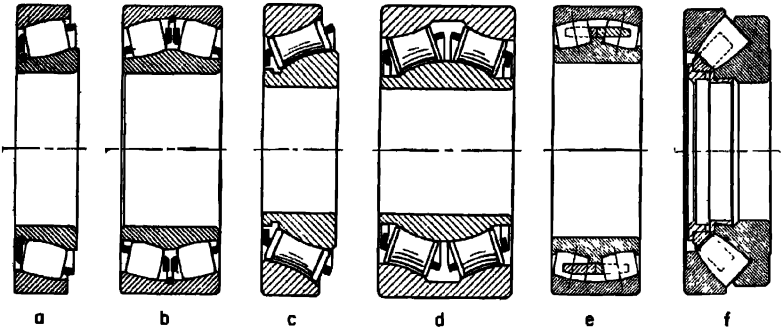


FIG. 15-24. Self-aligning roller-bearing types: (*a*, *b*) spherangular; (*c*, *d*) concavex; (*e*) spherical; (*f*) spherical thrust.

The ability to self-align is obtained by making either the inner or the outer race of the bearing a section of a sphere. The other ring and the roller assembly can align freely on this surface to adjust to any initial or operating misalignment.

Various designs which give approximate line contact between rollers and raceways and allow self-aligning between inner and outer rings are shown in Fig. 15-24 and described in detail in the material that follows.

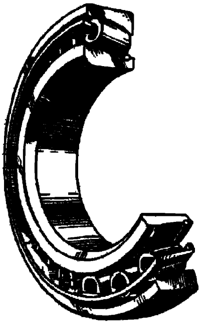


FIG. 15-25. Spherangular roller bearing.

SPHERANGULAR BEARING

The spherangular bearing is a symmetrical barrel roller type of angular-contact bearing designed for combined radial and thrust loads. The bearing is self-aligning, but the roller and ring curvatures are such that proper load distribution is obtained even under conditions of misalignment. The load direction is always through the largest diameter of the roller.

The single-row bearing is made with a separable outer ring, while the double row is nonseparable. In the larger sizes the double row is made with a center flange for roller guidance.

The thrust capacity depends on the angularity of the rollers, *i.e.*, the angle at which the rollers are placed with respect to the radial plane of the bearing. This angle varies from approximately 10 to approximately 25 deg, the higher angles being used where higher thrust capacity is required (see Fig. 15-25).

CONCAVEX ROLLER BEARING

The concavex roller bearing is an angular-contact roller bearing designed to carry combined radial and thrust loads. The rollers are curved so that the largest diameter is at the ends of the rollers, *i.e.*, the rollers are concave in one direction and convex in the other. These rollers conform to the spherical inner race to give line contact and self-alignment without reduction in carrying capacity.

The single-row bearing has a separable outer ring. The double-row bearing is sometimes made with two separable outer rings and sometimes with only one non-separable outer ring.

The single-row bearing can carry thrust or combined loads in one direction only, while the double-row bearing can carry reversed thrust or a pure radial load. A number of bearing series are available in this bearing type to handle a wide variety of load conditions.

SPHERICAL ROLLER BEARING

The spherical roller bearing is a double-row type designed to carry heavy radial, thrust, or combined loads. Since the two rows of rollers have opposed angles of contact, the bearings can support reversed thrust loads or combined loads in any direction. The outer race of the bearing is a section of a sphere in which the roller assembly and inner race move to adjust to any misalignment.

The rollers are usually made with a tapered barrel shape. This allows a close approach to true rolling motion between rollers and raceways and holds the ends of the rollers firmly against the central guiding flange assuring proper roller guidance.

This bearing is manufactured in a wide variety of dimensions and dimensional proportions so that a bearing selection can be made to suit any load condition. This bearing is available with either a straight bore or a tapered bore to suit a variety of mounting conditions (see Fig. 15-26).

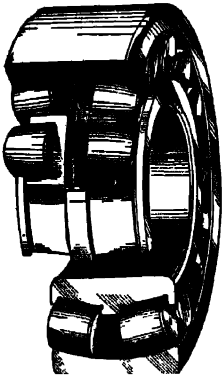


FIG. 15-26. Spherical roller bearing.

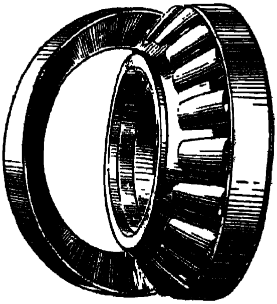


FIG. 15-27. Spherical roller thrust bearing.

SPHERICAL ROLLER THRUST BEARING

The spherical roller thrust bearing is a single-row angular-contact bearing designed to carry heavy thrust loads or combined loads that are predominantly thrust. The outer race is a spherical section making the bearing self-aligning.

The rollers have a tapered barrel shape which holds the roller ends against the high guide flange on the inner ring, assuring proper roller guidance.

The lubrication preferred for this bearing is an oil of fairly high viscosity. The bearing is designed so that the oil is pumped outward from the bearing center over the rollers, cage, and guide flange, giving uniform lubrication to all bearing parts.

Spherical roller thrust bearings may be mounted in pairs, back to back, or face to face, to carry heavy reversed thrust loads (see Fig. 15-27).

BALL BEARINGS

BY BRYCE RULEY

Senior Engineer, SKF Industries, Inc.

Ball bearings fall into a few main classes, each class having different characteristics. Figure 15-28 shows some common ball-bearing types and the following paragraphs describe these types more fully.

SINGLE-ROW RADIAL BALL BEARINGS

The *deep-groove* or *Conrad* type of radial ball bearing has raceways which consist of deep continuous grooves cut in the bearing rings. Since the radii of the grooves are only slightly greater than the radii of the balls, the balls conform closely to the raceways, giving good support to the areas of contact and accurately guiding the direction of rolling of the balls.

Deep-groove ball bearings can carry radial, thrust, or combined load. The wide range of load and operating conditions for which deep-groove ball bearings give excellent service make this bearing a popular general-purpose bearing for light and medium loads.

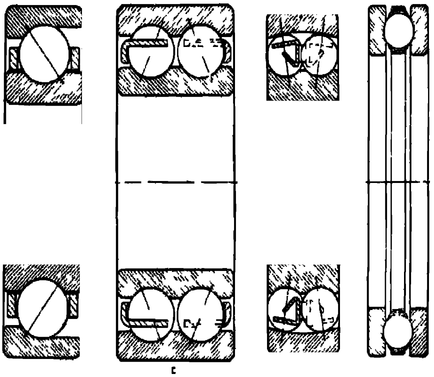


FIG. 15-28. Ball bearings. Deep-groove radial—*a*; angular-contact—*b*; double-row—*c*; self-aligning—*d*, ball thrust —*e*.

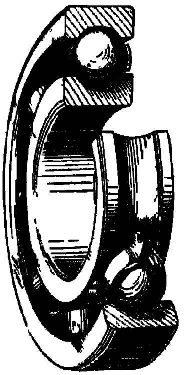


FIG. 15-29. Deep-groove ball bearing.

A variation of the deep-groove bearing is the *maximum* or *filling-notch* type. This bearing has raceways which are grooves cut in the bearing rings, but notches have been cut in the groove shoulders on one side of the bearing to allow a greater number of balls to be inserted in the bearing (see Fig. 15-29)

The maximum-type bearing is designed to carry radial loads and combined loads that are predominantly radial. It can carry moderate thrust loads but is not usually recommended for these.

Radial ball bearings are made in a *separable* type in which one shoulder of the outer race has been cut down to allow a full-complement ball assembly to be snapped in over the low shoulder. This bearing can carry radial load and thrust or combined load in one axial direction only. This type of bearing is sometimes called a *magneto*-type bearing.

A number of other modifications have been made to the basic radial ball bearing. In current manufacture are bearings with extended inner races, bearings with snap rings, tapered-bore bearings, bearings with built-in plates or friction seals, etc.

ANGULAR-CONTACT BALL BEARING

If a deep-groove bearing with considerable internal clearance or slackness is loaded by a thrust load, the bearing moves so that the load passes through the balls at an angle to the radial plane of the bearing. This angle is called a contact angle. Radial ball bearings have an initial contact angle of 0 and may under heavy thrust load have a contact angle of 15 deg or more.

Some bearings are made so that they have an initial contact angle. This improves their thrust-carrying ability but prevents them from carrying pure radial load unless they are opposed by a similar bearing. Bearings of this type are called *angular-contact* bearings. In current production are bearings whose initial contact angles range from 15 to 40 deg. These bearings are designed to carry light or medium combined loads which are mostly thrust and are often used to carry thrust loads at high speeds (see Fig. 15-30).

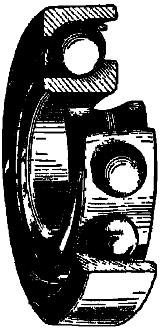


Fig. 15-30. Angular-contact ball bearing.

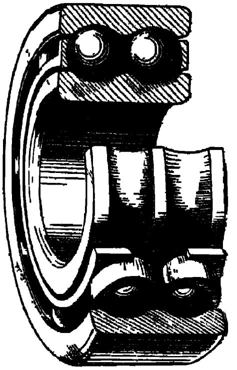


Fig. 15-31. Double-row ball bearing.

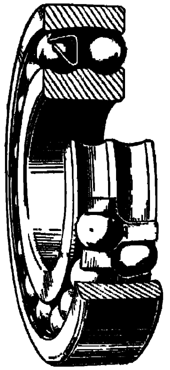


Fig. 15-32. Self-aligning ball bearing.

DOUBLE-ROW BALL BEARINGS

Double-row ball bearings are similar to two single-row radial bearings placed side by side except that the raceways for both rows are set in common rings. This bearing is sometimes made as a deep-groove type, sometimes as a maximum type, and sometimes as a double-row angular-contact bearing with opposed contact angles.

Double-row bearings have nearly twice as much radial capacity as single-row bearings, but their thrust capacity is usually about the same as the single-row bearing. This bearing is more sensitive to misalignment than the single row since a slight misalignment can result in one row of balls carrying all the load (see Fig. 15-31).

SELF-ALIGNING BALL BEARINGS

The *self-aligning* ball bearing is a double-row type in which the grooved outer races have been replaced by a single spherical-section raceway. This bearing is thus able to self-align to adjust for any misalignment between shaft and bearing-housing bore.

Self-aligning ball bearings usually have slightly less radial capacity and substantially less thrust capacity than the equivalent size of deep-groove ball bearing. However, their ability to adjust to misalignment gives them more effective capacity under conditions where considerable misalignment is present.

These bearings are made in a number of series so that the customer has some latitude in selecting bearings to match his needs (see Fig. 15-32).

BALL THRUST BEARINGS

Ball thrust bearings are grooved-type bearings with contact angles of 90 deg. This means that the grooves are cut in rings so that the load line passes through the balls in direction parallel to the shaft.

Ball bearings can carry pure thrust loads only. If radial loads occur, they must be carried by separate bearings. Ball thrust bearings run under considerable difficulty at high speeds since centrifugal and gyroscopic forces act on the balls to prevent true rolling motion. This increases the friction in the bearings and gives them a low limiting speed.

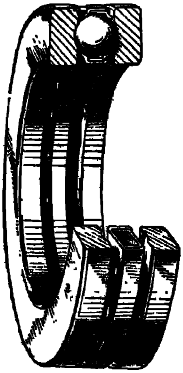


FIG. 15-33. Ball thrust bearing.

Ball thrust bearings are very sensitive to misalignment. Some bearings are made with self-aligning seats on one bearing ring. This proves fairly satisfactory in caring for initial misalignment but has not been nearly so successful with misalignment that varies with operating conditions.

Two-directional ball thrust bearings have been made to carry reversed thrust loads (see Fig. 15-33).

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SECTION 16

ELECTRICAL SPEED CONTROL

BY R. B. IMMEL

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  D. A-c wound-rotor induction-motor drives; speed adjustment of a three-phase wound-rotor induction motor by secondary resistance control
  E. A-c synchronous-motor drives
V. A-c and d-c universal-motor drives—a-c or d-c power sources; speed adjustment and control of a universal motor by a centrifugally operated governor
VI. D-c motor drives—a-c power source and conversion equipment
  A. D-c series adjustable-voltage drives
  B. D-c self-excited shunt adjustable-voltage drives
  C. D-c separately excited shunt adjustable-voltage drive
  D. D-c separately excited shunt adjustable-voltage drives with Rototrol regulators
  E. D-c electronic (Mot-o-trol) adjustable-voltage drives
VII. A-c motor drives—A-c power source and conversion equipment
  A. A-c adjustable-frequency drives—motor-generator sets
  B. A-c adjustable-frequency drives—induction-generator sets
VIII. Summary and comparison of adjustable-speed drive data

ELECTRICAL SPEED CONTROL

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INTRODUCTION

The successful operation of many motor-driven machines or devices depends upon the speed control. Some machines require operation over a wide range of speed, while others require operation at a closely regulated speed. The speed-control means may be mechanical, completely electrical, or a combination of mechanical and electrical means. However, electrical speed control is generally characterized by the ease with which relatively small changes in voltage and current can effect changes of very large magnitude in the power circuits and the motor output speed, torque, or horsepower characteristics. In general, electrical speed-control means are more flexible, quicker and easier to adjust, faster in response, and much more suitable for complicated sequence operations and interlocking than any of the other speed-control methods.

Electrical speed control can be accomplished by manual or automatic control devices. It is particularly well adapted to automatic speed control, and this type of control should be used where it is economical to do so. Compared to many mechanical speed-control mechanisms, the electrical control system may also provide a much wider range in speed control.

The selection of the proper control equipment for a specific application is equally as important as the selection of the most suitable motor. A great many factors, such as the type of power supply available, the type of machinery to be driven, the speed range desired, the type of control wanted, the location of the electrical equipment, the duty cycle, the operating personnel available, the cost, etc., must be taken into consideration in selecting the most suitable electrical apparatus for the application.

The type of power supply available is usually given the first consideration in the selection of an electrical drive which consists of the motor, control, and auxiliary control apparatus. However, in many applications the speed-control factor is predominant over the power supply in selecting the drive that will in the long run be the most economical and satisfactory.

A complete knowledge of the essential features of the driven machinery and the load characteristics is required before an intelligent selection of the electrical drive can be made. With the exception of the a-c synchronous-motor drives and drives with speed-regulating controls, *the speed of many electrical drives is determined directly by the torque requirements of the load.* When the speed-torque requirements of a driven load are definitely known, the electrical drive performance can be predicted reasonably accurately.

The relationship between the drive torque, the horsepower output, and the speed can be expressed by the following equation:

$$T = (\text{hp} \times K)/N \qquad \text{or} \qquad \text{Hp} = (T \times N)/K$$

where T = torque
hp = horsepower
N = speed
K = conversion constant

When the torque T is expressed in pound-feet, and the speed N in rpm, the constant K is equal to 5,250.

ELECTRICAL DRIVE CLASSIFICATION

Electrical drives may be classified by the load requirements which are generally known as constant torque, constant horsepower, a combination of constant torque and constant horsepower, and variable torque.

Constant-torque Electrical Drives. Compressors, chain-grate stokers, belt conveyors, cranes or hoists, plunger pumps, and loading machines are generally considered to be constant-torque loads. Figure 16-1 illustrates the speed, torque, and horsepower relationships of a constant-torque electrical drive. With a constant-torque load, the total pull at any time will be constant and independent of the load. Speed adjustment is required to vary the amount of material being transported, and the horsepower required will be a direct function of the speed. Friction also represents a constant-torque load, and in some inefficient machines, this friction load may be predominant and cause the machine to have a constant-torque characteristic regardless of the useful-load characteristics.

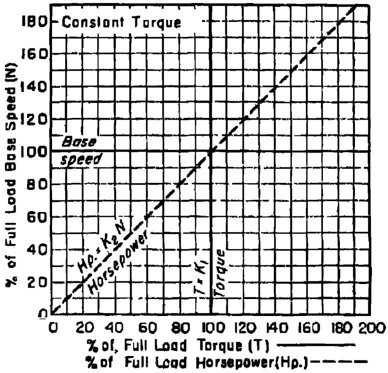


Fig. 16-1. Characteristics of a constant-torque electrical drive.

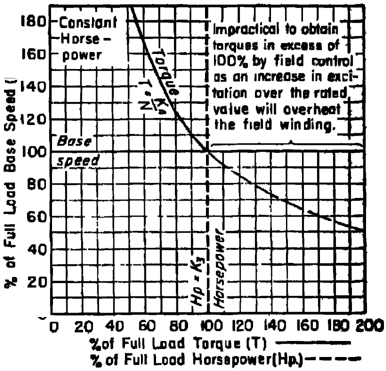


Fig. 16-2. Characteristics of a constant-horsepower electrical drive.

Constant-horsepower Electrical Drives. Figure 16-2 shows the relationship between the speed, horsepower, and torque for constant-horsepower electrical drives. Machine tools, lathes, planers, drill presses, and any machine with a controlled load is usually considered to be a constant-horsepower load. When the load (or torque) is light, the machine can be operated at a relatively fast speed; and when the load is heavy, the machine should be operated at a slower rate of speed. For example, consider the load characteristics of a lathe. As the cutting speed is increased, the depth of cut is decreased to reduce the torque required to drive the load. Since the horsepower is proportional to the product of the torque and speed, the horsepower required for this type of application is approximately constant at any speed. This is not unexpected as the volume of metal that can be removed per unit time is approximately constant over the entire speed range.

For a core-type paper-winding machine, as the diameter of the roll increases, the speed of the driving motor must be decreased to hold the sheet tension and linear paper speed constant. Constant tension at constant linear paper speeds requires constant horsepower over the complete range regardless of how the motor speed varies with the diameter of the roll.

Constant-torque—Constant-horsepower Electrical Drives. For loads requiring a wide speed range, a combination constant-torque and constant-horsepower drive is often utilized. The characteristics of this type of drive are shown by Fig. 16-3. This type of drive has been used very successfully for elevator and metal-working planers. This drive characteristic is typical of a d-c shunt motor which has armature-voltage control below the base speed for constant torque and field control above the base speed for constant-horsepower output.

Variable-torque Electrical Drives. Fans, blowers, and centrifugal pumps are considered to be variable-torque loads as the torque varies directly with the speed. The horsepower is usually considered to vary approximately as the square of the speed for a variable-torque load. The characteristics of a variable-torque load are illustrated in Fig. 16-4.

In Figs. 16-1 to 16-4 subscripts have been added to the constant K to indicate that the numerical value of K is not the same.

It should be thoroughly understood that the characteristics indicated by Figs. 16-1 to 16-4 are the maximum or boundary conditions for the torque or horsepower that the electrical drive motor can safely deliver. It is not likely that a single load will have exactly the same speed–torque characteristics as the drive motor and control; but the load-speed–torque requirements must never exceed those of the drive motor.

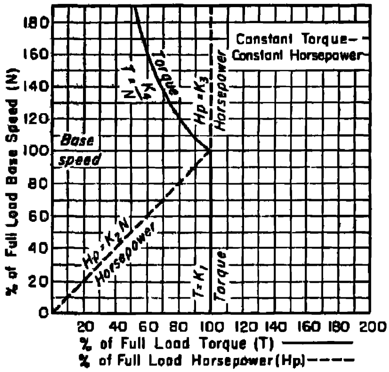


FIG. 16-3. Characteristics of a constant-torque–constant-horsepower electrical drive.

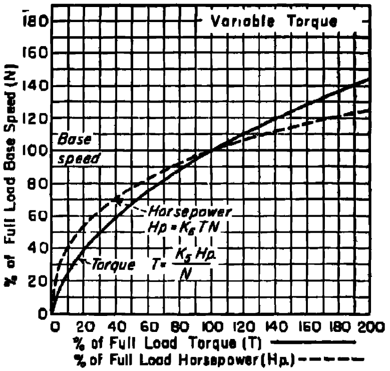


FIG. 16-4. Characteristics of a variable-torque electrical drive.

As the drive-motor torque and horsepower output are directly related to the current that the motor draws from the line, the maximum torque or horsepower limitations must not be exceeded, or the motor temperature will exceed its rated and safe temperature rise. As a motor generally has considerable thermal capacity, intermittent overloads are not objectionable for some applications like cranes, hoists, etc., where intermittently rated motors are usually used to obtain the most economical installation.

The drive motor and control for a given load must always be selected to satisfy the load requirements under all conditions of speed and torque. Obviously, the motor torque available at each speed must be equal to or greater than the requirements of the driven load. The maximum torque and speed requirements of a load usually determine the horsepower rating of the drive, and this rating may be many times that required when the machine is operated at other than maximum speed.

With the exception of a-c synchronous-motor drives and drives where the motors are operated in conjunction with automatic speed-regulating controls, the speed will be determined directly by the driven load. The motor will always operate at a speed where its speed–torque characteristics are exactly in equilibrium with the speed–torque requirements of the driven load. This condition is illustrated later in this section on some of the curves where the operating speeds are predicted graphically by plotting the load and drive-motor characteristics on the same set of coordinates.

Every electric drive motor possesses certain inherent speed characteristics by which it can be classified into a specific category. The following classifications have been adopted by the National Electrical Manufacturers Association (NEMA) and should help to suggest the type of motor to be applied to a specific application.

A *constant-speed* motor is one the speed of normal operation of which is constant or practically constant. For example, a synchronous motor, an induction motor with small slip, or an ordinary direct-current shunt-wound motor.

A *varying-speed* motor is one the speed of which varies with the load, ordinarily decreasing when the load increases; such as a series motor, or an induction motor with large slip.

An *adjustable-speed* motor is one the speed of which can be varied gradually over a considerable range, but when once adjusted remains practically unaffected by the load; such as a shunt motor with field-resistance control designed for a considerable range of speed adjustment.

An *adjustable varying-speed* motor is one the speed of which can be adjusted gradually, but when once adjusted for a given load will vary in considerable degree with change in load; such as a compound-wound direct-current motor adjusted by field control or a slip-ring induction motor with rheostatic speed control.

A *multispeed* motor is one which can be operated at any one of two or more definite speeds, each being practically independent of the load. For example, a direct-current motor with two armature windings, or an induction motor with windings capable of various pole groupings.

The basic methods of speed control for d-c and a-c motor drives have been illustrated with particular emphasis on the speed control. While this section is not directly concerned with methods of starting motors, speed control and starting are inherently linked together, and the starting means must also be considered. The apparatus shown in the illustrations may be manually or magnetically operated to illustrate the diversity of the choice of equipment. Manual speed-control apparatus is generally simpler and less expensive than magnetically operated control equipment. For automatic operation, overload protection, interlocking, and limit or auxiliary switch control, manual control is not generally satisfactory. The use of magnetically operated control is imperative to obtain this type of operation.

Emphasis has been placed in particular on the speed range, speed regulation, size of the drives available, and the economics of one speed-control system over another.

It should be borne in mind that there are many variations of the basic speed-control schemes shown; however, a thorough understanding of the basic motor characteristics and the methods that can be utilized to control the speed of a motor will make it possible to understand many of the more complicated control schemes. Some times artificial loads are imposed on the driving motor to provide more stable operation. These control systems are considered to be too special for this section and will not be discussed.

D-C MOTOR DRIVES—D-C POWER SOURCE

Although a-c induction motors are the most widely used type of motor because of their relatively low cost, simplicity, and the extensive distribution of a-c power, there are many applications where the use of d-c drive motors is absolutely necessary to meet satisfactorily the operating requirements. In general, an adjustable and wide speed range with a very accurate means of speed control can be obtained only with d-c drives.

D-c Shunt-wound Motor Drives. Shunt-wound d-c motors are essentially constant-speed motors with the field winding connected in parallel with the motor armature. When operated on a constant voltage supply and with a fixed value of field current, they operate at essentially constant speed regardless of variations in the load. There is some change in speed or regulation as the load varies from no load to full load, but in general, at the base speed this does not exceed 12 per cent for integral-horsepower motors up to 5 hp and 10 per cent for motors larger than 5 hp.

Constant-speed motors are usually suitable for a speed range up to 2:1 by field control and for constant-horsepower output, provided the top speed does not exceed the maximum safe rpm for which the motor was designed.

Direct-current motors which have a speed range by field control of 3:1 or more are considered to be adjustable-speed motors. Standard motors are available for 3:1

and 4:1 speed ranges. Motors with speed ranges up to 8:1 are available, but these motors are more expensive as special designs and field windings must be used.

D-c-motor Basic Characteristics and Speed Formulas. The direction of the motor rotation which results from the relationship between the direction of the current flow in the armature and the direction of the magnetic lines of force produced by the shunt field for motor operation is such that a counter emf E_A is generated in the armature conductors as they pass through the magnetic field. This counter emf varies from zero at armature standstill to approximately line voltage less the armature-circuit voltage drop I_AR_A at full speed. This counter emf E_A is almost directly proportional to the motor speed.

This characteristic can be represented by the following equation:

$$E_A = K\Phi N$$

where E_A = counter emf
K = motor constant
Φ = field flux
N = speed

Figure 16-5 shows this inherent motor characteristic graphically. The counter emf is in opposition to the impressed line voltage V. Hence, the net voltage that causes current to flow in the armature conductors and which produces motor action is the difference between the impressed line voltage V and the counter emf E_A.

This relationship can be expressed by the following formulas:

$$I_A = (V - E_A)/R_A \qquad \text{or} \qquad V = E_A + I_AR_A$$

where I_A = armature current
V = line voltage
E_A = counter emf
R_A = total armature circuit resistance

The basic d-c-motor speed equation can now be derived as follows:

as $$V = E_A + I_AR_A$$
and $$E_A = K\Phi N$$
then $$N = (V - I_AR_A)/K\Phi$$

where N = speed
V = line voltage
I_A = armature current
R_A = total armature circuit resistance
K = motor constant
Φ = field flux

As the product of the full-load armature current I_A and the armature resistance R_A can be considered a constant, an examination of the preceding speed equation shows that the motor speed can be varied by changing the line voltage V when the shunt-field flux Φ is held constant. This equation also shows that a properly designed motor should have an I_AR_A voltage drop which is small in comparison to the impressed line voltage V so that the speed will not vary appreciably with changes in the armature current I_A. In general, this value is approximately 10 per cent of the line-voltage value V for shunt motors.

For constant shunt-field excitation, the change in speed with a change in the per cent of line voltage V impressed on the armature of a shunt motor is shown by Fig. 16-6.

The motor speed can also be changed by varying the shunt-field flux Φ. If the impressed line voltage V is held constant, the speed will increase with a decrease in the flux Φ. This characteristic is shown by Fig. 16-7.

Because of the physical limitations, the shunt-field flux cannot readily be increased beyond the value that is produced when the field is energized at rated line voltage V for the slowest speed. As previously shown by Fig. 16-6, to obtain speeds below the base or full-field speed, the voltage applied to the armature must be reduced.

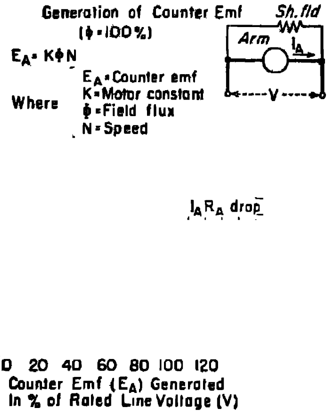


Fig. 16-5. Speed-counter emf curve for a typical d-c shunt motor.

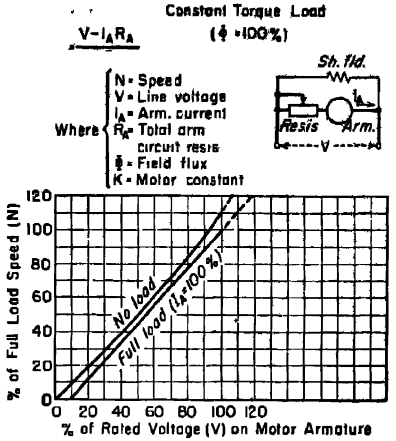


Fig. 16-6. Speed–per cent impressed armature-voltage curve for a typical d-c shunt motor.

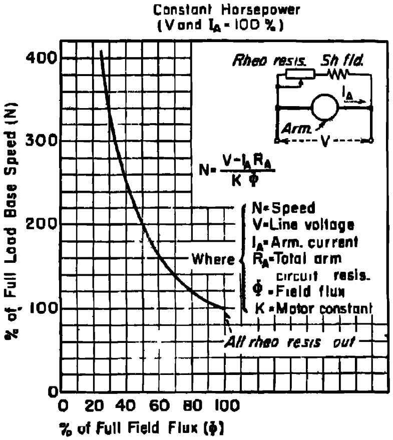


Fig. 16-7 Speed–field-flux curve of a typical d-c shunt motor.

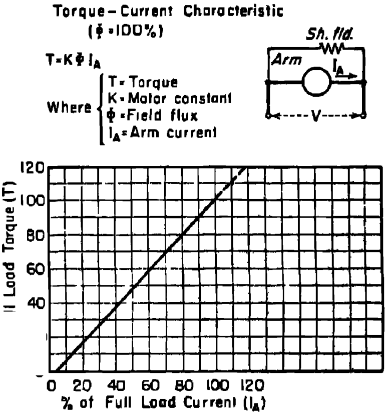


Fig. 16-8. Torque-armature-current characteristic of a typical d-c shunt motor.

D-c-motor Basic Torque Characteristic and Formula. The torque developed by a d-c shunt motor is proportional to the product of the field flux Φ and armature current I_A. The torque can be expressed by the following equation:

$$T = K\Phi I_A$$

where T = torque
K = motor constant
Φ = field flux
I_A = armature current

Most controllers are arranged so that the motor shunt field is fully excited during the starting and accelerating period. This helps the motor to develop the required torque without excessive armature current. A typical torque and armature-current curve is shown by Fig. 16-8. For constant field excitation, the torque is almost directly proportional to the armature current. The motor torque can be controlled by varying the applied armature voltage or by limiting the armature current by means of a series resistor.

Speed Adjustment by Armature-resistance Control.

Speed Range:
2:1 (50 per cent speed reduction by armature-resistance control)
Speed Decrease from No Load to Full Load:
Approximately 50 per cent at low speed
Approximately 10 per cent at base speed
Horsepower:
¼ to 50

Figure 16-9 shows the apparatus required for the simplest and least expensive d-c adjustable-speed drive. This type of controller is manually operated and provides a means for varying the resistance in the armature circuit and the voltage applied to the motor armature.

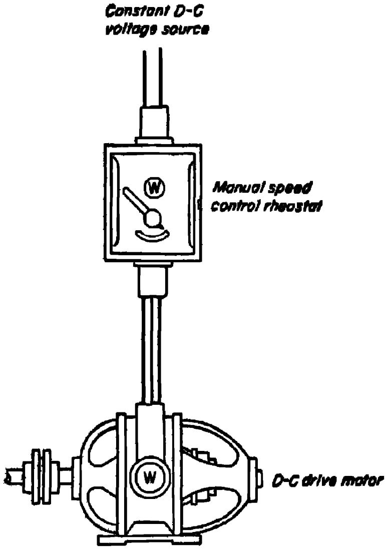


Fig. 16-9. Typical d-c drive consisting of a shunt motor and manually operated faceplate-type controller.

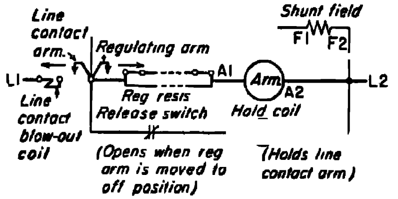


Fig. 16-10. Circuit diagram for a manual speed controller for speed adjustment by armature-resistance control.

The diagram for a typical manual controller which utilizes the apparatus illustrated in Fig. 16-9 is shown by Fig. 16-10. The motor is started by moving the control-rheostat handle in a counterclockwise direction. This movement closes the line contact, and the motor will start and accelerate to approximately 50 per cent of full-load speed if the load is correct for the resistors. A magnet energized by the "hold coil" will hold the line-contact arm in the closed position. The regulating arm may then be moved to short out the regulating resistor and to increase the speed by turning the handle in a clockwise direction. A star wheel and roller assembly on the operating-handle shaft provide positive positioning of the regulating-arm contacts on the stationary contact segments. The motor can be stopped by moving the regulating arm in the counterclockwise direction to the off position. The release switch opens just after the regulating arm is moved beyond the first or lowest speed point. Deenergization of the "hold coil" releases the line-contact arm and opens the circuit to stop the motor. As the line-contact arm is magnetically held in the closed position, this controller will provide low-voltage protection and always ensure that the motor is started with full resistance in the circuit. This type of controller usually does not provide any overload protection.

For armature-resistance control, the motor is almost always operated with full shunt-field excitation. Figure 16-11 shows typical speed-torque curves for a 10-point manual controller with a class 93 resistor for varying-torque applications. This resistor class provides 50 per cent speed reduction at not less than 35 per cent of full-load torque. The regulation of this drive is not very good as it varies from approximately 120 per cent at low speed to about 10 per cent at rated speed. Other intermediate speeds than those shown can be obtained by using more resistor steps.

Class 95 resistors are used for constant-torque applications and provide 50 per cent reduction in motor speed with not less than 80 per cent of full-load torque.

The speed–torque characteristic of a fan load is also shown on Fig. 16-11. When the line contact is closed, the motor will be accelerated on the first-point speed curve to the point indicated by *A*. When the regulating arm is moved to the second point, the motor will then increase in speed from point *B* to *C*. On the third point, the speed will change from *D* to *E*, etc.

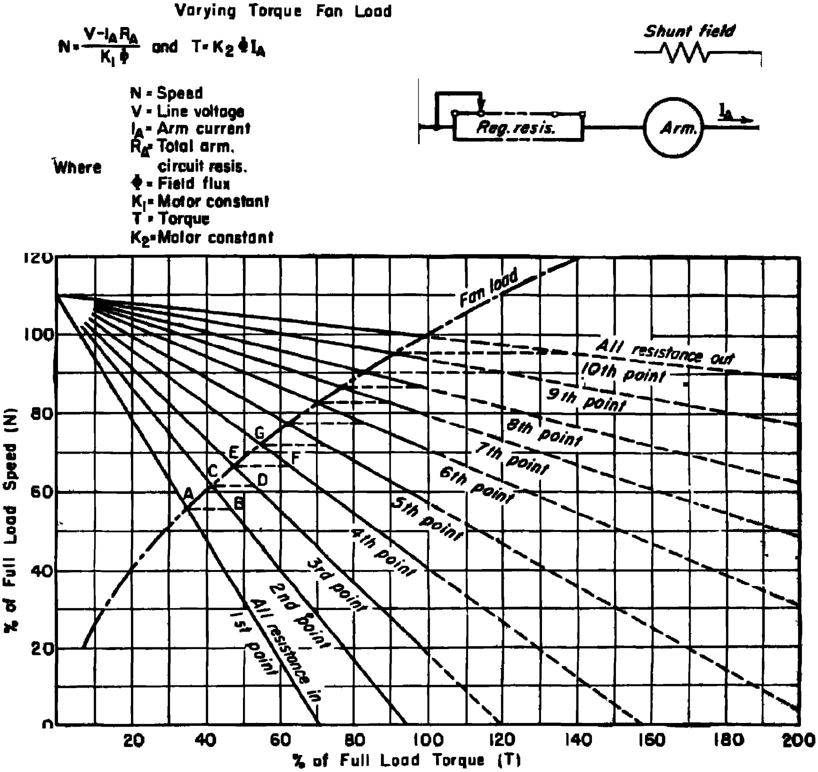


FIG. 16-11. Typical speed–torque curves for armature-resistance control of a d-c shunt motor. Class 93 resistor for varying-torque applications.

For speed reductions below 50 per cent of the rated speed (full-field excitation from rated voltage on the field), the armature is generally shunted by a resistor to improve the regulation and to reduce the differential between the no-load and full-load speeds. Figure 16-12 illustrates how the speed regulation can be greatly improved with the motor armature shunted. When the per cent of the impressed voltage on the armature is known, the per cent of base speed can be found for both no load and full load from the curve shown on Fig. 16-6.

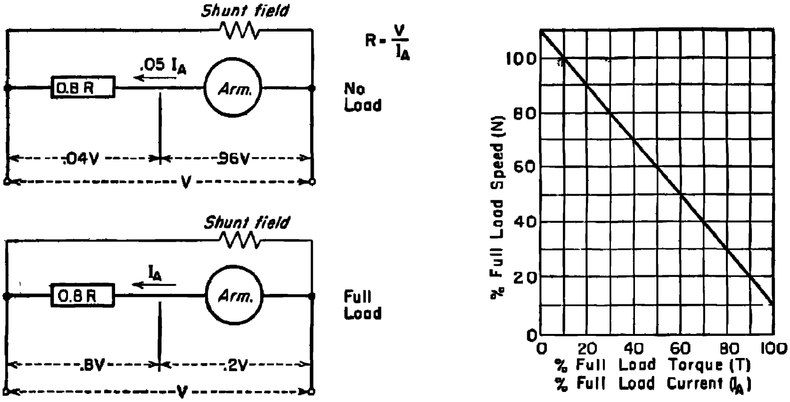
Speed Adjustment by Field-excitation Control.

SPEED RANGE:

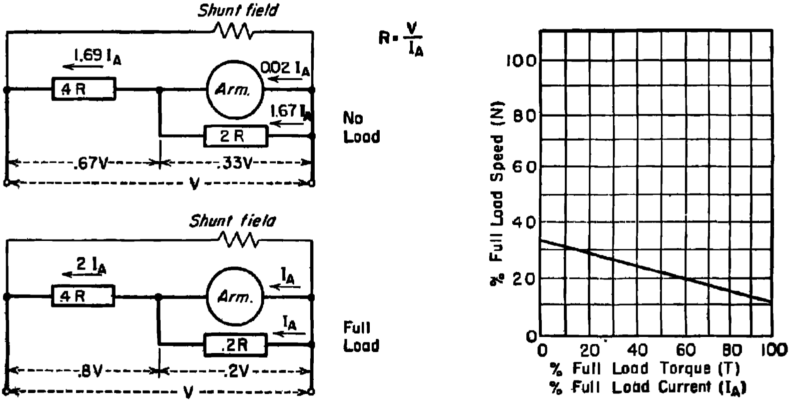
4:1 (4:1 speed increase by field control)

Speed Decrease from No Load to Full Load:
Approximately 10 per cent at base speed
Approximately 25 per cent at high speed
Horsepower:
Unlimited

An examination of the speed equation for a d-c shunt motor shows that the speed can be adjusted by varying the shunt-field excitation Φ. A rheostat in series with the field can be used to vary the field current. This in turn increases or decreases the excitation.



Armature Series Resistor



Armature Shunting Resistor

Fig. 16-12. Circuit diagrams and comparison of the regulation for a d-c shunt motor with a series or an armature-shunting resistor.

Figure 16-13 shows the apparatus required for a typical d-c drive with speed adjustment by field control.

Magnetically operated starters are advantageous over the manually operated type as they are more suitable for remote operation, interlocking, and providing overload protection. Figure 16-14 shows the circuit diagram for the drive shown in Fig. 16-13.

When the START push button is operated, the inductive accelerating contactor coil 1*A* will be energized. The main contacts 1*A* will open to permit the first step of the starting resistor to become effective, and the auxiliary contacts 1*A* will close to energize the second accelerating contactor coil 2*A*. The main contacts 2*A* will open, and the auxiliary contact 2*A* will close to energize the line-contactor coil *M*. Energization of coil *M* will close the line contacts *M*, and the motor will be started with reduced voltage on the armature.

The line-contactor auxiliary contact M_b also opens and deenergizes the coil 1*A* to initiate the time-delay period. After a time-delay period of several seconds, the main contacts 1*A* will close to short out step *R*1-*R*2 of the starting resistor. The auxiliary contacts 1*A* will also open to deenergize coil 2*A*. After another time-delay period, the main contacts 2*A* will close to short out all the starting resistor and to apply full voltage to the motor armature. The motor will continue to run until it is deenergized because of operation of the STOP push button, the overload relay, or a power failure.

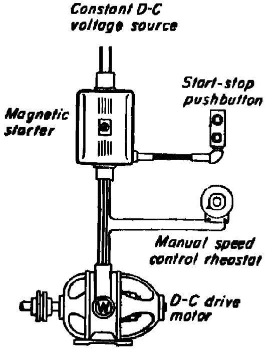


Fig. 16-13. Typical d-c drive consisting of a shunt motor, magnetic starter and push button, and a field rheostat.

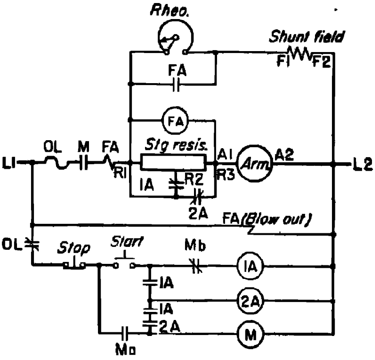


Fig. 16-14. Circuit diagram for speed adjustment of a d-c shunt motor by field control.

The field-accelerating relay *FA* is operated by a series and a shunt coil so that the motor is always started with full field. Also, full field is automatically applied when the motor current increases beyond a preset value. Full field is necessary to provide ample torque without excessive motor current for the load requirements.

After the motor has once been accelerated to a speed that corresponds to the speed for full line voltage on the motor armature, the contact *FA* in parallel with the field rheostat will open so that the rheostat will be effective to adjust the field current for speed control.

Figure 16-15 shows a family of curves for the speed-torque characteristic of a d-c drive motor with a 13-point field rheostat. Other intermediate speeds can be obtained by using a rheostat with smaller steps and a greater number of points.

The regulation is best at the base speed and increases as the speed increases. As this drive operates with a constant-horsepower characteristic, the safe drive torque available decreases as the speed increases. It should always be remembered that the motor will overheat if it is operated continuously at greater than rated full-load armature current. As the armature current I_A should not be increased above 100 per cent of the full-load value, the torque *T* value must decrease as the field flux Φ decreases for obtaining the higher speeds.

Speed Adjustment by Both Armature-resistance and Field-excitation Control.

SPEED RANGE:

8:1 (50 per cent speed reduction by armature-resistance control and 4:1 speed increase by field control)

SPEED DECREASE FROM NO LOAD TO FULL LOAD:
Approximately 50 per cent at low speed
Approximately 10 per cent at base speed
Approximately 25 per cent at high speed
HORSEPOWER:
¼ to 50

The speed adjustability of a d-c shunt motor drive can be greatly increased by using both armature-resistance and field-control means.

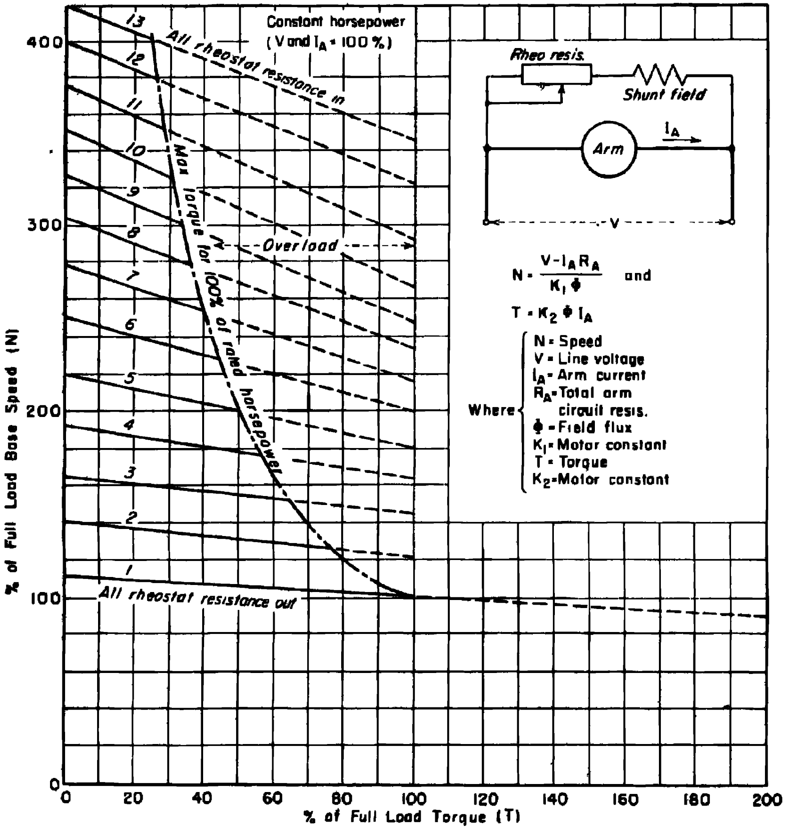


FIG. 16-15. Typical speed-torque curves for the speed adjustment of a d-c shunt motor by field control.

Figure 16-16 shows typical apparatus which can be used for this type of speed control. A drum controller functions similarly to the faceplate rheostat previously described, but it is more rugged and can be manufactured in much higher ratings than the faceplate type of starter and regulating controller. In general, the cost of a magnetic controller is higher than that for a drum type of controller. However, ease of operation, remote or automatic, and the possibility of interlocking features which are characteristic of magnetic controllers should always be considered in the selection of the apparatus for an electrical drive. Figure 16-17 shows a typical diagram for a five-point drum-type controller. The F contact on the drum opens only on the last or fifth point and ensures that the motor shunt field is always fully excited when reduced voltage is applied to the motor armature. When all the regulating resistor is shorted out, the field rheostat may be used to adjust the speed to a higher value. This particular controller does not have any provision for overload protection. This type of d-c drive falls in the constant-torque and constant-horsepower classification. The set of speed–torque curves shown in Fig. 16-18 illustrates the characteristics obtained from a drive which has five points of armature-resistance control and 17 points of field control. Intermediate speeds can be obtained by using a drum controller and a field rheostat with more points or steps. It should be noted that, for this drive, the speed regulation is best at the base speed.

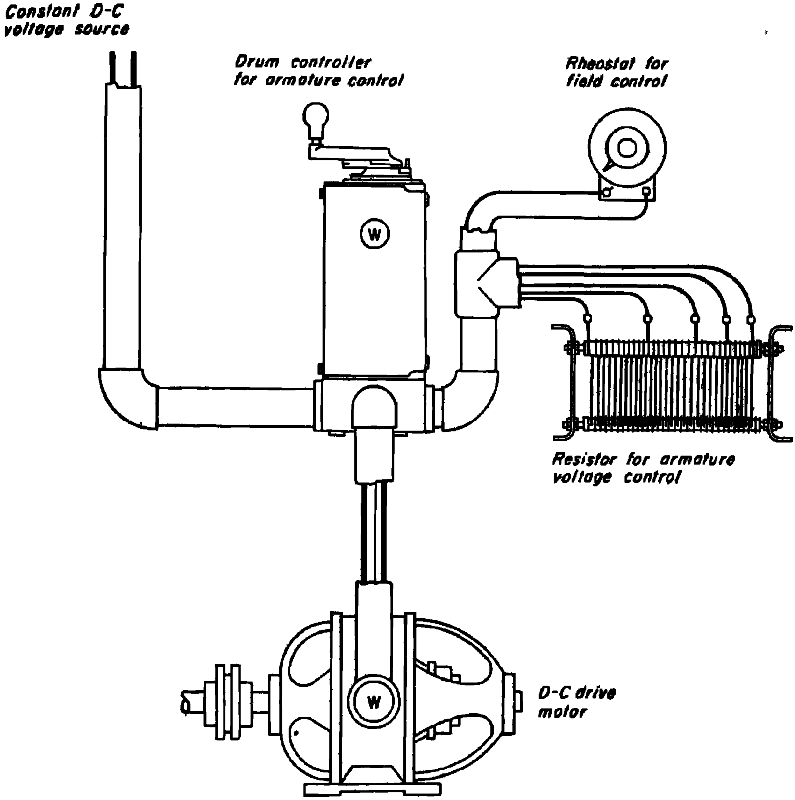

Constant D-C voltage source
Drum controller for armature control
Rheostat for field control
W
Resistor for armature voltage control
W
D-C drive motor

FIG. 16-16. Typical apparatus for a d-c shunt-drive motor with speed adjustment by armature resistance and shunt-field control.

D-c Compound-wound Motor Drives. Compound-wound motors have both shunt and series fields, and the relative strength of the two fields determines to what extent the motor operation approaches the shunt- or series-motor characteristics. The formulas previously given for the counter emf, speed, and torque of a d-c shunt motor also apply in general to a compound-wound motor. A compound-wound motor provides a relatively high starting torque which approaches that of a series motor and yet has the desirable characteristics of a safe no-load speed which is typical of a shunt-wound motor.

From no load to full load, the drop in speed or regulation of a compound-wound motor is approximately 25 per cent for operation at full shunt-field excitation.

Compound-wound motors are used extensively on applications where a reasonably constant speed is required and where a motor must develop a high starting torque to accelerate the driven machine. This type of motor is extensively used on piston pumps, conveyors, compressors not equipped with unloading devices, mining and loading machines, and other similar applications.

In general, compound-wound motors are not recommended for adjustable-speed applications, as the weakening of the shunt field to obtain a speed increase changes the proportionate ratio of the shunt and series fields. Also, the speed characteristics of a motor operated with less than full excitation on the shunt field are similar to those of a series-wound motor with the series field shunted.

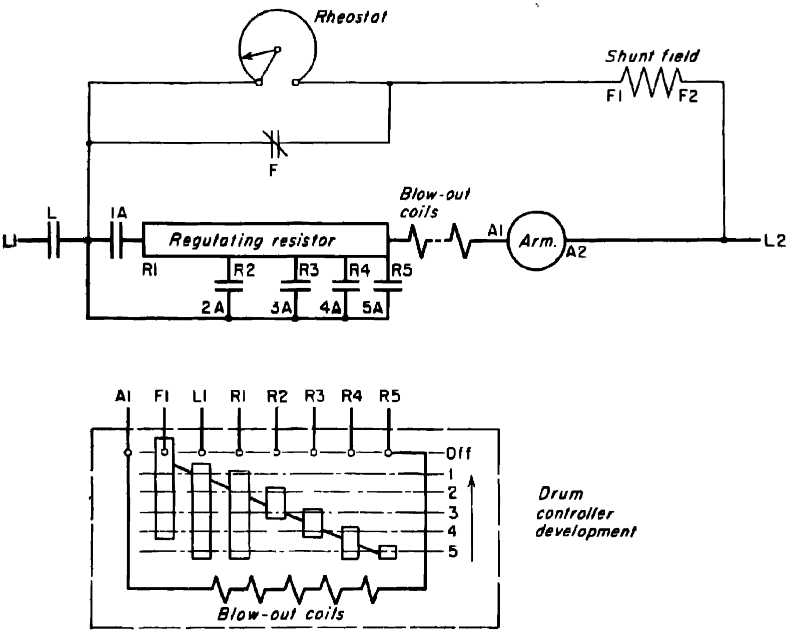


Fig. 16-17. Circuit diagram for speed adjustment of a d-c shunt motor by armature-resistance and shunt-field control.

Compound-wound motors have a speed-torque characteristic which is suitable for application to machines with flywheels or which have a high inertia-load characteristic as the motor drooping-speed trait will permit the flywheel to give up its kinetic energy if the load is suddenly applied. When a flywheel is permitted to give up its energy, the motor power requirements and the disturbances on the power supply are appreciably reduced. It also results in less heating of the motor.

Compound-wound motors can be started by the faceplate, drum, or magnetic type of controller which was previously described for shunt-wound motors.

D-c Series-wound Motor Drives. D-c series-wound motors are inherently varying-speed motors as the field winding is in series with the armature. Any change in the armature current will vary the field flux. This type of motor provides a relatively high starting torque with a relatively low starting current. Series motors are capable of starting and accelerating heavy loads with lower values of current drawn from the power supply than any other type of motor.

On light or no loads, the motor speed may be dangerously high since the series-field flux decreases as the armature current decreases. For this reason, series motors should be employed only where the load is never entirely removed from the motor or where close supervision is maintained. A series motor should never be connected by a belt to the load; it should always be connected by gears, chains, couplings, or some positive means.

Series-wound motors are applied mainly for widely varying loads, such as are typical for hoists, cranes, winches, etc., where wide speed changes are permissible and desirable. Sometimes series motors are provided with a shunt field which produces just enough field to prevent the motor from having an excessive speed on very light loads.

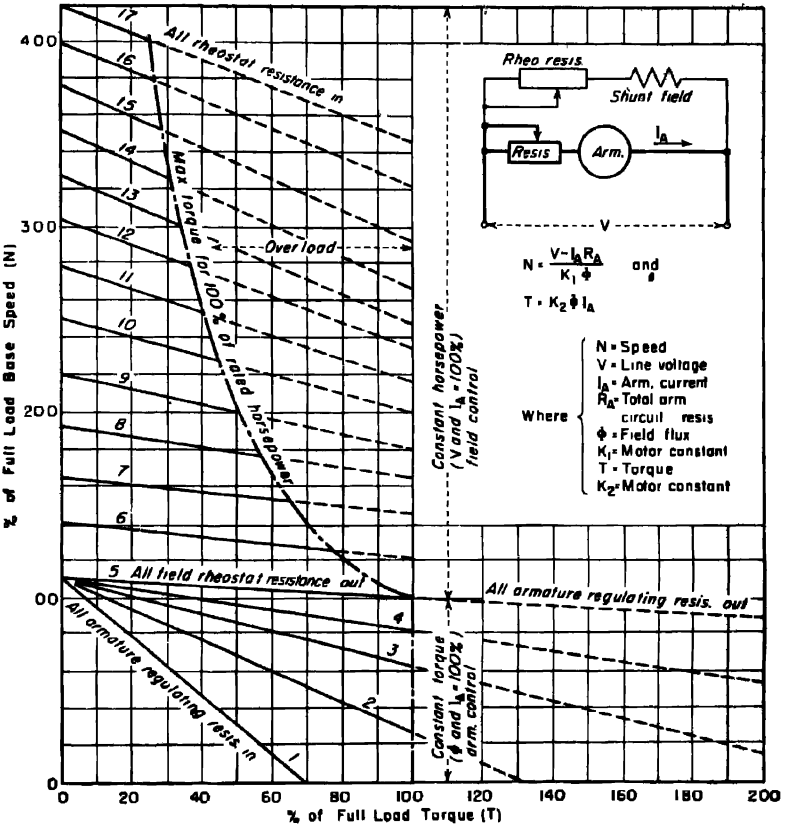


FIG. 16-18. Typical speed-torque curves for the speed adjustment of a d-c shunt motor by armature-resistance and shunt-field control.

While the general counter emf, speed, and torque equations given for d-c shunt motors also apply to series-wound motors, the only practical way to vary the speed is to vary the voltage applied to the armature and series field. Any attempt to control the series-field excitation separately would nullify the reason for using a series-wound motor in the first place.

Speed Adjustment by Armature-resistance Control.

SPEED RANGE:
Indeterminate (depends entirely on load)
SPEED DECREASE FROM NO LOAD TO FULL LOAD:
Approximately 100 per cent or more at low speed
Approximately 100 per cent or more at base speed
HORSEPOWER:
1 to 200 intermittent

Series motors are often controlled by manually operated drum controllers as that type of control provides a very rugged and compact control unit. Figure 16-19 shows the apparatus required for a typical d-c series-motor drive that is often applied to cranes or hoists.

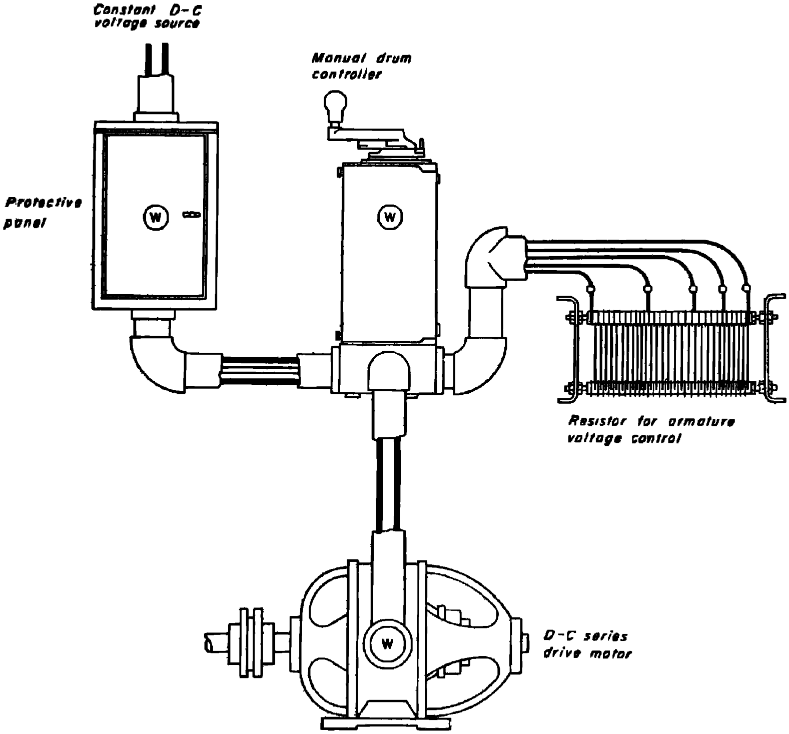

Constant D-C voltage source
Manual drum controller
Protective panel
W
W
Resistor for armature voltage control
W
D-C series drive motor

FIG. 16-19. Typical d-c series-motor drive with a protective controller.

Where ease of operation is required and a higher first cost of the controller can be justified, automatic magnetic control operated by a master switch is often used.

For some applications, such as encountered for large cranes or steel-mill drives, the motor currents that must be carried and interrupted are beyond the values that can be handled safely and efficiently by manually operated controllers; for such applications, only magnetically operated controllers are satisfactory.

Figure 16-20 shows a typical circuit diagram for a series-motor drive. For overload protection, limit-switch control, and interlocking, a protective panel with a line contactor *M* and an overload relay *OL* must be used. The line-contactor coil *M* is initially energized only in the drum OFF position through the RESET contact. The coil remains energized through its auxiliary contact M_a when the drum is moved to any other position. In case of a voltage failure or operation of the overload relay, the coil M is deenergized, and the contactor will then open the line contacts M to stop the motor. The motor can be restarted only by moving the drum controller to the OFF position. The protective panel prevents starting of the motor unless all the starting and regulating resistance is in series with the motor armature and series field.

Figure 16-21 shows typical speed–torque curves for a series-motor drive. From this set of curves it should be noted that the series motor has an excessive speed characteristic when the motor has a very small load.

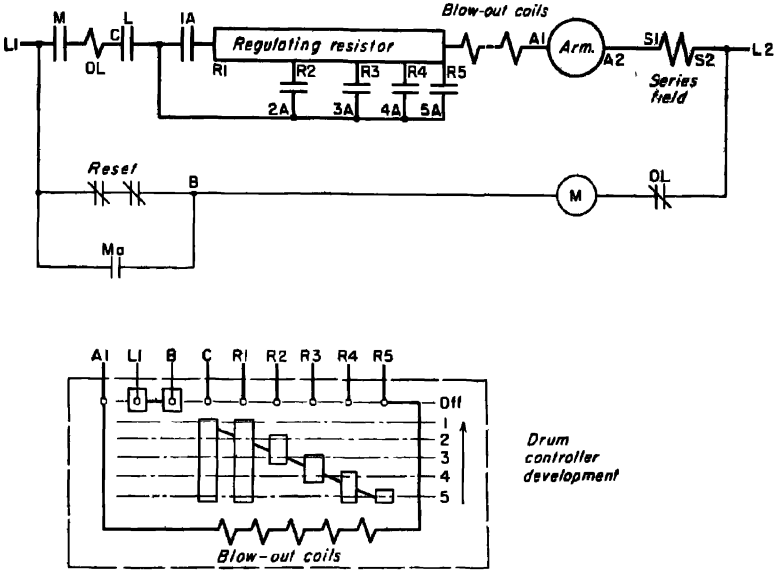


FIG. 16-20. Circuit diagram for a d-c series motor with speed adjustment by armature-resistance control.

Comparison of Typical D-c-motor Characteristics. Figure 16-22 shows the comparison of the typical speed, current, torque, and horsepower characteristics for shunt-, compound-, and series-wound motors operated at their base speeds. It should be noted that the current–horsepower curves are plotted to a different ordinate scale as they would otherwise obscure the current–torque curves. The careful consideration of this curve should be very helpful in the selection of the drive motor which has the proper speed, regulation, torque, etc., characteristics.

Graphical Analysis for the Selection of a D-c Shunt-wound Motor Drive. In selecting the proper drive for a specific application, the speed range, size, cost, regulation, etc., must be considered. Figure 16-23 illustrates how to analyze a typical application where it is desired to apply a d-c shunt-wound motor to a constant-torque load which requires a 4:1 speed range or speed adjustment from 450 to 1,800 rpm. The speed–torque and speed–horsepower requirements of the load are plotted in Col. *a* of Fig. 16-23.

Column *b* shows the application of a constant-speed shunt-wound motor with 75 per cent base-speed reduction by armature-resistance control. While this drive meets the torque requirements of the load and is the lowest in cost, it should be borne in mind that the speed regulation for this type of drive is very poor compared to other drives, and the number of speed points is limited. Standard drives with speed adjustment by armature-resistance control are usually recommended for only 50 per cent speed reduction. However, by using an armature-shunting resistor on the lower speed point and carefully selecting the motor, it is possible to obtain the desired speed reduction.

Column *c* of Fig. 16-23 illustrates the characteristics of a drive with a constant-speed motor with speed adjustment by both armature-voltage and field control. From

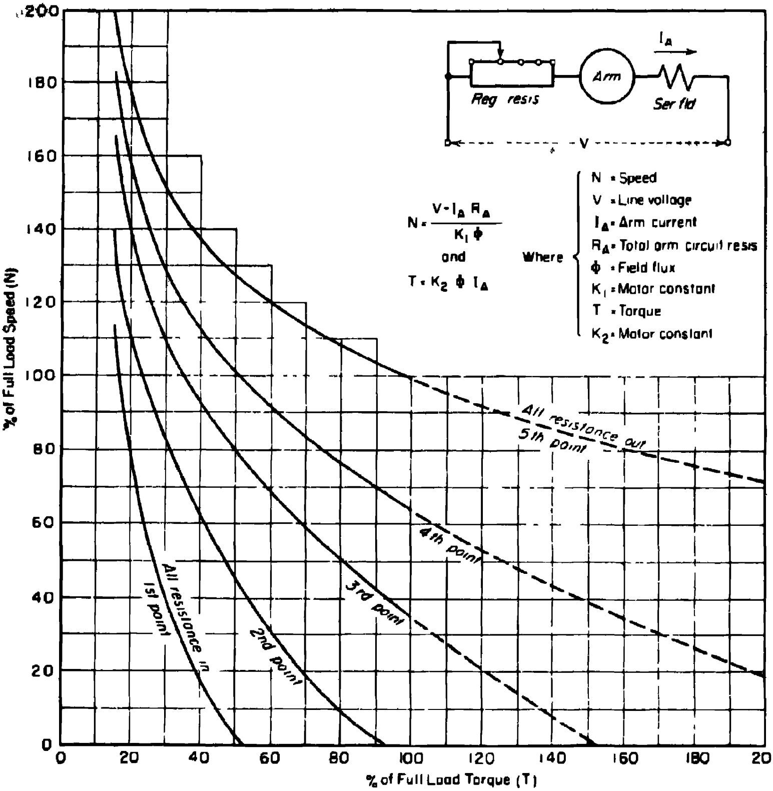


FIG. 16-21. Speed–torque curves for a d-c series motor with speed adjustment by armature-resistance control.

the curve it should be noted that the armature-regulating resistor needs capacity for only 40 per cent of the rated horsepower at the base speed.

Column *d* shows a drive which has an adjustable-speed motor that will provide a 4:1 speed adjustment by control of the shunt-field excitation only. This drive gives a fairly good speed regulation and provides a relatively large number of points. The cost of this particular drive is considerably higher than the other two, and it should be noted that the full horsepower rating of the motor can be utilized only when the motor is operated at the maximum speed required by the load. The weight and size of this particular drive are larger than those of the other drives considered.

As a general rule, it should be remembered that, the higher the shunt-motor base speed, the smaller the motor-frame size and the lower the motor cost. It is recommended that a graphical analysis similar to the one discussed always be made in order to obtain the most economical and satisfactory drive.

A-C MOTOR DRIVES—A-C POWER SOURCE

The electrical energy distributed in the United States today is predominantly 60-cycle alternating current. For this reason it is generally more economical to apply a standard a-c motor and controller to the application if it provides suitable drive characteristics.

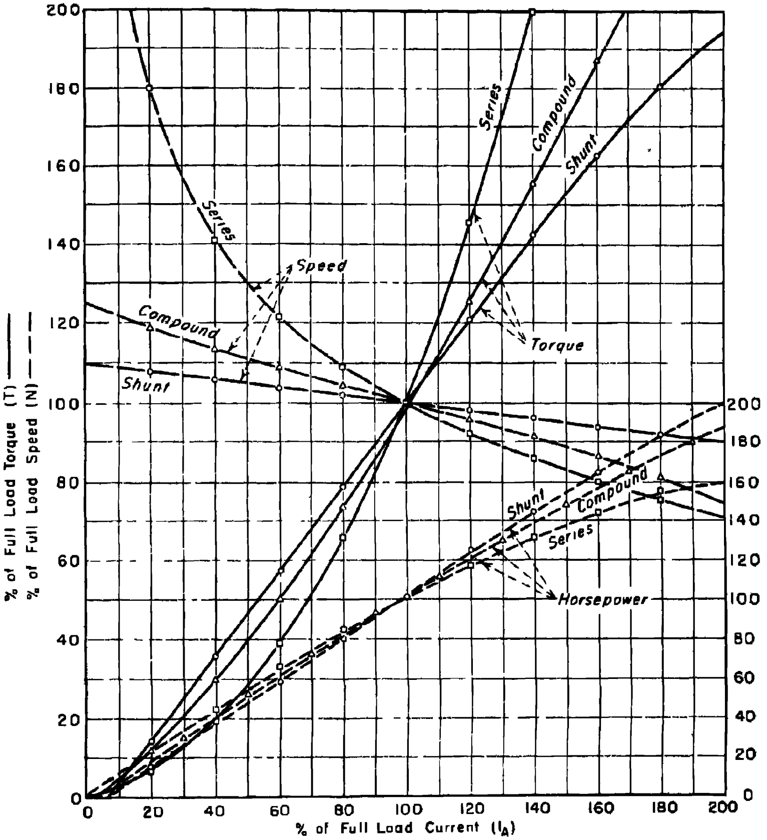


FIG. 16-22. Comparison curves of the d-c shunt-, compound-, and series-wound-motor characteristics.

A-c-motor Basic Speed Formula. The speed of an a-c motor is determined by the number of poles in the field winding and by the frequency of the applied voltage to these windings. The basic equation for the "synchronous" or maximum theoretical speed in terms of the number of poles and the frequency can be expressed by the following formula:

$$N = KF/P$$

where N = synchronous speed
K = constant
F = frequency of power supply
P = number of poles

When the speed N is expressed in rpm and the frequency F in cycles per second, the constant K is equal to 120.

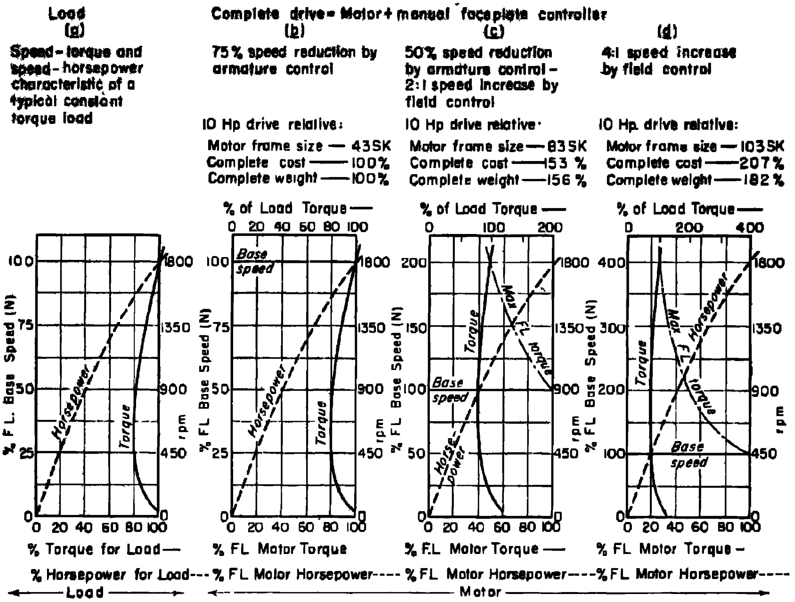


FIG. 16-23. Graphical analysis of three different speed-control methods for a typical constant-torque load.

Except for an a-c synchronous motor, the synchronous speed is a theoretical speed which a motor could attain only if it had no losses or load. An a-c induction motor can never reach synchronous speed as there is no torque developed when the rotor revolves at exactly the same speed as the rotating field. The difference between the synchronous speed and the actual speed for the torque that is required to drive the motor and the load is called the "slip." The speed of an induction motor is also determined by the load requirements and will be a speed at which the motor-torque output and the load-torque requirements are exactly in equilibrium.

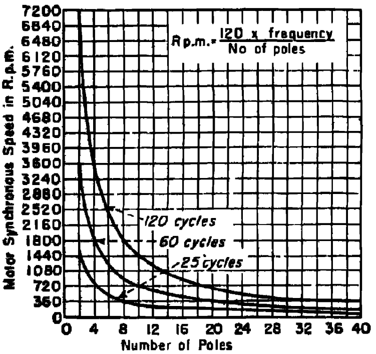


FIG. 16-24. Relationship between speed and the number of poles for a-c motors operated at given frequencies.

As shown by the speed equation, the motor speed can be varied by changing the frequency of the power supply or the number of poles in the field winding. It should always be borne in mind that an a-c motor must always have an even number of poles (same number of North as South poles), and the minimum number of poles is two. Figure 16-24 graphically shows the relationship between the speed and the number of poles at given frequencies of 25, 60, and 120 cycles.

Figure 16-25 graphically shows the speed of a motor with a given number of poles for frequencies up to 120 cycles. It should be noted that, for a speed change by adjustment of the frequency, the change in speed per cycle is greatest for a two-pole motor and least for a forty-pole motor.

A more complete table of the synchronous speeds for various pole and frequency combinations is given by Fig. 16-26.

A-c-motor Basic Torque Formula. In general and except for some conditions of synchronous-motor operation, the torque developed by an a-c motor is proportional to the square of the impressed voltage V applied to the stator or field windings. The approximate torque equation can be expressed as follows:

$$T = KV^2$$

where T = torque
K = motor design and conversion constant
V = voltage applied to the field or stator windings

The speed of an a-c induction motor can be controlled to some extent by adjusting the voltage applied to the field windings. As a variation in the applied voltage will change the torque output of the motor, the speed of the motor must change for a given load. This method of speed control is not too satisfactory for general application, as a reduction in the applied voltage may result in a very low breakdown torque. For some single-phase fan applications, this method of control is quite widely used and satisfactory, as the load characteristic is such that the horsepower requirements decrease rapidly with a decrease in speed.

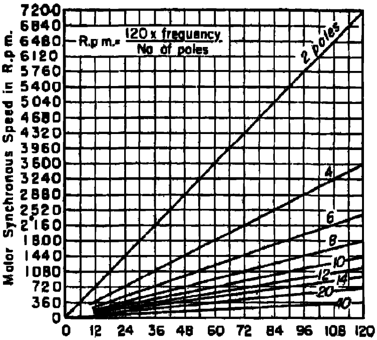


Fig. 16-25. Relationship between speed and the frequency applied to an a-c motor with a given number of poles.

No. of poles	Motor Synchronous Speeds in R.p.m. for Frequencies of							
	25~	50~	60~	90~	120~	180~	360~	420~
2	1500	3000	3600	5400	7200	10800	21600	25200
4	750	1500	1800	2700	3600	5400	10800	12600
6	500	1000	1200	1800	2400	3600	7200	8400
8	375	750	900	1350	1800	2700	5400	6300
10	300	600	720					
12	250	500	600					
14	214	429	514					
16	188	375	450					
18	167	333	400					
20	150	300	360					
22	136	273	327					
24	125	250	300					
26	115	231	277					
28	107	214	257					
30	100	200	240					
32	94	188	225					
36	83	167	200					
40	75	150	180					

Fig. 16-26. Motor synchronous speeds for various frequencies and pole combinations.

A-c Squirrel-cage Induction-motor Drives. Alternating-current squirrel-cage induction motors are considered to be the simplest and sturdiest of the electrical rotating devices that are now manufactured.

In the field of integral-horsepower motors, the polyphase induction motor is predominantly the most widely used type of motor applied to constant-speed industrial drive applications. Also, polyphase a-c power is usually the only type of power available at the majority of the industrial sites.

In the field of fractional-horsepower motors, the single-phase induction motor is the most widely used type of motor. In general, polyphase power is not usually available at the places where the majority of the fractional-horsepower motors are applied.

Squirrel-cage induction motors are essentially constant-speed motors, and in general, no relatively inexpensive control methods are known by which a reliable speed change and control may be obtained on a motor of a relatively large size. About the only practical means available for varying the speed of a relatively small induction motor without changing the number of poles or the frequency of the power supply is to adjust the voltage applied to the motor field and permit the load to govern the speed at which the motor will then operate with a change in the output torque.

Speed Adjustment of a Single-phase Squirrel-cage Capacitor Motor by Primary Voltage Control (Autotransformer).

Speed Range:
  Approximately 2:1 or 3:1 (50 to 33 per cent speed reduction by voltage control)
Speed Decrease from No Load to Full Load:
  Approximately 70 to 50 per cent at low speed
  Approximately 5 per cent at base speed
Horsepower:
  Fractional

Figure 16-27 shows the apparatus required for a typical single-phase drive motor for a fan application. The control for this drive is manually operated and provides a means for varying the voltage applied to the motor.

The diagram for the drive illustrated in Fig. 16-27 is shown in Fig. 16-28. When the line contact M is closed, the motor will be connected to a tap on the autotransformer. The speed for a particular voltage will be determined directly by the load.

From the speed-torque curves shown in Fig. 16-29, it should be noted that the curve for the lowest value of voltage applied to the motor field windings has a very steep characteristic and a slope similar to the load at the lower speeds. The speed at which the motor and the load torques are in equilibrium will be very indefinite unless the load requirements are known quite accurately. The set of curves in Fig. 16-29 illustrates how four different fan-operating speeds can be obtained quite efficiently. This type of drive is successful only because the fan-load characteristics are such that the torque requirements fall off quite rapidly with a decrease in speed.

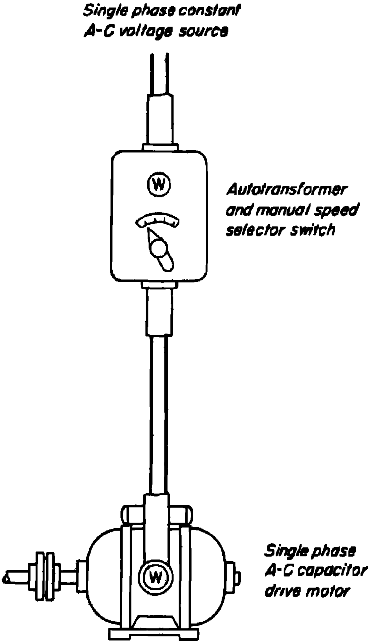


Fig. 16-27. Single-phase a-c capacitor-motor drive with speed adjustment by primary voltage control.

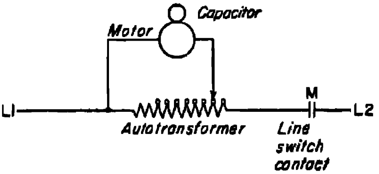


Fig. 16-28. Circuit diagram for a single-phase manually operated controller for speed adjustment by primary voltage control.

Typical Three-phase A-c Squirrel-cage Drive-motor Characteristics. For a given a-c squirrel-cage motor, the speed-torque characteristic is determined primarily by the rotor design and resistance. Linestart Class I and II motors operate at a relatively constant speed and differ primarily in the starting torque that each can develop.

High-torque high-slip motors have a drooping speed-torque characteristic which can be very effectively utilized for providing automatic speed control for applications where the load is pulsating and has load peaks of very large magnitude. As the motor speed automatically decreases quite rapidly with an increase in the load, the flywheel or mass of the driven machinery is permitted to give up its kinetic energy to the load. As the energy is supplied by the inertia of the rotating system, the motor-current peaks and power-supply disturbances are reduced to a minimum when this type of motor is applied to a pulsating load such as encountered in a punch press, crusher, etc. A normal or low-slip (Class I or II) motor is not generally satisfactory as a high-torque

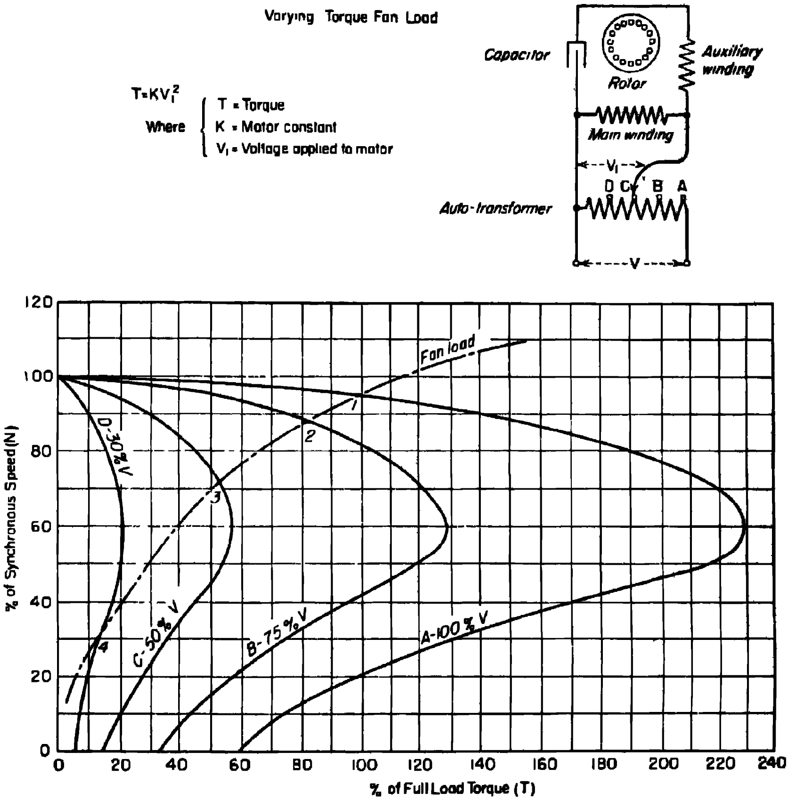


FIG. 16-29. Typical speed-torque curves for primary voltage control of a single-phase capacitor motor.

cyclic load would also probably seriously overheat the motor. These characteristics are shown in Fig. 16-30.

This example illustrates why it is very important to select the most suitable motor, as automatic speed control can be obtained very simply and at a relatively low cost.

Speed Adjustment of a Three-phase A-c Squirrel-cage Drive Motor by Primary Voltage Control (Primary Resistor).

SPEED RANGE:

From full speed to 0 speed with operation at very indefinite speeds

SPEED DECREASE FROM NO LOAD TO FULL LOAD:

Approximately 100 per cent at low speed

Approximately 3 to 4 per cent at high speed

HORSEPOWER:

Not practical except for very low ratings and variable-torque loads

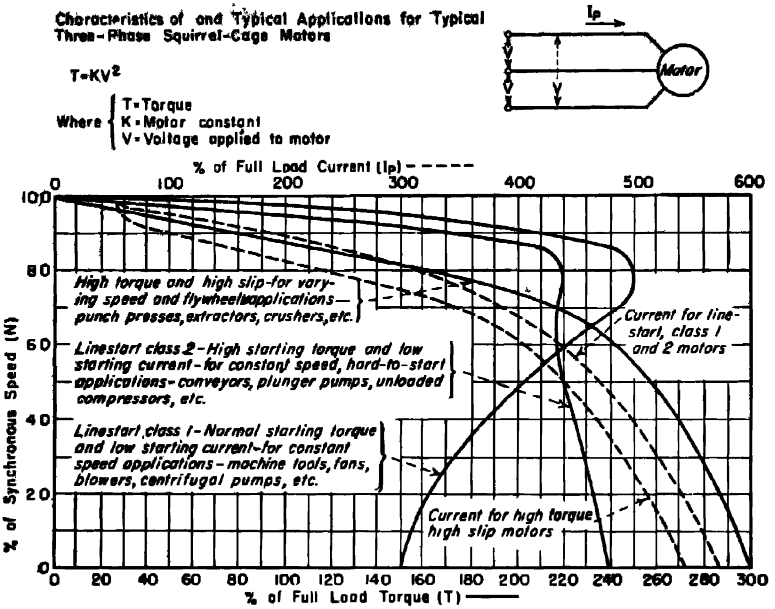


FIG. 16-30. Typical speed–torque and speed–current curves for three-phase a-c squirrel-cage motors.

The output torque of a squirrel-cage induction motor can be controlled by inserting resistance in each phase of the primary circuit to reduce the voltage actually applied to the field windings. Figure 16-31 shows a three-phase squirrel-cage motor drive with a manually operated primary-resistor controller for adjusting the motor torque.

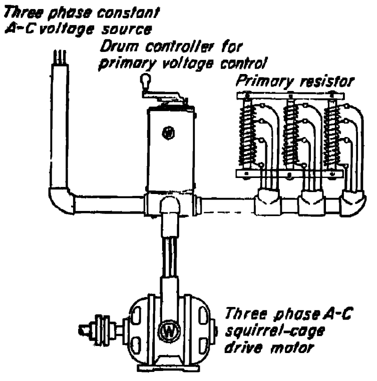

Three phase constant A-C voltage source
Drum controller for primary voltage control
Primary resistor
Three phase A-C squirrel-cage drive motor

FIG. 16-31. Three-phase a-c squirrel-cage motor drive with speed adjustment by primary voltage control.

The circuit diagram for a primary-resistor controller with a manually operated drum is shown by Fig. 16-32. When the drum-controller handle is moved to the first point, the motor is energized with all the resistance in series with each phase. The speed will be determined only by the torque requirements of the load.

Figure 16-33 shows typical speed–torque curves for a four-point manual controller. It should be noted that, for this particular resistor design, the speed can be varied only in a very small range, or from approximately 75 to 95 per cent for a varying-torque load.

For the constant-torque load shown, the motor can be started only on the fourth point or with full voltage on the motor. After the motor is once started, the speed can be reduced to only 90 per cent of the synchronous-speed value by moving the drum back to the second point.

This particular type of controller was illustrated only to show that, for general application, this means of speed control is very ineffective and not satisfactory. This type of control also has another undesirable feature, as the drum controller must be capable of carrying and interrupting relatively large primary currents.

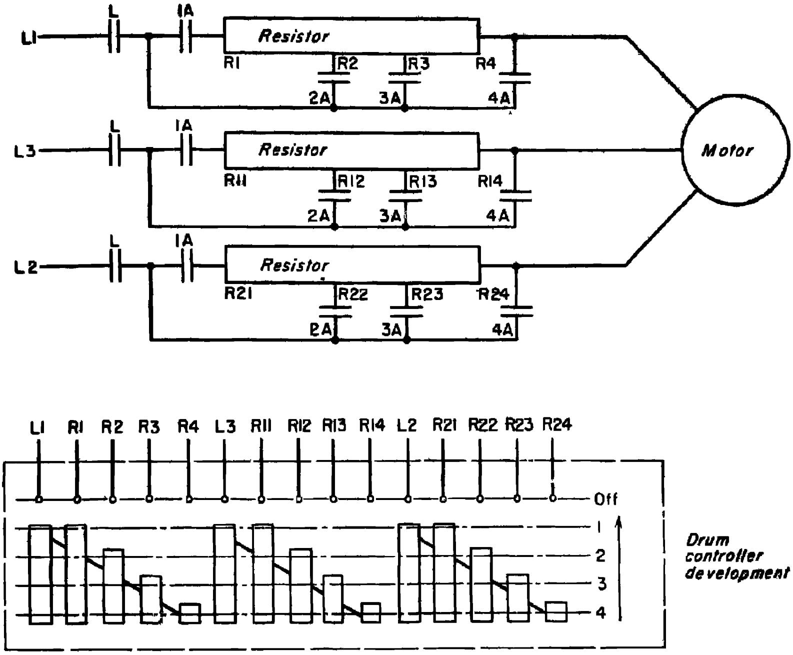

L1
L3
L2
L
1A
Resistor
R1
R2
R3
R4
2A
3A
4A
R11
R12
R13
R14
R21
R22
R23
R24
Motor
L1 R1 R2 R3 R4 L3 R11 R12 R13 R14 L2 R21 R22 R23 R24
Off
1
2
3
4
Drum controller development

FIG. 16-32. Circuit diagram for a three-phase manual speed controller for speed adjustment by primary voltage control.

Speed Adjustment of Three-phase A-c Squirrel-cage Multispeed Drive Motors by Pole-changing Control.

SPEED RANGE:
  Single-winding motors 2:1
  Multiple-winding motors 2:1, 3:1, 4:1, etc.
SPEED DECREASE FROM NO LOAD TO FULL LOAD:
  Approximately 3 to 4 per cent
HORSEPOWER:
  Unlimited

The basic speed equation given for a-c motors also shows that, for a given frequency, the speed can be varied by changing the number of poles in the motor field windings. Multispeed motors are constructed so that it is possible to reconnect the windings to provide two or more pole combinations.

Figure 16-34 shows typical connection diagrams for two- and four-speed motors. It is not practically possible to obtain any speed adjustment between the synchronous speeds for the various pole combinations, but this type of motor is widely applied to fans, machine tools, etc., where only two, three, or four different speeds are entirely satisfactory.

It should be noted that three speeds can be obtained by using only three pole combinations of a four-speed motor assembly. On motors with two or three separate windings, all the individual windings that are not in use to obtain a certain pole combination are generally open-circuited to prevent circulating currents from being induced in those windings.

For two-speed motors with single windings, the ratio between the high and low speeds is always 2:1. Standard two-speed two-winding motors can have almost any high-speed to low-speed ratio, such as 1.33:1, 1.6:1, 2:1, 2.66:1, 3:1, 4:1, etc.

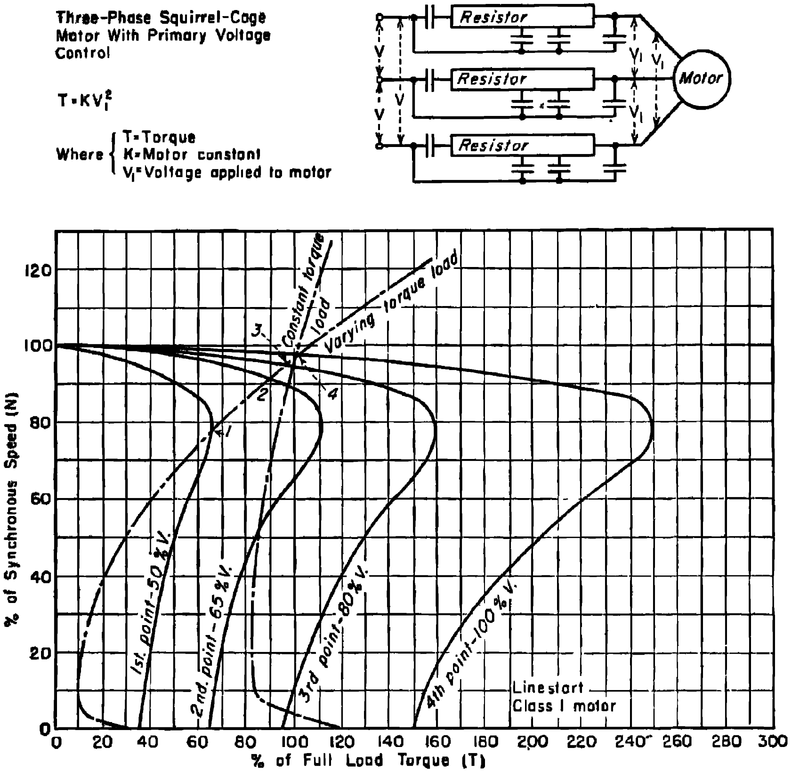


Fig. 16-33. Typical speed–torque curves for a three-phase squirrel-cage motor operated with resistance in the primary circuit.

Three- and four-speed motors can have numerous pole combinations which will provide a relatively large range in the speed ratios.

For the most satisfactory selection of a multispeed drive motor, the exact characteristics of the load must be accurately known. A comparison of the horsepower ratings and physical size of 10-hp constant-horsepower, constant-torque, and variable-torque motors is shown in Fig. 16-35. A variable-torque motor cannot be applied to a constant-horsepower load as it would overheat at the lower speeds. A constant-horsepower motor could be applied and could handle any of the three types of loads, but it would be very uneconomical for constant- and variable-torque loads as the motor would be much larger in size than necessary, have excess thermal capacity at the lower speeds, and cost considerably more than the motor that has just the right capacity for the load.

Figure 16-36 shows a typical multispeed motor drive with a manually operated speed-selector station and a magnetically operated pole-changing starter.

Manually operated drum switches are also available for multispeed-motor control. This type of control does not generally provide any means for interlocking or overload protection.

Typical two-speed magnetic-starter diagrams are shown in Fig. 16-37. For these particular circuits the motor can be started and operated for either speed depending upon the push button operated. When a push button is operated, its respective speed-contactor coil is energized, and the main contacts close to start the motor to operate ultimately at the desired speed. It is not necessary in this circuit to hold the push button down as an auxiliary interlock contact will close to provide continuous energization of the coil when the push button is released. The push buttons are interlocked so that operation of the other speed push button will deenergize one contactor coil and energize the other coil for the desired speed.

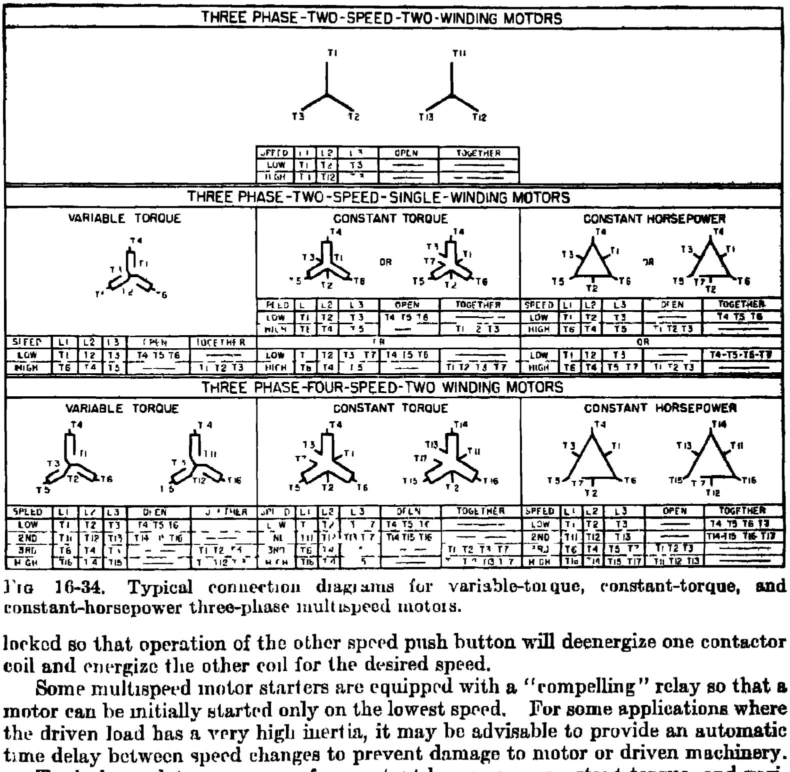


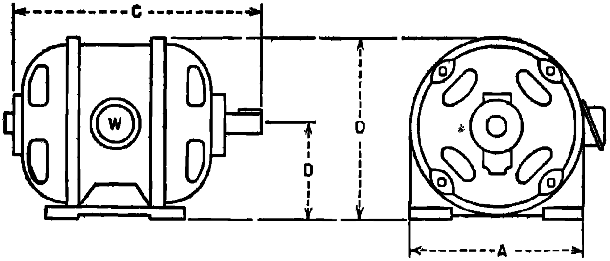
Fig 16-34. Typical connection diagrams for variable-torque, constant-torque, and constant-horsepower three-phase multispeed motors.

Some multispeed motor starters are equipped with a "compelling" relay so that a motor can be initially started only on the lowest speed. For some applications where the driven load has a very high inertia, it may be advisable to provide an automatic time delay between speed changes to prevent damage to motor or driven machinery.

Typical speed-torque curves for constant-horsepower, constant-torque, and variable-torque motors are shown in Fig. 16-38. It should be noted that the full-load torque for the lowest speed of a constant-horsepower motor must be twice the value of the high-speed torque. It is also two times the value of that for the low speed of a constant-torque motor and four times that of a variable-torque motor. The curves in Fig. 16-38 are based on motors that have the same rating at the highest speed and have a high- to low-speed ratio of 2:1. For motors with two separate windings, the ratio between the high and low speeds does not necessarily need to be 2:1.

Bearing in mind that the hp = TN/K (where hp = horsepower, T = torque, N = speed, and K = a conversion constant), this set of curves graphically illustrates the data tabulated in Fig. 16-35.

A-c Wound-rotor Induction-motor Drives. The limited capacity of the power systems in the early days when induction motors were first developed and applied forced the extensive use of wound-rotor motors to keep the starting currents to a minimum. As squirrel-cage motors were developed and perfected and the capacity of the power systems was greatly increased, the squirrel-cage type of induction motor supplanted the wound-rotor type of motor for the majority of the a-c-motor applications.



3ϕ-60∼-40°C. CONT-OPEN-MULTISPEED SQUIRREL-CAGE MOTORS											
TYPE	NO. OF SPEEDS	TYPE OF WINDING	HP AT SYNCHRONOUS RPM OF				FRAME NO.	DIMENSIONS			
			1800	1200	900	600		A	C	D	O
CONSTANT HORSEPOWER	2	Single	10		10		365	18	29 1/2	9	18 1/8
	2	Double	10	*(10)	10		404	20	31 3/4	10	19 15/16
	4	Double	10	10	10	10	444	22	36 1/2	11	21 15/16
CONSTANT TORQUE	2	Single	10		5		324	16	25 3/4	8	16
	2	Double	10	*(6 6)	5		326	16	27 1/4	8	16
	4	Double	10	6 6	5	3 3	365	18	29 1/2	9	18 1/8
VARIABLE TORQUE	2	Single	10		2 5		324	16	25 3/4	8	16
	2	Double	10	*(4 4)	2.5	*(1.1)	326	16	27 1/4	8	16
	4	Double	10	4 4	2 5	1 1	326	16	27 1/4	8	16

* Motors with two windings can have other than a 2 1 ratio between the high and low speeds.
This rating obtainable for other than a 8 pole winding on this same frame size

FIG. 16-35. Comparison of the horsepower ratings at various speeds and the size for typical constant-horsepower, constant-torque, and variable-torque 10-hp multispeed motors.

However, there are still several types of applications where the wound-rotor type of motor is preferable over any other type of a-c motor. First, the wound-rotor induction motor is well adapted to applications where particularly high starting torques and low starting currents are required simultaneously. Pumps, compressors, and other continuous-duty and constant-speed applications are often driven by wound-rotor motors.

The second major type of application for the wound-rotor induction motor is where speed adjustment is required. This type of motor is often applied for driving fans and conveyors that require speed adjustment and when only a-c power is available. With special control equipment, wound-rotor motors have been very successfully applied to cranes and hoists.

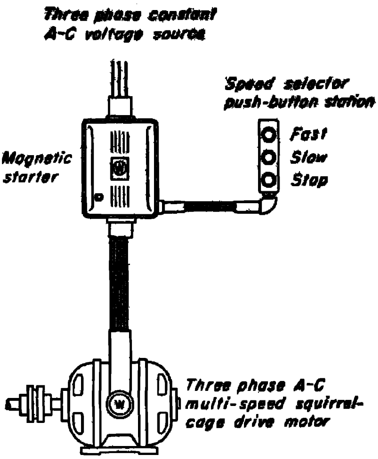
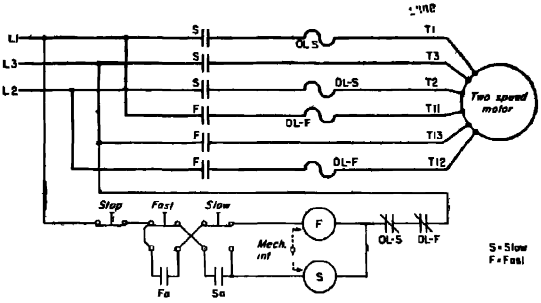
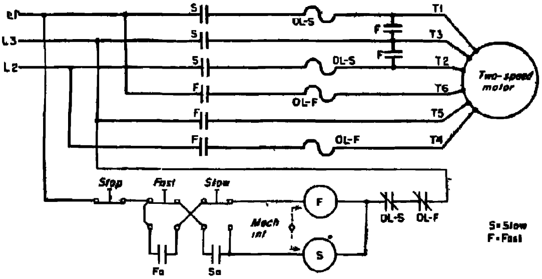

Three phase constant A-C voltage source
Speed selector push-button station
Fast
Slow
Stop
Magnetic starter
Three phase A-C multi-speed squirrel-cage drive motor

FIG. 16-36. Typical a-c multispeed squirrel-cage motor drive with speed adjustment by pole-changing control.



Two-speed two-winding motor



Two-speed single-winding variable or constant torque motor

FIG. 16-37. Typical diagrams for a two-speed motor with speed adjustment by pole-changing control.

The inherent characteristics of a wound-rotor induction motor are such that, with resistance in the secondary circuit, the driven load will directly determine the motor speed. The motor slip depends almost entirely upon the load, and if the load is removed, the motor speed will approach the maximum synchronous value. In general, wound-rotor motors should not be applied where it is expected that very low speeds at very light loads will be required. Under these conditions the operation may not be very satisfactory unless a very costly investment is made in special resistors; for many applications this extra cost is not practicable or feasible. Usually, a

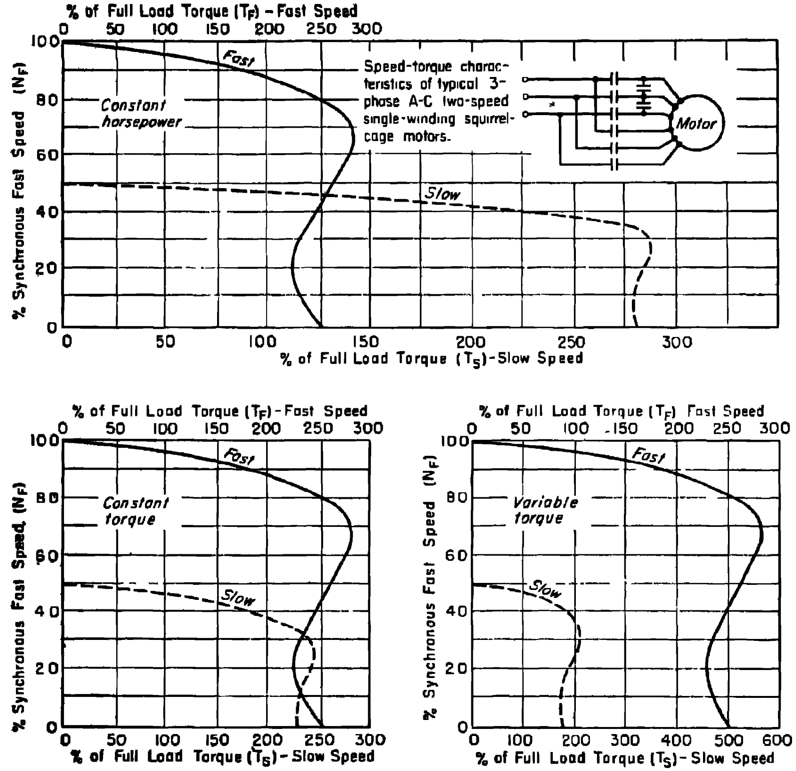


FIG. 16-38. Comparison of the speed-torque characteristics of typical multispeed squirrel-cage motors.

50 per cent speed reduction is considered to be the maximum reduction that should be required from this type of motor if stable operation is desired. Special control equipment, artificial electrical loading means, etc., are sometimes used to obtain stable operation at low speeds and useful light loads.

Speed Adjustment of a Three-phase Wound-rotor Induction Motor by Secondary Resistance Control.

SPEED RANGE:

Approximately 2:1 (50 per cent speed reduction by secondary resistance control)

SPEED DECREASE FROM NO LOAD TO FULL LOAD:

Approximately 50 per cent at low speed

Approximately 5 per cent at base or high speed

HORSEPOWER:
Unlimited

A manually operated starting and speed-regulating rheostat is generally the simplest type of controller for a wound-rotor motor. Figure 16-39 shows a typical wound-rotor drive motor with a magnetically operated primary starter and a manually operated secondary controller. Motor-operated drums, magnetically operated contactor controllers, or liquid slip regulators are generally used for the relatively large wound-rotor motors.

The circuit diagram for a typical manually operated starter and speed-regulating rheostat is shown in Fig. 16-40. The rheostat is arranged so that the motor is always started with all the resistance in the secondary circuit. A control contact on the faceplate arm momentarily provides a circuit for energizing the primary-starter contactor coil. Energization of this coil will close the contacts *M* and energize the primary circuit of the motor. The motor will then operate at a speed depending upon the load and the value of the resistance in the secondary circuit.

This circuit will provide low-voltage protection and make it necessary for the controller always to be returned to its OFF position before the motor can be restarted in case of an overload or voltage failure. Normally, the motor is stopped by returning the controller handle to the OFF position which will deenergize the line-starter coil *M*.

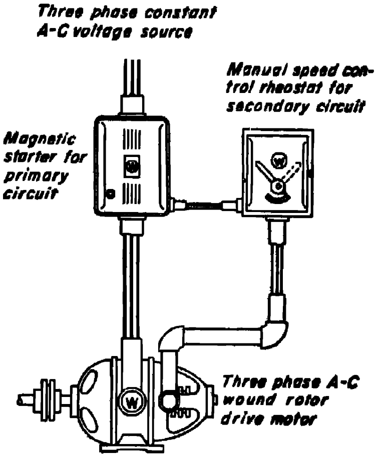
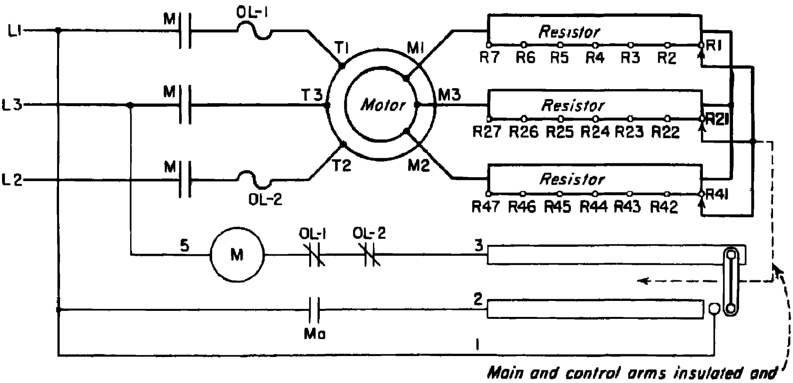

Three phase constant A-C voltage source
Manual speed control rheostat for secondary circuit
Magnetic starter for primary circuit
Three phase A-C wound rotor drive motor

FIG. 16-39. Typical a-c three-phase wound-rotor motor drive with speed adjustment by secondary resistance control.



Main and control arms insulated and mechanically connected together.

FIG. 16-40. Circuit diagram for a three-phase wound-rotor drive motor with secondary resistance control.

A typical set of speed–torque curves for a seven-point secondary resistor controller is shown by Fig. 16-41. These curves are similar to those which would be obtained from a resistor designed according to the specifications for a NEMA class 93 resistor for varying-torque applications. With these resistor values, on the first point the motor will develop approximately 35 per cent of its rated torque and will operate at approximately 50 per cent of synchronous speed if the load is correct.

For some applications where "softer" starting and a finer speed control are required, special control is sometimes arranged so that one phase of the secondary circuit is opened. For single-phase operation, the motor will develop only about 50 per cent of the three-phase starting torque.

It should be noted that, within certain operating regions, the speed–torque curves of a wound-rotor motor with secondary resistance control are very similar to those of the d-c shunt motor operated with resistance in the armature circuit (see Fig. 16-11).

A-c Synchronous-motor Drives. An a-c synchronous motor operates at absolutely synchronous speed. This characteristic makes it indispensable for many applications that must be operated at a constant speed. However, there are no very simple or inexpensive means available for varying the speed.

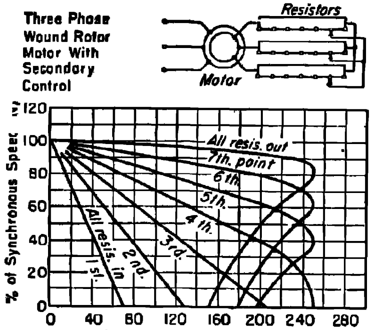


FIG. 16-41. Typical speed–torque curves for a three-phase wound-rotor motor with secondary resistance control.

Synchronous motors also operate according to the same unescapable relationship expressed by the speed equation previously given for a-c motors. The synchronous speed can be changed only by adjusting the frequency or changing the number of poles. Unless a number of synchronous motors are required for a particular application, it is not practical to provide an adjustable-frequency power system. Synchronous motors are sometimes arranged so that the windings can be connected for several different pole combinations.

The synchronous type of a-c motor is quite widely applied today as, in addition to providing a constant speed, it can also provide some power-factor correction for a power system.

Compared to the other starters for various types of a-c motors, the synchronous-motor controller is a little more complicated as it must control and synchronize both the a-c and d-c circuits.

A-C AND D-C UNIVERSAL-MOTOR DRIVES—A-C OR D-C POWER SOURCES

The universal motor is a series-wound and commutator type of motor which can be operated on either a-c or d-c power supplies. The performance and operating characteristics are essentially the same when it is operated on either type of power supply. The speed–torque characteristic curve is similar to that for a d-c series-wound motor.

This type of motor provides a relatively large starting torque and may have a no-load speed as high as 10,000 or 20,000 rpm. It is customarily built in fractional-horsepower sizes, and the armature is designed and constructed in such a manner that it can be operated at these high speeds safely. Some degree of speed control can be obtained on this motor by a series resistor or by the use of a tapped series field.

This type of motor has found wide application in electric adding machines, cash registers, drills, hammers, vacuum cleaners, motion-picture projectors, sewing machines, and other built-in applications where the motor housing is an integral part of the machine or appliance.

Universal motors have laminated armatures and stators so that they will operate satisfactorily on either a-c or d-c sources. For a given motor assembly, the armature will rotate in the same direction on either alternating current or direct current. When this type of motor is operated on alternating current the direction of the current in the armature circuit and the direction of the series-field flux reverse simultaneously; therefore, the average torque output always has the same direction.

Speed Adjustment and Control of a Universal Motor by a Centrifugally Operated Governor.

Speed Range:
Approximately 12:1 (or 500 to 6,000 rpm for one particular and typical governor design. Other governor assemblies can be used to provide other speed ranges)
Speed Variation from No Load to Full Load:
Approximately ±2 per cent
Horsepower:
$\frac{1}{50}$ to $\frac{1}{10}$

By means of a centrifugal-governor assembly mounted on the shaft, the universal motor can be adapted to provide a constant speed–torque characteristic from no-load to approximately 150 per cent of full-load torque. A typical universal drive motor with a centrifugal governor for speed adjustment and control is shown by Fig. 16-42. An adjusting screw on the governor assembly is provided for varying the constant output-speed value and is accessible only when the motor shaft is at standstill. Some governor assemblies are arranged so that the motor speed can be adjusted while the motor is running.

Figure 16-43 shows a diagram for a typical universal-motor drive. When the handle of the manual switch is moved to the ON position, the line contacts *M* will close and energize the motor. When the motor armature accelerates to a preset speed, the governor contacts will open and insert a resistor in series with the motor armature and field. The insertion of the resistor will prevent a further increase in speed. When the motor speed drops to a certain value, the governor contacts will reclose, and the motor will be again accelerated to the preset speed. This same type of control could be obtained by opening the circuit completely and omitting the resistor. However, more satisfactory operation can be obtained by inserting the resistor into the circuit to reduce the motor torque. The capacitor is connected in parallel with the governor contacts to improve the contact life.

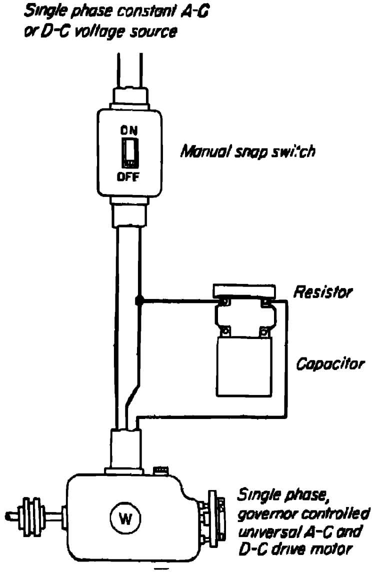

Single phase constant A-C or D-C voltage source
ON
OFF
Manual snap switch
Resistor
Capacitor
W
Single phase, governor controlled universal A-C and D-C drive motor

Fig. 16-42. Typical universal-motor drive with a speed-control governor.

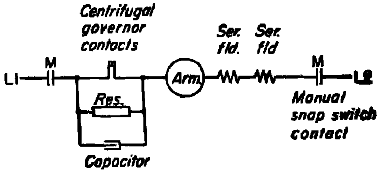


Fig. 16-43. Circuit diagram for a universal-motor drive with a centrifugal governor for speed adjustment and control.

When a universal motor with this type of control is in operation, the governor contacts open and close very rapidly, at approximately 100 to 200 times per second depending upon the natural resonant frequency of the moving contact. The contact vibration is thought to be maintained by the explosive force of the arc. The centrifugal force on the contact assembly tends to hold the contacts in the closed or open position depending upon the speed.

Figure 16-44 shows a comparison between the speed–torque curves for an universal motor with and without a governor for adjusting and maintaining the output speed constant. For a governor-controlled motor the extremely high speed characteristic is eliminated, and the speed is maintained at a constant value regardless of a change in the voltage, frequency, or load. This type of drive has been very successfully and economically applied to electric typewriters, calculating machines, motion-picture cameras and projectors, etc.

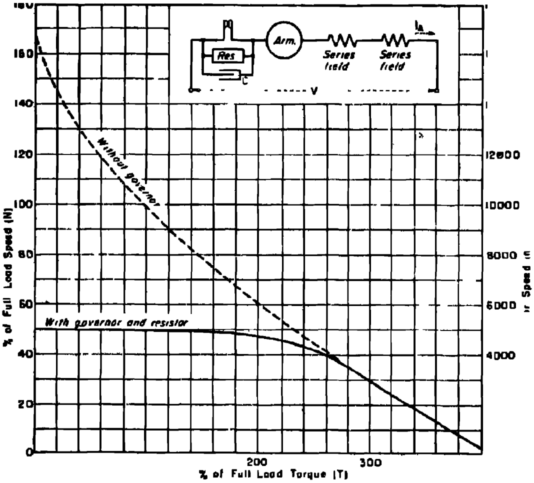


FIG. 16-44. Comparison of typical speed–torque curves for a universal motor with and without a speed governor.

D-C MOTOR DRIVES—A-C POWER SOURCE AND CONVERSION EQUIPMENT

Despite the fact that the electrical energy distributed in the United States today is predominantly 60-cycle alternating current, more d-c drives are being applied than ever before. For all the simplicity and inherent advantages of the a-c system, no a-c drives have yet been developed which are readily adjustable over the wide speed range which is now attainable with certain d-c drives. The popularity of the adjustable-speed d-c drives is steadily increasing because the versatility of the basic adjustable-voltage system has been increased by important new modifications and arrangements.

Some recently developed drives are very simple but have some limitations; other systems, more complex, provide possible speed ranges as great as 120:1. The d-c drive motor is unequaled for speed range and flexibility of control. The basic speed equations and motor characteristics have been discussed in previous sections and should be referred to for a more complete description of the principle or combination of principles utilized to obtain economical and flexible adjustable-speed drives.

Adjustable-speed drives are extensively used for paper mills, rubber-mill machinery, winders, machine-tool drives, hoists, and other similar applications.

Some adjustable-speed drives provide regenerative braking. This is an important feature which should always be considered when the drive is large. No emphasis has been placed on this feature in the following sections as speed control is the main subject.

While the cost of an adjustable-speed drive operating from an a-c power source is relatively high, it has been very widely used. It is now used for motors as small as 1 hp and even for fractional-horsepower motors in some cases. The additional first cost of this type of drive can often be saved in the design of the driven machine by the elimination of gears, cone pulleys, belts, and other complicated mechanisms.

D-c Series Adjustable-voltage Drives.

Speed Range:
10:1 (90 per cent speed reduction by armature-voltage control)
Speed Decrease from No Load to Full Load:
Approximately 15 per cent at low speed
Approximately 25 per cent at base or top speed
Horsepower:
Up to 15

An adjustable-speed drive utilizing a series generator and motor is one of the simplest means so far developed for obtaining adjustable-speed operation over a wide speed range from an a-c power source.

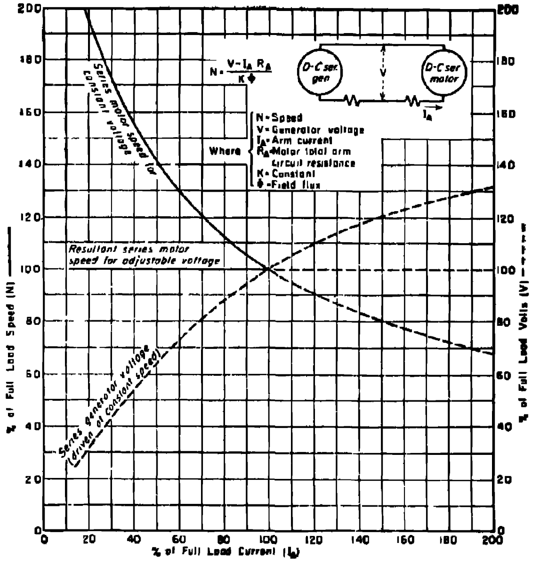


Fig. 16-45. Idealistic speed characteristic of a series motor operated in series with a series generator.

The basic speed equation for a d-c series motor shows that the predominant factors in the resultant motor speed are the value of the voltage applied to the armature and the magnitude of the field. Figure 16-45 illustrates the idealistic speed–current characteristic of a series motor operated from a series generator. The low voltage of the series generator at small values of current limits the inherently high no-load speed of a series motor; and the relatively high voltage of the series generator at high currents increases the normally low speed of a series motor which is heavily loaded.

The apparatus required for a typical series drive of this type is shown in Fig. 16-46. The d-c series generator is usually driven at constant speed by an a-c squirrel-cage induction motor. This drive has been very successfully applied to applications requiring a wide speed range with reasonably good regulation and where the motor operates in one direction or is infrequently reversed.

Conveyors, kilns, machine tools, stokers, feed-water pumps, agers, dye jigs, padders, splashers, and washers are typical applications where this particular drive can be used advantageously.

The simplicity of this drive is illustrated by the schematic diagram shown in Fig. 16-47. The only control equipment required consists of an a-c Linestarter, a start-stop push button, and a speed-adjusting rheostat.

To start the drive it is necessary only to operate the START push button which energizes coil *M* of the line contactor. Energization of coil *M* closes the *M* contacts and starts the a-c to d-c motor-generator set. As the series generator is accelerated up to its rated speed, the d-c-series drive motor also starts to rotate. This method of starting acts as a magnetic cushion and prevents shock on the driven load during the acceleration period.

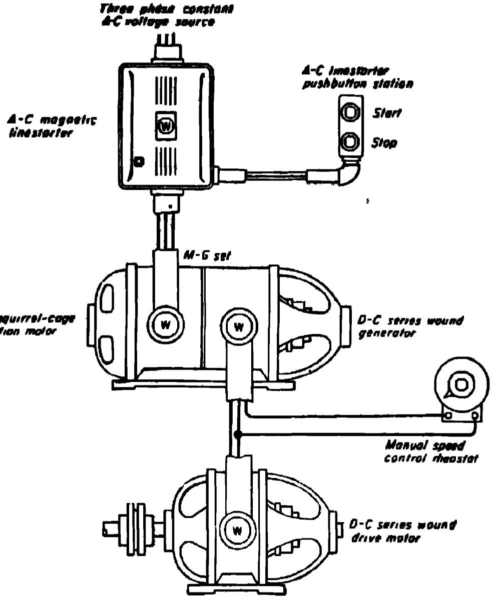

Three phase constant A-C voltage source
A-C linestarter pushbutton station
Start
Stop
A-C magnetic linestarter
M-G set
A-C squirrel-cage induction motor
D-C series wound generator
Manual speed control rheostat
D-C series wound drive motor

FIG. 16-46. Typical series adjustable-speed drive with speed variation by generator-field control.

The speed of the d-c driving motor can be varied by changing the setting of the generator-field rheostat. When all the resistance in the rheostat is effective, the generator voltage and motor speed are at their maximum value. As the rheostat resistance in parallel with the generator field is reduced, the voltage is decreased, and the motor operates at a lower speed.

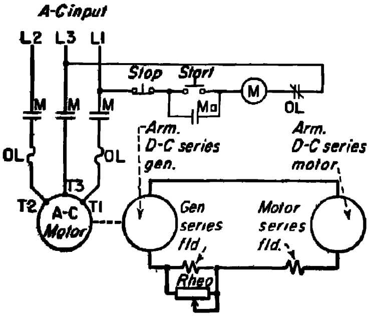


FIG. 16-47. Circuit diagram of adjustable-speed drive.

Once the rheostat has been set for the desired speed, the drive can be started and stopped without touching the rheostat. The driving motor will accelerate to the selected and desired speed after the START push button has been operated. This drive provides constant-torque drive characteristics. Regenerative braking is not possible with this type of drive.

Shunt-wound d-c motors have an inherent speed-increase characteristic of 10 per cent or more as the shunt field heats up. With the series motor used in this type of drive, there is practically no change in speed from cold to hot operation.

The generator rheostat is the chief factor limiting the size of series drive that can be manufactured economically. As the rheostat is in parallel with the generator field and must shunt up to 95 per cent of the full-load current at low operating speeds, the rheostat must consequently have a low resistance and should be mounted reasonably close to the motor-generator set.

A family of typical speed–torque curves taken from actual test data is shown in Fig. 16-48. The top curve is the speed–torque characteristic obtained when the generator is operating at full-field excitation. This set of curves was obtained by gradually shunting the generator-field current. It should be noted that the low-speed curves are practically flat except for a rather abrupt increase in speed for very light loads.

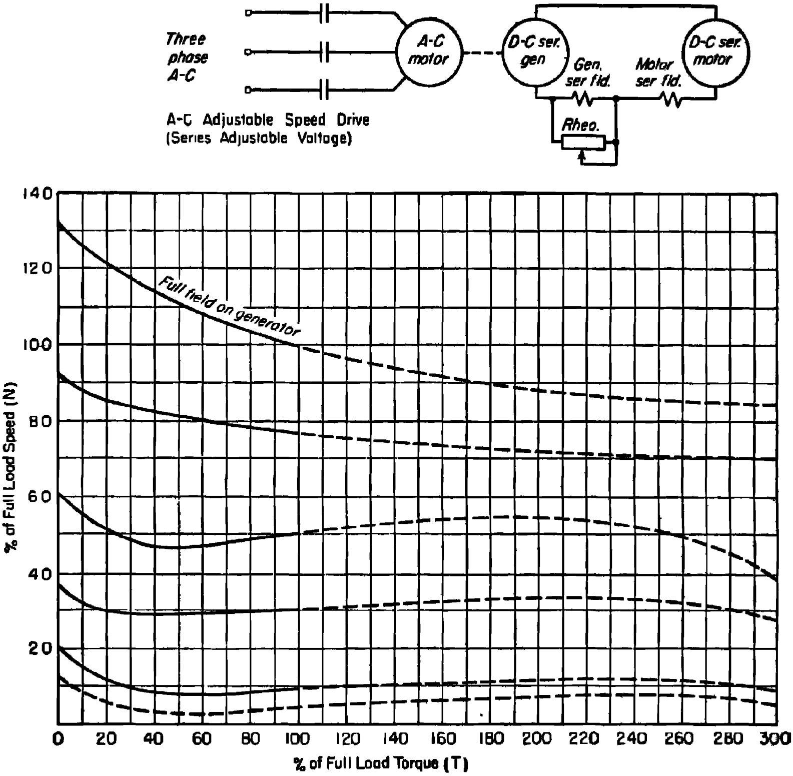


Fig. 16-48. Typical speed–torque curves for speed adjustment of a series-drive motor by series generator-field control.

The speed regulation of this drive based upon the indicated full-load and no-load speeds is rather high. However, for all practical purposes, on most applications the speed regulation is good because there is generally sufficient friction in the driven load so that the increase in speed at very light loads is never actually obtained.

When the rheostat is set for 10 per cent of the full-load base speed, the drive motor can exert a starting torque of five or six times the full-load torque value. Also, the drive can actually be operated at 5 per cent of the base speed as shown by the bottom and dotted curve for short periods for threading or inching operations. Continuous operation at the extremely low speed will cause an excessive temperature rise in the motor because of the decreased ventilation.

D-c Self-excited Shunt Adjustable-voltage Drives.

SPEED RANGE:
12:1 (67 per cent speed reduction by armature-voltage control and 4:1 speed increase by field control)
SPEED DECREASE FROM NO LOAD TO FULL LOAD:
Approximately 20 per cent at low speed
Approximately 10 per cent at base speed
Approximately 25 per cent at high speed
HORSEPOWER:
Up to 15

The self-excited shunt adjustable-voltage drive is another simplified form of the adjustable-voltage group of drives. This type of drive requires a generator that will provide stable operation at reduced voltage while operating self-excited. A stable generator is one that will repeat and hold its output voltage within a reasonable degree of accuracy. An ordinary self-excited generator controlled with a standard rheostat is stable down to approximately 60 per cent of rated voltage at no-load and allows operation over a voltage range slightly less than 2:1.

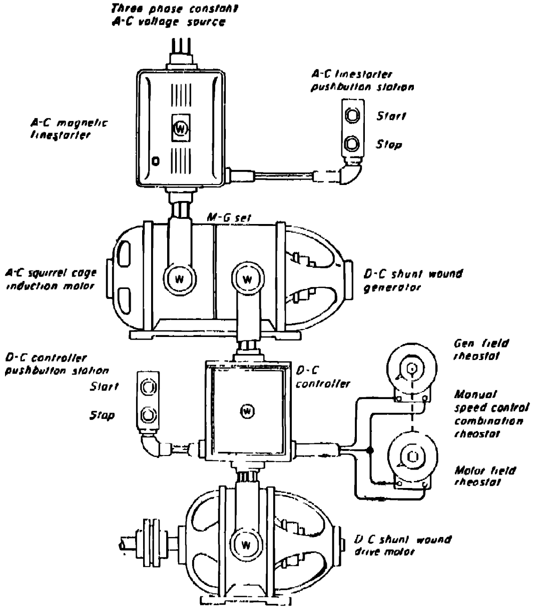

Three phase constant A-C voltage source
A-C linestarter pushbutton station
Start
Stop
A-C magnetic linestarter
M-G set
A-C squirrel cage induction motor
D-C shunt wound generator
Gen field rheostat
D-C controller pushbutton station
Start
Stop
D-C controller
Manual speed control combination rheostat
Motor field rheostat
D C shunt wound drive motor

FIG. 16-49. Typical d-c self-excited shunt adjustable-speed drive with speed variation by armature-voltage and field control.

The generator for a self-excited drive is generally designed to have field saturation at a relatively lower voltage than a standard generator. With this special characteristic, stable operation can be obtained over a voltage range of at least 3:1.

The d-c shunt drive motor differs from a standard motor in that the shunt field must provide full excitation with only one-third the rated output voltage of the generator on the motor field. This is necessary if the speed range of the system is to be 3:1.

Figure 16-49 shows the apparatus required for a typical self-excited shunt adjustable-voltage drive.

The schematic diagram for a typical self-excited shunt drive is shown in Fig. 16-50.

A self-excited shunt generator does not build up its voltage very quickly; therefore, the motor-generator set is usually started without any load. Also, it should be allowed to run continuously, and the d-c drive motor should be started and stopped by the contactor *L* in the d-c circuit. A field relay *FA* ensures that, when the line contactor *L* is open, the motor has full field, and the generator has minimum field, which provides about one-third the rated voltage. When the line contactor *L* closes, the field relay operates to increase the generator excitation and to weaken the motor field to accelerate the motor to the speed determined by the rheostat setting. The series coil *FA* on the field-accelerating relay also operates to increase the motor field and to decrease the generator output voltage automatically if the armature current exceeds a preset value. This field-acceleration relay is sometimes known as a fluttering relay as its contacts close or open continuously during the acceleration period.

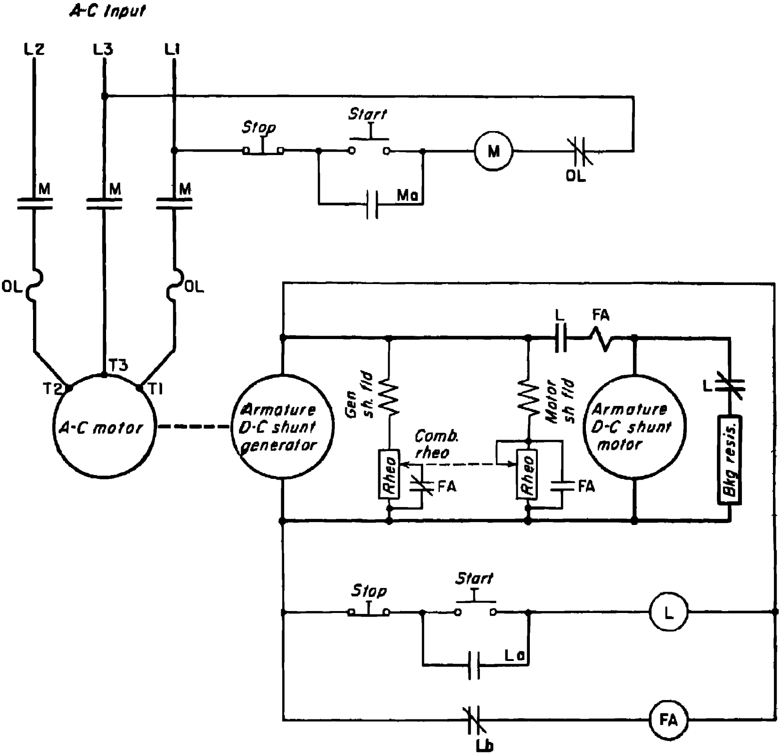


Fig. 16-50. Schematic diagram for a self-excited shunt adjustable-speed drive.

Figure 16-50 also shows how dynamic braking can be provided to stop the drive motor in a minimum of time. The braking resistor is connected across the drive-motor armature by a normally closed contact on the *L* contactor. As the motor field is continuously energized, the motor will operate as a generator when it is paralleled with a resistor. The kinetic energy of the armature is dissipated in the form of heat in the braking resistor.

This type of drive provides a speed range of approximately 12:1. Typical speed-torque curves are shown in Fig. 16-51.

The speed control of this drive is obtained with a combination rheostat which has a resistance section in each field circuit. The sliders on the two rheostats are mechanically connected so that, as one circuit is increased in resistance, the resistance in the other field circuit is decreased. In the rheostat low-speed position, the motor is operated at full field, and the generator is operated at minimum field for a low output voltage. As resistance is cut out of the generator field to raise the voltage, the resistance in series with the motor field is increased at such a rate as to maintain the full-field motor current value. After the rheostat handle has been rotated approximately 180 deg to the full-field generator position, further rotation of the rheostat handle only weakens the motor field to obtain additional speed adjustment by field control. Rated voltage output is maintained from the generator while the motor-field excitation is reduced. This particular drive is more flexible than the series type, as dynamic braking, reversing, inching, and slowdown operations can be obtained relatively easily.

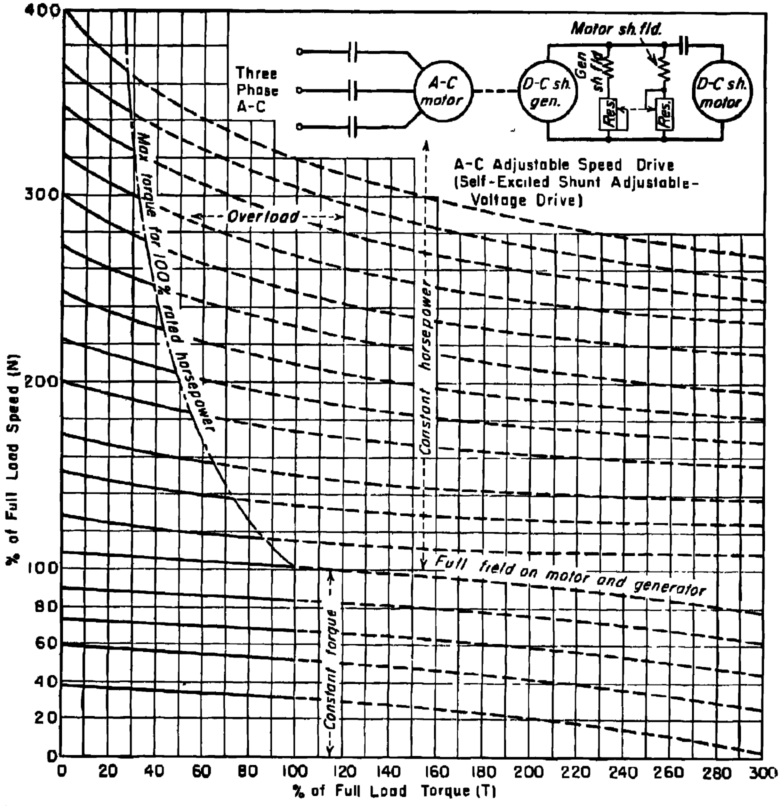


Fig. 16-51. Typical speed-torque curves for speed adjustment of a self-excited shunt drive by armature-voltage and field control.

D-c Separately Excited Shunt Adjustable-voltage Drive.

Speed Range:

40:1 (90 per cent speed reduction by armature-voltage control and 4:1 speed increase by field control)

Speed Decrease from No Load to Full Load:

Approximately 50 per cent at low speed

Approximately 10 per cent at base speed
Approximately 25 per cent at high speed
Horsepower:
Unlimited

The conventional adjustable-voltage drive, which is also known as the Ward-Leonard system, is more expensive than the series drive previously described. However, the increased cost is justified for many applications as the speed range is four times greater than that for the other drive.

This conventional adjustable-voltage drive was originally developed and used for large machines of several hundred to several thousand horsepower where it was impractical to close or open the main armature circuits. Also, for some large steel-mill applications, this method of motor starting and speed control must be used as contactors are not available with the current rating sufficient to handle the motor-armature current.

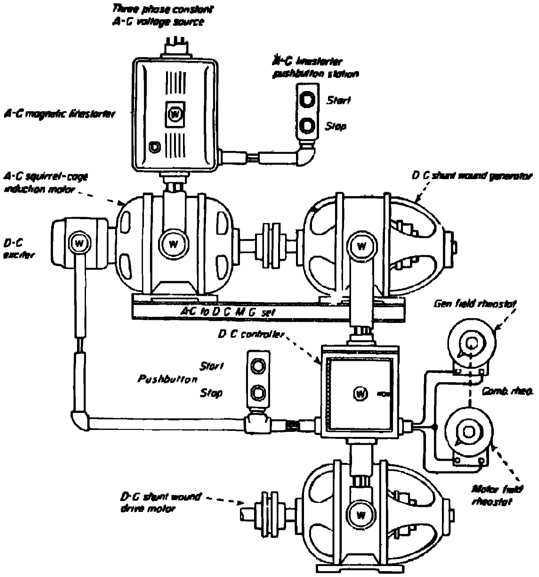


Fig. 16-52. Conventional d-c adjustable-voltage drive with speed variation by armature-voltage and field control.

The apparatus required for a typical adjustable-voltage drive is illustrated in Fig. 16-52. It should be noted that the equipment required is similar to that for a self-excited shunt drive with the exception of the addition of an exciter. Exciters are designed to provide excitation voltages which are considerably lower than those which can be obtained from self-excited generators.

The circuit diagram for a moderately rated adjustable-voltage drive is shown in Fig. 16-53. The motor generator set a-c drive motor is started by operating the START push button to close the *M* contactor. For some applications it is convenient to control the starting and stopping of the d-c drive motor by means of the *L* contactor in the d-c circuit. A field-accelerating relay is generally used to control the motor acceleration when the speed range obtained by field weakening is greater than 2:1.

For applications where the drive motor is relatively large, the line contactor *L* is omitted, and the generator- and motor-field rheostats are electrically connected to the controller in such a manner that the generator must be operating at a low voltage before the motor can be started. The speed of the motor can then be readily adjusted, first with the generator-field rheostat and then with the motor-field rheostat. These rheostats are often motor operated for remote or automatic operation.

When it is necessary to reverse the direction of the d-c motor, the control can be readily arranged to reverse the generator field. The contactors required for reversing the generator-field connections are usually available control devices and relatively very small compared to the switching equipment that would be required to reverse the motor-armature circuit. Reversal of the generator field will reverse the polarity of the generated voltage. This in turn will cause reversal of the drive motor as the direction of the motor field is not changed.

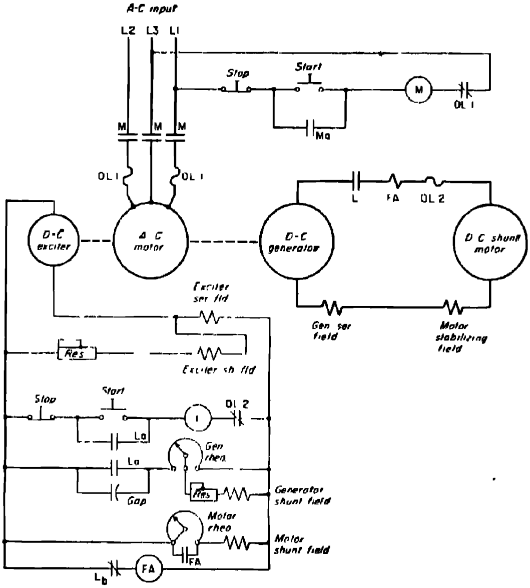


FIG. 16-53. Diagram for a typical conventional adjustable-voltage drive.

Figure 16-54 shows typical speed–torque curves for the separately excited adjustable-voltage drive. This drive provides constant torque below the base speed and constant horsepower above the base speed.

For adjustable-voltage drives the power losses, which make speed adjustment by armature-circuit resistance in constant-voltage drives inefficient, are eliminated. Also, the absence of additional resistance in the armature circuit improves the regulation.

Field rheostats can be readily built with a large number of steps and will make it possible to start and stop a motor smoothly. This is absolutely essential for high-speed elevators and papermaking machines.

The adjustable-voltage system also provides a means for controlling a group of motors simultaneously. Steel-mill roll-out tables are sometimes controlled in this manner.

D-c Separately Excited Shunt Adjustable-voltage Drives with Rototrol Regulators.

SPEED RANGE:

120:1 (approximately 98 per cent speed reduction by armature-voltage control and 2:1 speed increase by field control)

Speed Variation from No Load to Full Load:
±3 per cent
Horsepower:
Unlimited

The addition of a Rototrol regulator to a conventional adjustable-voltage drive provides a drive with a greater speed range than any of the other types of drives now commercially available.

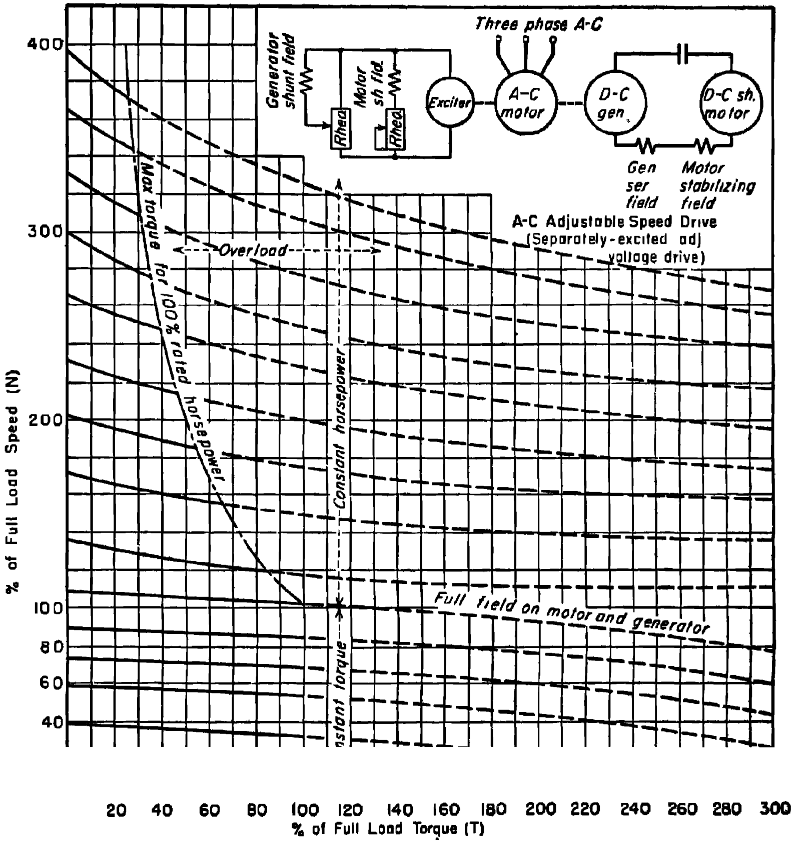


Fig. 16-54. Typical speed–torque curves for a conventional adjustable-voltage drive.

In order to have a drive that operates with a speed characteristic which does not change appreciably from no load to full load, compensation must be made for certain inherent generator and motor characteristics that normally cause poor regulation.

The residual voltage of a commercially built generator and the motor-armature *IR* drop are the chief factors that prevent low speeds and good regulation. A Rototrol, which is a small d-c generator with special field windings, can be added to the conventional drive to compensate for these factors.

A typical adjustable-voltage drive with a Rototrol regulator unit is shown by Fig. 16-55. The unit assembly of the motor-generator set a-c drive motor, the d-c shunt-wound generator, the d-c exciter, and the Rototrol unit makes a very compact unit which facilitates the installation.

The residual voltage of a typical d-c generator is between 3 and 4 per cent of the normal operating voltage. For this reason, a d-c motor that is energized from such a generator is limited to a speed range of about 25:1 for an adjustable-voltage control system. The residual field of the generator prevents the output voltage from being any lower even though the generator-field windings are not even excited. As the magnitude of the residual field is a function of the previous magnetic history of the generator, it is also necessary that the control device or means used to compensate for and overcome the residual voltage be able to measure the actual residual voltage accurately.

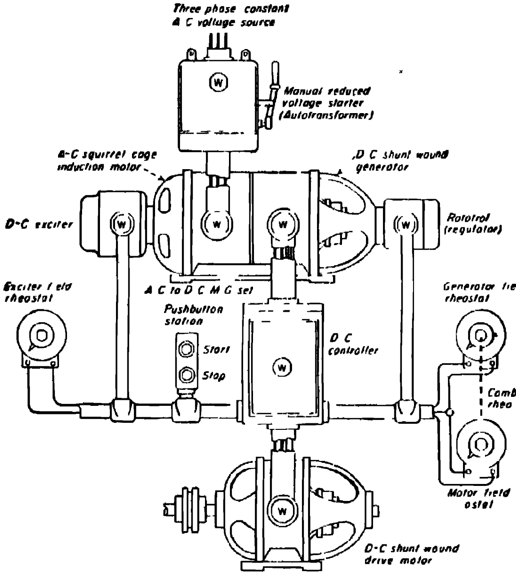

Three phase constant A C voltage source
Manual reduced voltage starter (Autotransformer)
A-C squirrel cage induction motor
D C shunt wound generator
D-C exciter
Rototrol (regulator)
Exciter field rheostat
A C to D C M G set
Pushbutton station
Start
Stop
D C controller
Generator field rheostat
Comb rheo
Motor field ostat
D-C shunt wound drive motor

Fig. 16-55. D-c adjustable-voltage drive with compensation for the generator residual voltage and the motor-armature *IR* drop.

Because of the motor-armature *IR* drop, the speed of a d-c motor decreases as the load current increases. In order to obtain a flat speed-torque characteristic, a speed-regulating means must always automatically increase or decrease the generator output voltage which is applied to the motor by an amount exactly equal to the change in the motor *IR* drop for all conditions of load. A Rototrol can compensate for the motor-armature *IR* drop and has been very effectively used for this purpose.

The basic circuit of an adjustable-voltage drive with a Rototrol for constant speed regulation is illustrated in Fig. 16-56. A manually operated autotransformer type of starter for the motor-generator set a-c drive motor has been shown as this type of starter is frequently used to start a-c motors where the line disturbances must be kept to a minimum. When lower than rated voltage is applied to an a-c motor, the starting current and resultant torque are decreased as previously described for a-c induction motors.

In general, Rototrol generators are designed for operation on the straight or air-gap portion of their saturation curves. The voltage that they generate will be directly proportional to the net field excitation. The Rototrol unit is usually directly connected to the motor-generator set and is driven at a constant speed.

The main generator-field winding is divided in two equal sections and is connected in a special bridge circuit. The resistances of the main generator- and Rototrol-field sections are balanced with resistor units in the other two sides of the bridge so that, with zero voltage from the Rototrol armature, there is no difference in potential between the points to which the Rototrol armature is connected. The main generator-field sections are accumulative, and the desired speed of the motor is controlled by the generator-field rheostat in the usual manner. The resistance in series with the Rototrol armature is used to tune the circuit so that the Rototrol operates on the air-gap portion of the saturation curve.

At no-load on the motor, resistor $R6$ is adjusted so that the fields produced by $RF1$ and $RF2$ are completely neutralized by the shunt pilot field $RF3$. (Magnetic fields $RF1 + RF2 - RF3 = 0$.) Under these conditions the generated output voltage of the Rototrol armature is zero as the net magnetic field is zero.

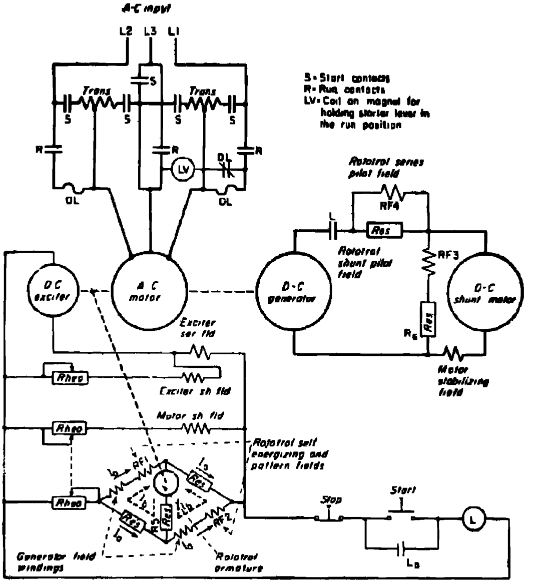


FIG. 16-56. Diagram for an adjustable-voltage drive with automatic speed regulation by means of a Rototrol.

The Rototrol shunt $RF3$ and series $RF4$ pilot fields are proportioned so that, for any load conditions, the net ampere-turns are proportional to the counter emf of the d-c drive motor. The speed of a d-c motor with constant field excitation is proportional to its counter emf which, in turn, is equal to the applied voltage minus the motor-armature IR drop.

As an increase in the motor load results in an increase in the motor-armature current, the ampere-turns of the Rototrol series pilot field $RF4$ also increase. The ampere-turns of the series field $RF4$ add to those of the self-energizing fields $RF1$ and $RF2$. Furthermore, a decrease in the main generator output voltage because of the regulation also decreases the ampere-turns produced in the Rototrol shunt pilot field $RF3$. As a result of these changes in the net field excitation, the Rototrol now generates a voltage which circulates the current I_b through the Rototrol self-energizing and pattern fields and the main generator fields to supplement the excitation normally produced by the current I_a from the exciter bus. This additional main generator-field excitation increases the generator output voltage to a value that is sufficient to restore the d-c drive-motor speed to the desired and preset value.

The Rototrol series field *RF*4 also compensates the circuit for the resistance drop in the lines and motor armature. The shunt pilot field *RF*3 compensates the circuit for the resistance drop as well as the magnetizing effects that occur in the main generator armature. All these conditions are integrated by the Rototrol, and the resultant compensation produces a flat speed–torque characteristic.

It should be noted that the d-c exciter supplies the normal generator-field excitation and that the Rototrol supplies only the regulation excitation necessary to maintain the motor speed at a substantially constant value.

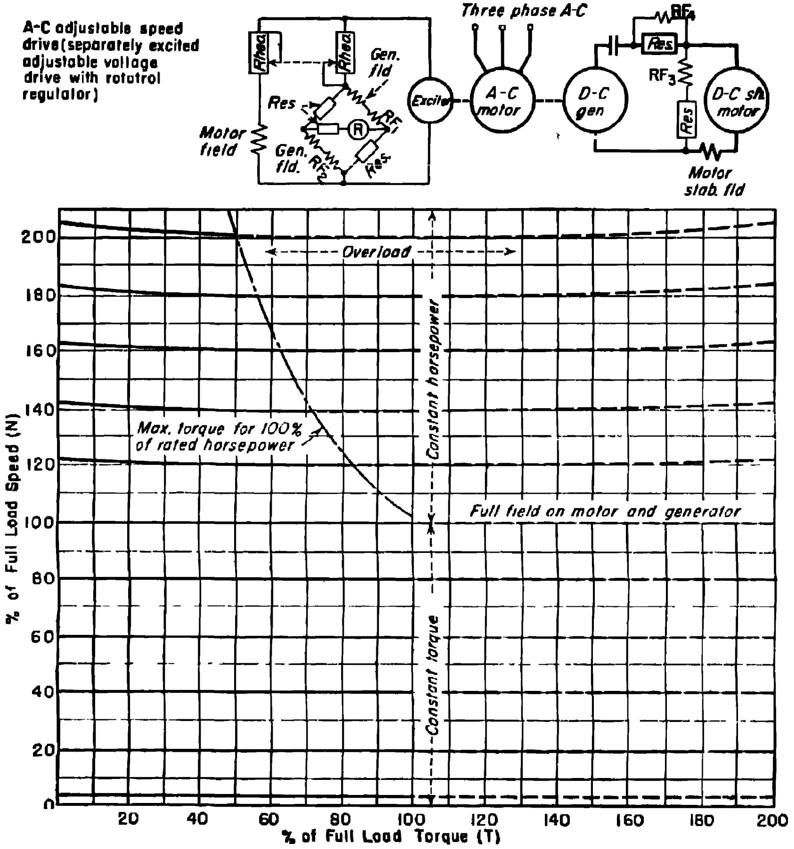


FIG. 16-57. Typical speed–torque curves of an adjustable-voltage drive with a Rototrol unit for 120:1 speed range.

Figure 16-57 shows typical speed–torque curves taken from actual test data and illustrates the speed control that can be obtained from this type of drive. The motor has no tendency to stall even at speeds of one-sixtieth of the base speed. At light loads for this very low speed, the voltage required by the drive motor is less than the residual voltage of the main generator. The Rototrol reverses the current in the field of the generator to buck the residual voltage to a value that gives the required motor speed. As the motor load increases, the Rototrol voltage polarity is restored to the original condition in order to build up the generator voltage to compensate for the motor *IR* drop.

The use of a Rototrol unit in an otherwise conventional adjustable-voltage drive gives a speed range far greater than the supposed limits of a few years ago without greatly complicating the electrical apparatus or circuit. These wide speed ranges were formerly obtainable only with laboratory apparatus.

The particular drive illustrated has a combination main generator- and motor-field rheostat so that full field is continually maintained on the motor when it is operating below the base speed. This drive provides constant torque below the base speed and constant horsepower above the base speed.

Other basic circuits have been developed so that a Rototrol unit can be applied to a drive to provide more accurate regulation and control for a specific speed, to maintain constant-torque output, to maintain constant-horsepower output, to give smooth acceleration and deceleration, or for any other drive characteristic that can be measured in units of voltage or current.

This type of drive is ideal for automatic welding heads, elevators, metal planers, core-type paper reels, paper machines, skip hoists, electric shovels, machine tools, etc.

D-c Electronic (Mot-o-trol) Adjustable-voltage Drives.

Speed Range:
40:1 (approximately 95 per cent speed reduction by armature-voltage control and 2:1 speed increase by field control)
Speed Decrease from No Load to Full Load:
Approximately 10 per cent at low speed
Approximately 1½ per cent at base speed
Approximately 8 per cent at high speed
Horsepower:
Up to 15

An electronic power supply can be used in place of the motor-generator set in a conventional adjustable-voltage drive. This is especially advantageous, particularly in the smaller drive ratings, where the customary additional rotating and relatively large control equipment may not be suitable for a particular installation.

The Mot-o-trol electronic drive provides a wide and stable speed range, good speed regulation, and a very smooth and automatic acceleration of the d-c drive motor.

The heart of the Mot-o-trol is the electronic power supply for the armature and field of the d-c motor. Grid-controlled thyratron tubes are used in the rectifiers which convert either a single-phase or polyphase a-c power source, depending upon the rating, to a d-c voltage supply.

The control characteristic of a typical thyratron tube is shown in Fig. 16-58. The positive anode voltage at which a thyratron tube will conduct current can be controlled by a very small potential on the grid with respect to the cathode.

As shown in the right-hand figure, a tube can be prevented from firing when a positive half cycle of voltage with a rms value of 440 (or a peak of 620) volts is impressed between the anode and cathode if the control grid is energized with a negative potential of more than 4 volts.

A thyratron type of tube will conduct current only when the anode is at a positive potential with respect to the cathode, and once the mercury vapor in the tube ionizes, the conduction can be stopped only when the anode-to-cathode voltage decreases to a very low value.

In the Mot-o-trol electronic drive, an adjustable voltage for the d-c motor armature is obtained by advancing or delaying the point on the a-c anode-voltage wave at which the thyratron tube conducts current.

Figure 16-59 shows the control means and principle which are used to vary the rectifier output voltage from zero to the rated value. A small transformer with a center tap, a resistor, and a capacitor form a phase-shifting circuit. This circuit is adjusted for many applications so that the voltage E_g lags the anode voltage E_A by approximately 90 deg. By connecting a d-c voltage E_b source which is adjustable from a suitable negative to a positive value in series with the a-c grid voltage E_g, the a-c grid voltage can be shifted vertically up and down with respect to the zero axis of the anode voltage E_A. As shown by the three voltage-relationship diagrams, the tube-firing or conduction time can be accurately controlled for a conduction period of zero to a maximum time of ½ cycle merely by varying the d-c grid-voltage component from a negative to a positive value. This type of control is used for controlling both the armature- and shunt-field voltages.

The apparatus required for a typical nonreversing electronic drive is shown by Fig. 16-60. A complete Mot-o-trol electronic drive is usually comprised of an a-c power transformer, an electronic controller, an operator's control station, and a d-c shunt motor.

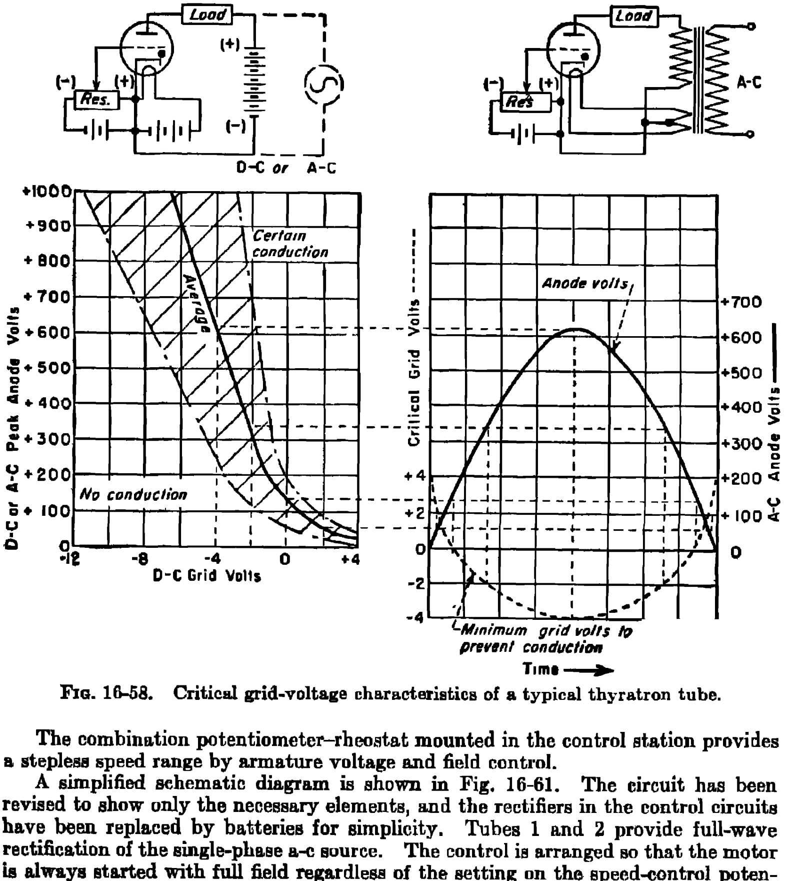


Fig. 16-58. Critical grid-voltage characteristics of a typical thyratron tube.

The combination potentiometer–rheostat mounted in the control station provides a stepless speed range by armature voltage and field control.

A simplified schematic diagram is shown in Fig. 16-61. The circuit has been revised to show only the necessary elements, and the rectifiers in the control circuits have been replaced by batteries for simplicity. Tubes 1 and 2 provide full-wave rectification of the single-phase a-c source. The control is arranged so that the motor is always started with full field regardless of the setting on the speed-control potentiometers *P*2 and *P*2*A*. The field-weakening control does not become effective until the motor accelerates to its base speed.

Transformer 3 and tubes 5 and 9 provide armature-current limit control. Fast, smooth, and shockless motor acceleration is automatically provided by these devices which control the d-c voltage component of the grid voltage which controls the 1 and 2 thyratron tubes. In this manner, the voltage output of the armature-voltage rectifier tubes is always precisely controlled so that a preset limit on the current is never exceeded.

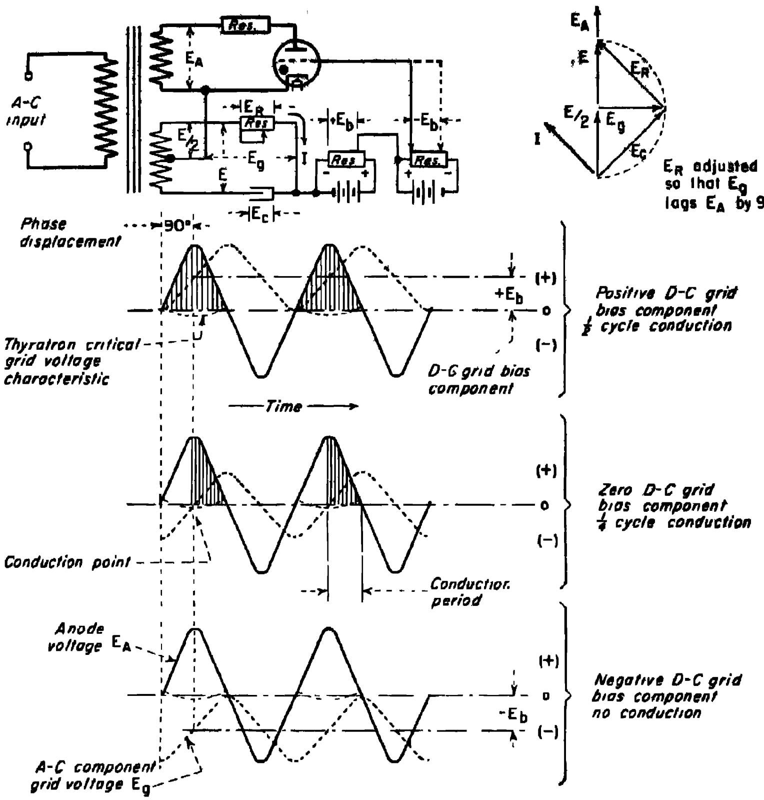

Res.
E_A
A-C input
$E/2$
E_R
Res
E_g
I
E
$+E_b$
$-E_b$
Res
Res.
E_c
E_A
E
E_R
$E/2$
E_g
I
E_c
E_R adjusted so that E_g lags E_A by 90°
Phase displacement
90°
(+)
$+E_b$
0
(−)
Positive D-C grid bias component ½ cycle conduction
Thyratron critical grid voltage characteristic
D-C grid bias component
Time
(+)
0
(−)
Zero D-C grid bias component ¼ cycle conduction
Conduction point
Conduction period
Anode voltage E_A
(+)
0
$-E_b$
(−)
Negative D-C grid bias component no conduction
A-C component grid voltage E_g

FIG. 16-59. Voltage control of a thyratron tube by means of a lagging a-c voltage wave and an adjustable d-c voltage.

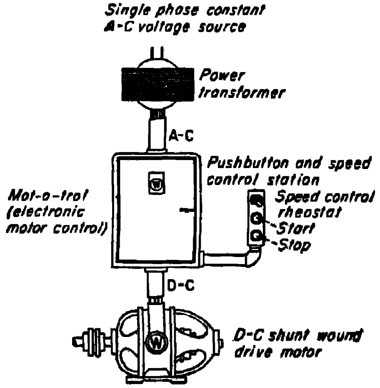
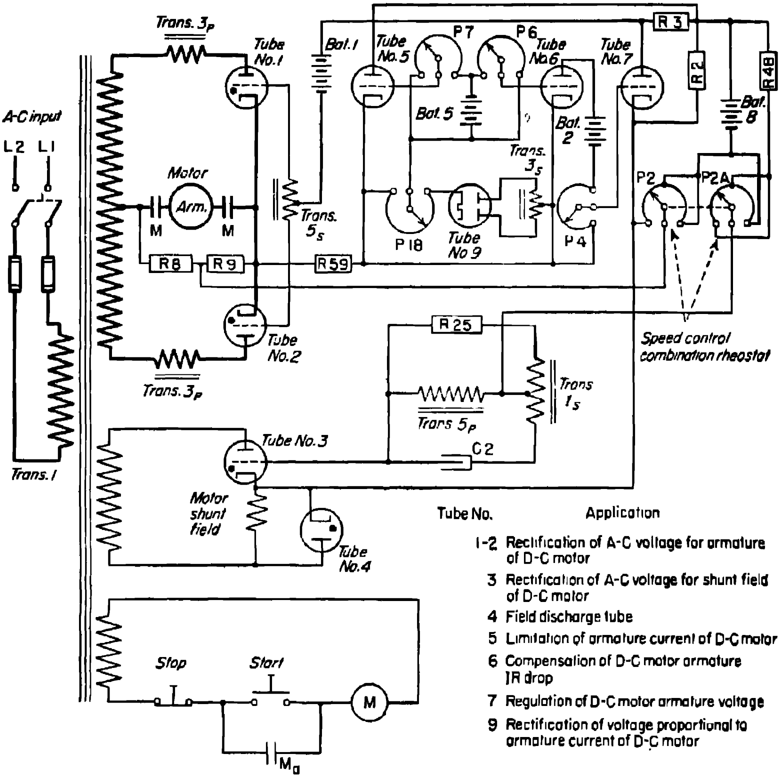

Single phase constant A-C voltage source
Power transformer
A-C
Pushbutton and speed control station
Mot-o-trol (electronic motor control)
Speed control rheostat
Start
Stop
D-C
D-C shunt wound drive motor

FIG. 16-60. Typical Mot-o-trol electronic drive with speed control by armature-voltage and field control.

Tube 6 also automatically controls the grid voltage of tubes 1 and 2 so that the motor *IR* drop is compensated for and does not cause the motor speed to decrease with an increase in the load.

Tube 7 corrects the inherently poor voltage regulation of tubes 1 and 2. The coordinated operation of all the control elements provides an output voltage from the rectifier that will give a flat speed-torque characteristic.

The shunt-field excitation current is rectified by tube 3 which is manually controlled by the potentiometer *P2A* and is automatically controlled by the voltage-regulation-tube network. Tube 4 provides a discharge circuit for the field current during the time that tube 3 is not conducting.


Trans. 3p
Tube No. 1
Bat. 1
Tube No. 5
P7
P6
Tube No. 6
Tube No. 7
R3
R2
R48
Bat. 8
A-C input
L2 L1
Bat. 5
Bat. 2
Trans. 3s
Motor
Arm.
M
M
Trans. 5s
P18
Tube No 9
P4
P2
P2A
R8
R9
R59
Tube No. 2
R25
Speed control combination rheostat
Trans 1s
Trans. 3p
Trans 5p
C2
Tube No. 3
Motor shunt field
Tube No. 4
Trans. 1
Stop
Start
M
Ma
Tube No. Application
1-2 Rectification of A-C voltage for armature of D-C motor
3 Rectification of A-C voltage for shunt field of D-C motor
4 Field discharge tube
5 Limitation of armature current of D-C motor
6 Compensation of D-C motor armature IR drop
7 Regulation of D-C motor armature voltage
9 Rectification of voltage proportional to armature current of D-C motor

(Batteries shown only to simplify the circuits)

FIG. 16-61. Simplified schematic diagram of a typical Mot-o-trol electronic drive.

Typical speed-torque curves are shown in Fig. 16-62. Much development has been and is being done on improving the operation and increasing the horsepower range of electronic motor controls.

The d-c motors used in this type of drive are usually furnished with a somewhat larger than standard frame than that required for motors operating at constant speed and from nonpulsating d-c power sources. This is necessary as the pulsating d-c current from the rectifier circuit causes more heating in the motor than results from operation on steady direct current. Also, at low speeds, the ventilation of the motor is materially reduced. An electronically controlled drive of this type generally provides the most satisfactory operation when one manufacturer supplies the complete equipment.

This type of drive can be readily supplied suitable for inching and reversing service and dynamic braking. It is not suitable for regenerative braking. The Mot-o-trol is particularly well suited for machine-tools, conveyors, key seaters, gear-cutting machines, and any other applications where a well-regulated and wide speed range is desired. With the exception of the actual drive motor, the electrical apparatus is also vibrationless and provides a more compact unit than an equally rated motor-generator set and its control.

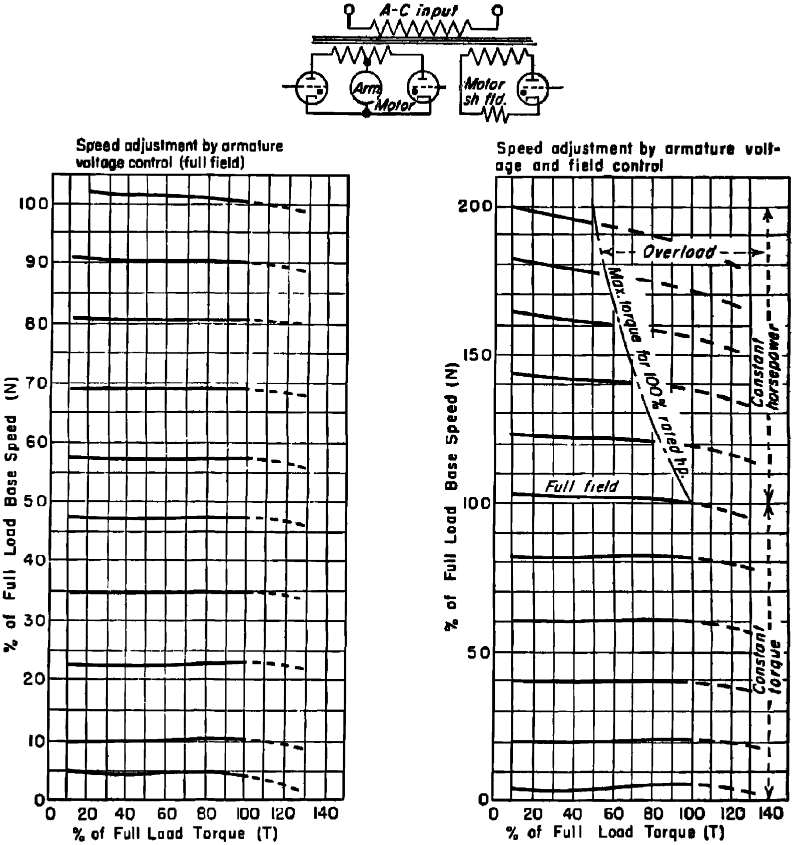


FIG. 16-62. Typical speed–torque curves for a d-c shunt motor operated from an adjustable-voltage electronic power source.

A-C MOTOR DRIVES—A-C POWER SOURCE AND CONVERSION EQUIPMENT

The speed of an a-c motor can be adjusted by changing the frequency of the applied voltage. Refer to Figs. 16-24, 16-25, and 16-26 for curves and a table of motor speeds vs. frequencies. The main disadvantage to this type of speed control is that the adjustable-frequency system generally requires considerable conversion equipment.

A-c Adjustable-frequency Drives—Motor-generator Sets.

SPEED RANGE:
  5 or 6 : 1 for typical drives (80 or 84 per cent speed reduction by frequency control)
SPEED DECREASE FROM NO LOAD TO FULL LOAD:
  Without d-c to a-c motor-generator set drive-motor compensation
    Approximately 80 per cent at low speed
    Approximately 15 per cent at base speed
  With d-c to a-c motor-generator set drive-motor compensation
    Approximately 50 per cent at low speed
    Approximately 5 per cent at base speed
HORSEPOWER:
  Unlimited

A typical adjustable-frequency system is illustrated in Fig. 16-63. It consists of an a-c synchronous drive motor and a d-c shunt-wound generator for the a-c to d-c adjustable-voltage d-c motor-generator set, a d-c drive motor and an a-c generator for the d-c to a-c adjustable-voltage and -frequency motor-generator set, and an a-c squirrel-cage drive motor.

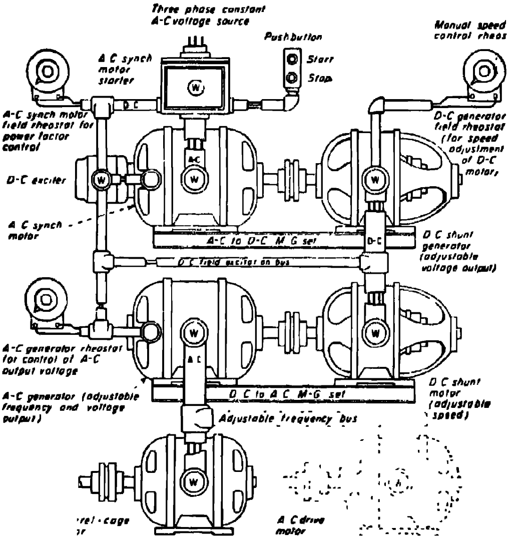


FIG. 16-63. Typical adjustable-frequency drive with speed adjustment by frequency control.

It should be very obvious that such a drive is practical and economical only when a number of drive motors are to be simultaneously controlled and operated at exactly the same speed. Such a drive is not feasible for a single drive motor as the ultimate speed range could be obtained much better from the d-c motor which drives the a-c generator. This type of drive has often been used for steel-mill roll-out-table drives.

An undesirable characteristic of this drive is the fact that the speed regulation of the actual drive motor is affected by the summation of the per cent regulation of all the machines in the system. This poor regulation characteristic can be minimized and reduced to almost that of the squirrel-cage drive motor by using a synchronous drive motor in the a-c to d-c motor-generator set and by employing a Rototrol regulator to maintain the speed of the d-c drive motor of the d-c to a-c motor-generator set at a reasonably constant value.

When a synchronous motor is used for the driving means in the a-c to d-c motor-generator set, power-factor correction can be obtained in addition to driving the set at an absolutely constant speed.

The schematic diagram of a typical adjustable-voltage drive is shown by Fig. 16-64. To put the a-c drive motor in operation, the START push button is operated to start the synchronous motor. When the START button completes a circuit to and energizes the coils M and T, the M contacts close and apply voltage to the synchronous motor. The usual type of synchronous motor has damper bars in the rotor and starts similarly to a squirrel-cage induction motor. During the starting period, the synchronous-motor d-c field winding is shunted by a resistor which is in series with a normally closed contact F. The diagram symbol T represents a time-delay relay which operates to close the T contact after a definite time period. After the synchronous motor accelerates to a speed high enough for synchronization, the relay contact T closes and energizes the coil F of the field contactor. This opens the shunting-resistor circuit and applies direct current to the field. The synchronous motor should now pull into step and run at an absolutely constant speed. The synchronous-motor field is shunted to prevent high voltage from being induced in the winding during the starting and stopping periods.

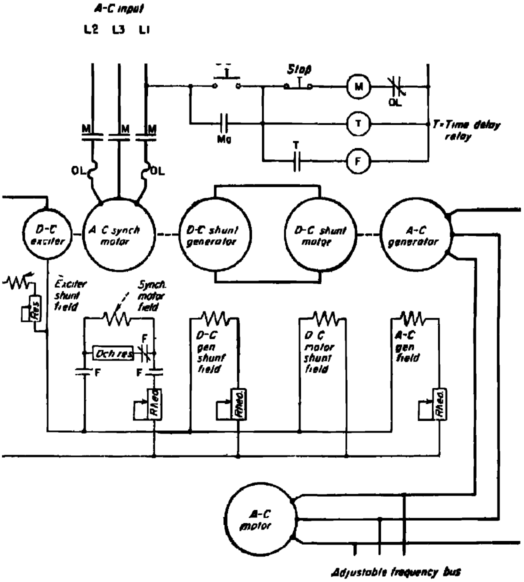


FIG. 16-64. Circuit diagram for a typical a-c adjustable-speed drive with speed variation by frequency control.

The control can be arranged so that the actual drive motors are always started at the lowest speed. This speed control and adjustment can be readily obtained by operating only the d-c generator shunt-field rheostat.

Typical speed-torque curves for a three-phase squirrel-cage drive motor are shown in Fig. 16-65. These curves were plotted from actual test data on a motor for application to a roll-out table. The motor developed approximately 0.9 hp when operated on 65 cycles for the base speed.

As the inductive reactance of an a-c motor-field winding varies directly with the frequency, the applied voltage and frequency must always be in approximately the same ratio of volts per cycle for any speed. This is essential in order to develop the necessary torque and to limit the current to a safe value. ($X_L = 2\pi fL$, where X_L = the inductive reactance, $\pi = 3.14$, f = the frequency, and L = the inductance of the winding.) The winding-resistance factor is predominant at the lower frequencies.

These curves are based upon the regulation only of the a-c squirrel-cage drive motor. Unless the a-c and d-c motor-generator set drive motors operate at constant speed for any setting, the actual regulation of the drive will be greater than that indicated. Any intermediate speeds between those indicated can be readily obtained.

With this type of control, a load can be started smoothly as the starting torque steadily increases as the applied voltage and frequency to the drive motor are increased.

When it is necessary that the drive motors operate at an absolutely constant speed, synchronous motors can be used. Also, nonexcited synchronous motors for lower ratings can be operated very satisfactorily from the adjustable-frequency set. Nonexcited synchronous motors with fractional-horsepower ratings have been applied to drive nylon-spinning buckets where an absolutely constant operating speed is desired with adjustability in a range of approximately 6:1.

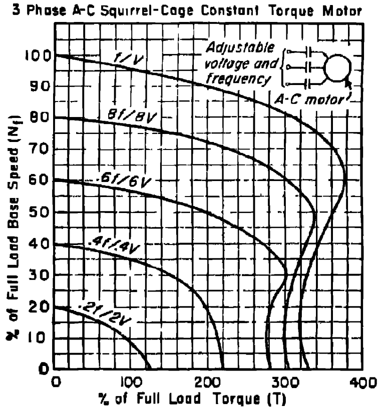


Fig. 16-65. Typical speed-torque curves for a three-phase squirrel-cage motor operated from an adjustable-frequency system.

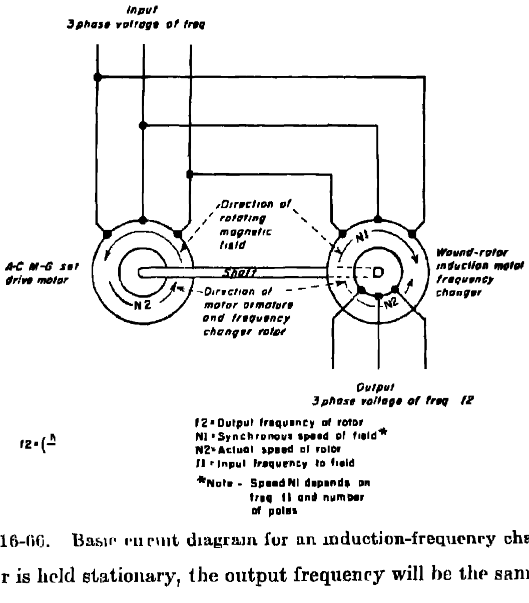
A-c Adjustable-frequency Drives—Induction-generator Sets.

Speed Range:
  2:1 or 3:1 for typical drives (2:1 or 3:1 speed increase by frequency control)
Speed Decrease from No Load to Full Load:
  With induction motor driven set and induction drive motor
    Approximately 10 per cent at base speed
    Approximately 3 per cent at high speed
  With synchronous motor driven set and induction drive motor
    Approximately 5 per cent at base speed
    Approximately 1½ per cent at high speed
  With synchronous motor driven set and synchronous drive motor
    0.0 per cent at base and high speed
Horsepower:
  Up to 60 (for induction motor driven sets)
  Up to 100 (for synchronous motor driven sets)

Induction frequency changers are used extensively for supplying power to high-frequency drive motors in the wood-working and rayon industries where relatively high speeds are required. The induction type of generator is ideal for applications where sturdiness, simplicity, reliability, low maintenance, and good regulation are important. Commercial sets are available for converting from 25-, 50- and 60-cycle frequencies up to 175 or 180 cycles. Induction generators are not used to obtain lower than supply frequencies as the lower than base speeds can be readily obtained with multispeed motors.

An induction generator is actually a wound-rotor induction motor whose rotor is driven opposite to the direction of its rotating magnetic field.

When an induction-motor primary field is energized with a polyphase voltage, a rotating magnetic field is produced. The speed of the field rotation depends upon the frequency and the number of poles. This primary-circuit revolving field induces a voltage in the rotor or secondary winding of the motor similar to the action that occurs in a two-winding transformer. The basic circuit diagram for and the relative directions of rotation for the primary winding field and the rotor for an induction frequency-changer set are shown by Fig. 16-66.


Input
3 phase voltage of freq
Direction of rotating magnetic field
A-C M-G set drive motor
Shaft
D
N1
N2
N2
Direction of motor armature and frequency changer rotor
Wound-rotor induction motor frequency changer
Output
3 phase voltage of freq f2

f2 = (N

f2 = Output frequency of rotor
N1 = Synchronous speed of field*
N2 = Actual speed of rotor
f1 = Input frequency to field

*Note - Speed N1 depends on freq f1 and number of poles

Fig. 16-66. Basic circuit diagram for an induction-frequency changer.

If the rotor is held stationary, the output frequency will be the same as the input frequency.

To secure a frequency higher than that of the power supply, the rotor must be driven by external means in a direction opposite to that of the rotating field. The relationship between the output and input frequencies is given in the equation at the bottom of Fig. 16-66.

A portion of the power output from an induction generator is supplied by direct transformer action between the primary winding and the rotor winding. This power transfer varies with the ratio of input to output frequency. For example, on 60- to 180-cycle conversion, one-third of the power output is supplied by direct transformer action. The set drive motor must be large enough to supply the other two-thirds of the output plus a portion of the induction-generator losses.

A typical three-speed drive is illustrated by Fig. 16-67. A motor-generator set can be readily constructed from standard machines. With an induction motor and two wound-rotor motors with a different number of poles, two frequencies higher than that of the line supply can be obtained. A standard a-c Linestarter with additional overload protection for the induction generators can be used for primary control.

For the secondary control, an a-c controller and a selector switch can be used to operate the drive motor at the desired speed.

The schematic diagram for the three-speed drive is illustrated by Fig. 16-68. The induction-generator set can be put in operation by depressing the START button. This will start the motor-generator drive motor and energize the primary windings of the induction generators.

The manually operated cam selector switch can be set for the desired speed. This particular control scheme provides low-voltage protection. The slow *S*, medium *M*, and fast *F* speed contactors are interlocked electrically by electrical interlocks to prevent the coil on a contactor from being energized unless the other two contactors are fully open.

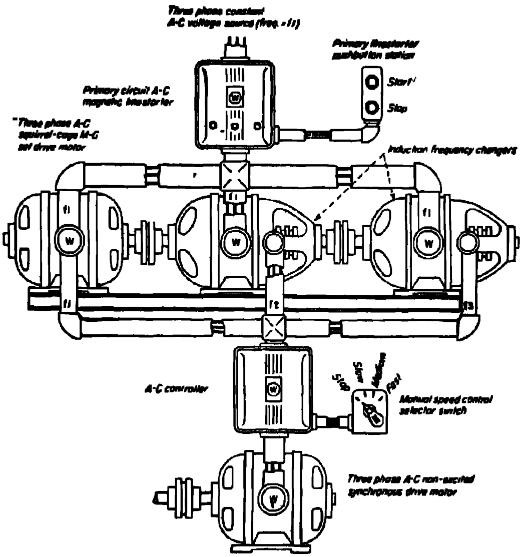


Fig. 16-67. Typical three-speed drive with speed adjustment by frequency control. Induction generator set for higher-than-line frequencies.

The squirrel-cage type of induction motor is frequently operated on the power output of an induction-generator set. However, to illustrate how an absolutely constant output speed can be obtained, the speed–torque characteristics of a nonexcited synchronous motor are shown in Fig. 16-69. These curves are based upon a power supply with 0 per cent regulation. This frequency regulation of 0 per cent can be obtained by using a synchronous motor to drive the induction generators at absolutely constant speed.

A nonexcited synchronous motor is similar to an ordinary squirrel-cage induction motor except for slots that are milled in the armature. This alteration changes the reluctance of and the distribution of the magnetic lines of flux in the armature iron. The flux lines are concentrated in certain armature regions similar to machines with salient poles. The starting torques shown for this type of motor are greater than those for a three-phase squirrel-cage motor of the same full-load rating, as the rated output of a given nonexcited synchronous motor is only about one-third that of the standard rating of the motor from which it was made.

This particular type of drive motor pulls into step at approximately 150 to 225 per cent of full-load torque and then operates at a constant speed. The pull-out torque is higher than the pull-in value and is approximately as shown in Fig. 16-69. The pull-in torques for the 100 and 200 per cent base speeds can be increased to that at 300 per cent speed by slightly increasing the applied voltage.

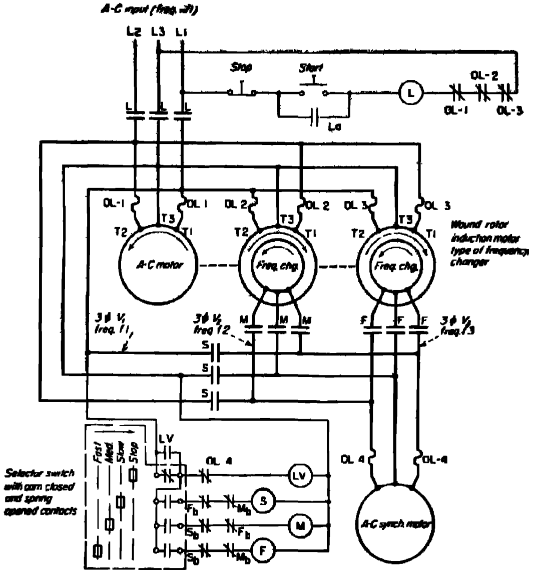


FIG. 16-68. Typical circuit diagram for a three-speed drive.

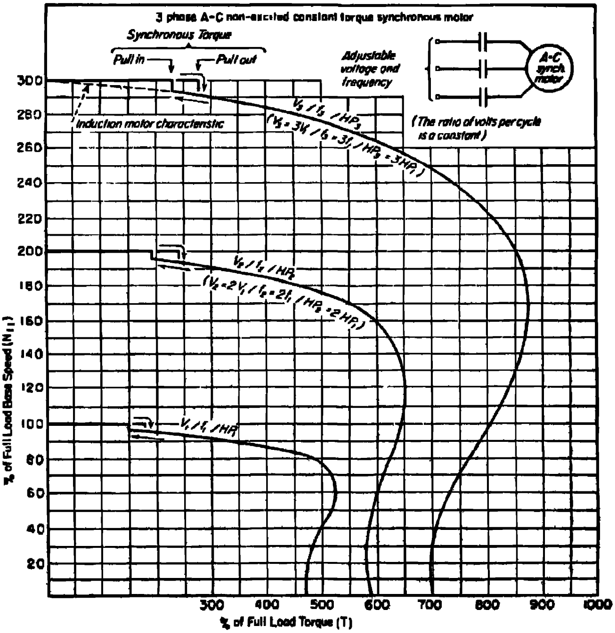
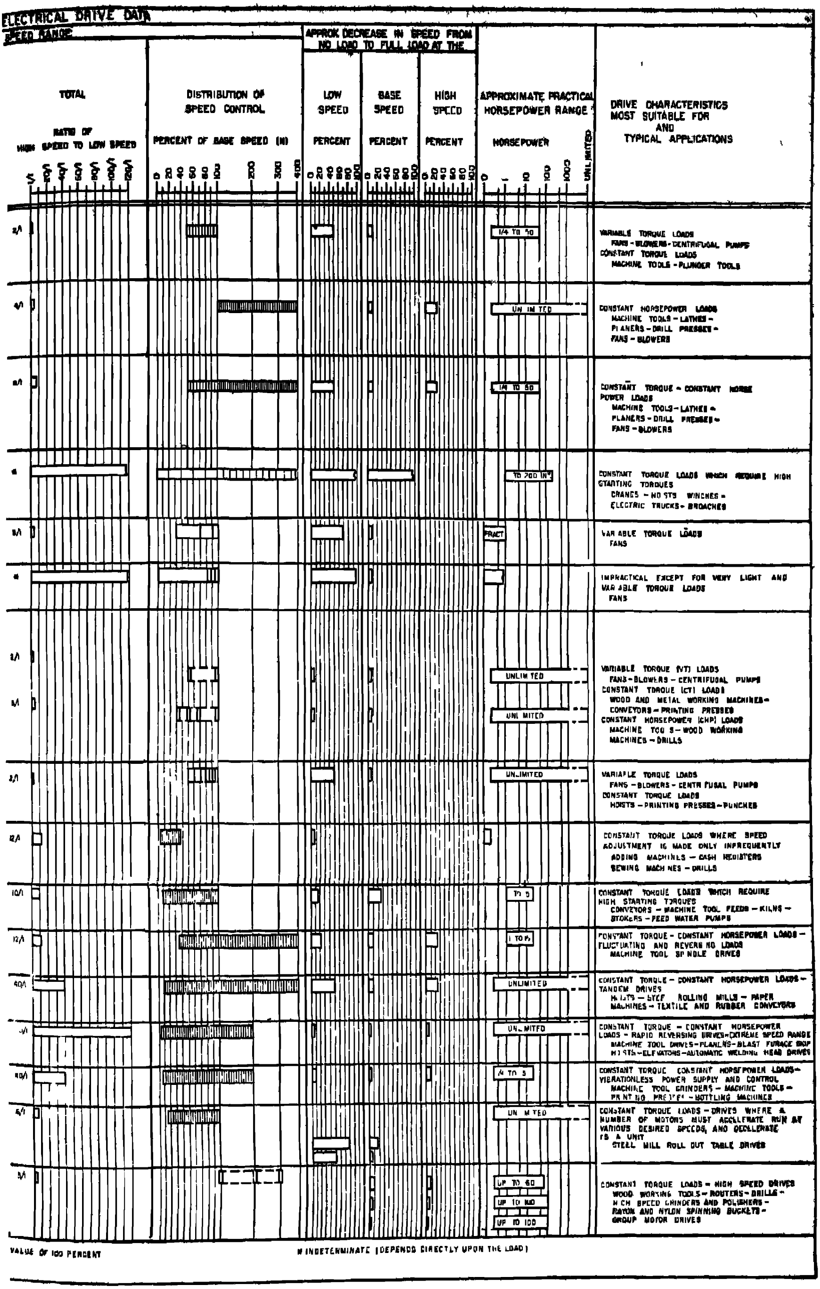


FIG. 16-69. Speed-torque curves of a nonexcited synchronous motor operated on three different frequencies.

POWER SUPPLY	MOTOR	TYPE OF DRIVE CONTROL AND TYPE OF MOTOR AND TYPE OF CONTROLLER	FIG. NUMBER FOR TYPICAL			RELATIVE ORDER OF		DEGREE OF SPEED ADJUSTMENT	ADJUSTABLE SPEED DRIVE MOTOR TYPICAL OBTAINED BY CONTROL OF						
			EQUIPMENT REQUIRED	CIRCUIT DIAGRAM	SPEED-TORQUE CURVES	SIMPLICITY	COST IN PERCENT FOR A 10 HP MOTOR WITH A MAXIMUM SPEED OF 1800 R.P.M.●		ARMATURE RESISTANCE	ARMATURE VOLTAGE	FIELD EXCITATION	PRIMARY VOLTAGE	PRIMARY FREQUENCY	SECONDARY RESISTANCE	POLE GROUPING
D-C	D-C	ARMATURE RESISTANCE CONTROL D-c SHUNT WOUND MOTOR MANUALLY OPERATED RHEOSTAT MANUALLY OPERATED DRUM	16-9	16-10	16-11	3	(B-1750 L-875) 225% 260%	FAIRLY GOOD (IN MODERATE AND DEFINITE STEPS)	2/1	—	—	—	—	—	—
		FIELD EXCITATION CONTROL D-c SHUNT WOUND MOTOR MANUALLY OPERATED RHEOSTAT MAGNETIC STARTER AND FIELD RHEOSTAT	16-13	16-14	16-15	5	(H-1800 B-450) 475% 510%	GOOD (IN SMALL AND DEFINITE STEPS)	—	—	4/1	—	—	—	—
		ARMATURE RESISTANCE AND FIELD EXCITATION CONTROL D-c SHUNT WOUND MOTOR MANUALLY OPERATED RHEOSTAT MANUAL DRUM AND FIELD RHEOSTAT	16-16	16-17	16-18	8	(H-1800 B 450 L-225) 530% 530%	GOOD (IN MODERATE OR SMALL AND DEFINITE STEPS)	2/1	—	4/1	—	—	—	—
		ARMATURE RESISTANCE CONTROL D-c SERIES WOUND MOTOR (INT. RATING) MANUALLY OPERATED DRUM MAGNETIC CONTROLLER AND MASTER SWITCH	16-19	16-20	16-21	9	(B-1750 L *) 300% 430%	COARSE (IN LARGE AND INDEFINITE STEPS)	*	—	—	—	—	—	—
A-C	A-C	PRIMARY VOLTAGE CONTROL A-c SINGLE PHASE CAPACITOR MOTOR AUTOTRANSFORMER AND SELECTOR SWITCH	16-27	16-28	16-29	2	—	COARSE (IN RATHER LARGE STEPS	—	—	—	3/1	—	—	—
		PRIMARY VOLTAGE CONTROL A-c 3 PHASE SQUIRREL-CAGE MOTOR RESISTOR AND DRUM CONTROLLER	16-31	16-32	16-33	7	—	PRACTICALLY NO ADJUSTMENT	—	—	—	*	—	—	—
		POLE GROUPING CONTROL A-c 3 PHASE SQUIRREL-CAGE MULTI-SPEED MOTOR MANUALLY OPERATED DRUM 2 SPEED - 1 WINDING VT 2 SPEED - 1 WINDING CT 2 SPEED - 1 WINDING CHP MAGNETICALLY OPERATED CONTROLLER 4 SPEED - 2 WINDING VT 4 SPEED - 2 WINDING CT 4 SPEED - 2 WINDING CHP	16-36	16-37	16-38	4	(B-1740 L-870) 100% 100% 150% (B-1740 3RD-1159 2ND 870 L-580) 260% 300% 400%	VERY COARSE (ONE, TWO, THREE OR FOUR DEFINITE SPEEDS)	— —	— —	— —	— —	— —	— —	2/1 3/1
		SECONDARY RESISTANCE CONTROL A-c 3 PHASE WOUND-ROTOR MOTOR PRIMARY MAGNETIC STARTER AND SECONDARY MANUALLY OPERATED RHEOSTAT	16-39	16-40	16-4	6	(B-1745 L 875) 210%	FAIRLY GOOD (IN MODERATE AND DEFINITE STEPS)	—	—	—	—	—	2/1	—
A-C OR D-C	A-C AND D-C	ARMATURE RESISTANCE CONTROL A-c AND D-c UNIVERSAL MOTOR WITH CENTRIFUGALLY OPERATED GOVERNOR MANUALLY OPERATED SNAP SWITCH	16-42	16-43	16-44	1	—	VERY GOOD (GOVERNOR ADJUSTED WHEN MOTOR IS AT STANDSTILL)	12/1	—	—	—	—	—	—
A-C	D-C	SERIES ADJUSTABLE VOLTAGE CONTROL D-c SERIES WOUND MOTOR A-c MAGNETIC STARTER AND MANUALLY OPERATED RHEOSTAT IN D-c CIRCUIT	16-46	16-47	16-48	10	(B-1750 L-175) 500%	GOOD (IN SMALL AND DEFINITE STEPS)	—	10/1	—	—	—	—	—
		SELF-EXCITED ADJUSTABLE VOLTAGE CONTROL D-c SHUNT WOUND MOTOR A-c MAGNETIC STARTER AND D-c CONTROLLER AND MANUALLY OPERATED RHEOSTATS	16-49	16-50	16-51	11	(H 1800 B-450 L 150) 1000%	VERY GOOD (IN SMALL STEPS)	—	3/1	4/1	—	—	—	—
		SEPARATELY EXCITED ADJUSTABLE VOLTAGE CONTROL D-c SHUNT WOUND MOTOR A-c MAGNETIC STARTER AND D-c CONTROLLER AND MANUALLY OPERATED RHEOSTATS	16-52	16-53	16-54	12	(H-1800 B 450 L-45) 850%	EXCELLENT (IN SMALL STEPS)	—	10/1	4/1	—	—	—	—
		ADJUSTABLE VOLTAGE CONTROL WITH ROTOTROL D-c SHUNT WOUND MOTOR A-c AUTOSTARTER AND D-c CONTROLLER WITH ROTOTROL AND MANUALLY OPERATED RHEOSTATS	16-55	16-56	16-57	13	(H-1800 B-900 L-15) 1080%	EXCELLENT (ALMOST STEPLESS ADJUSTMENT)	—	60/1	2/1	—	—	—	—
		ELECTRONIC ADJUSTABLE VOLTAGE CONTROL D-c SHUNT WOUND MOTOR MOT-O-TROL AND MANUALLY OPERATED RHEOSTAT	16-60	16-61	16-62	16	(H-1700 B-850 L-43) 1020%	SUPERB (STEPLESS ADJUSTMENT)	—	20/1	2/1	—	—	—	—
	A-C	ADJUSTABLE FREQUENCY CONTROL — M-G SETS A-c 3 PHASE SQUIRREL-CAGE MOTOR PRIMARY MAGNETIC STARTER AND MANUALLY OPERATED RHEOSTATS IN D-c CIRCUIT D-c TO A-c M-G SET MOTOR NOT COMPENSATED D-c TO A-c M-G SET MOTOR COMPENSATED	16-63	16-64	16-65	15	(B-1740 L-290) ★ 1500% 1750%	EXCELLENT (ALMOST STEPLESS ADJUSTMENT)	—	—	—	—	6/1	—	—
		ADJUSTABLE FREQUENCY CONTROL—INDUCTION GENERATOR SET A-c 3 PHASE SQUIRREL-CAGE MOTOR (INDUCTION MOTOR M-G SET) PRIMARY AND SECONDARY MAGNETIC CONTROLLERS A-c 3 PHASE SQUIRREL-CAGE MOTOR (SYNCHRONOUS MOTOR M G SET) PRIMARY AND SECONDARY MAGNETIC CONTROLLERS A-c 3 PHASE SYNCHRONOUS MOTOR (SYNCHRONOUS MOTOR M-G SET) PRIMARY AND SECONDARY MAGNETIC CONTROLLERS	16-67	16-68	16-69	14	(H-5400, B-1800 2ND-3600) 650% ▲ — —	VERY COARSE (TWO OR THREE DEFINITE SPEEDS)	—	—	—	—	3/1	—	—

NOTES: ● MOTORS ARE FOR THREE PHASE 220 VOLT A-C OR 230 VOLT D-C SERVICE. B=BASE, H=HIGH, AND L=LOW SPEED. THE DRIVE WITH THE LOWEST COST IS CONSIDERED TO HAVE A COST
★ PRACTICAL ONLY FOR A GROUP OF DRIVE MOTORS
▲ NOT PRACTICAL BELOW BASE SPEED (USE MULTI SPEED MOTORS BELOW BASE SPEED)

FIG. 16-70. Graphical summary and compari-



son of the adjustable-speed drives available.

For just one or two speeds and where intermediate operating speeds are not required, the induction frequency-changer set is much more economical to use than the adjustable-frequency set which was illustrated in Fig. 16-63.

SUMMARY AND COMPARISON OF ADJUSTABLE-SPEED DRIVE DATA

Figure 16-70 gives a graphical summary and comparison of the adjustable-speed drives that are now available. In selecting the most satisfactory drive for a given application, the cost, speed range, speed in rpm, change in speed from no load to full load, and the size available must all be taken into account.

SECTION 17

HYDRAULIC POWER SYSTEMS

CONTENTS

Fluid Drives (Hydraulic Couplings)..

BY AMERICAN BLOWER CORP., HYDRAULIC COUPLING DIVISION STAFF

- I. Variable-filling–variable-speed design
  - A. Lever-controlled fluid drive
    - 1. Arrangements
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  - B. Pump-controlled fluid drive
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Hydraulic Transmissions—Positive-displacement Type.........BY RANSOM TYLER

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  - D. Pascal's theorem
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- II. Hydrodynamics
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FLUID DRIVES (HYDRAULIC COUPLINGS)

BY AMERICAN BLOWER CORP., HYDRAULIC COUPLING DIVISION STAFF

Hydraulic couplings, more popularly known as "fluid drives," operate on the hydrokinetic principle, using two vaned rotors to transmit power, one of which is connected to the prime mover and the other to the output shaft (Fig. 17-1).

There are basically two commercial fluid drives available—the constant-filling unit popularized in automobiles and the variable-filling unit widely used with constant-speed electric motors for variable-speed industrial drives. The variable-speed units are manufactured in several types according to the duty, as discussed in this subsection.

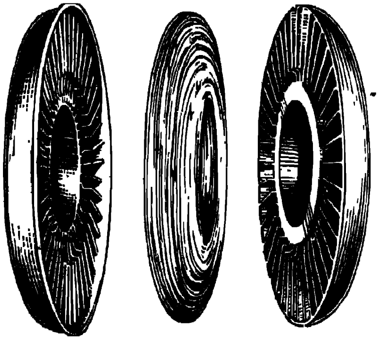

FIG. 17-1. Fluid drive principal parts: (*left to right*) impeller, oil vortex, and runner.

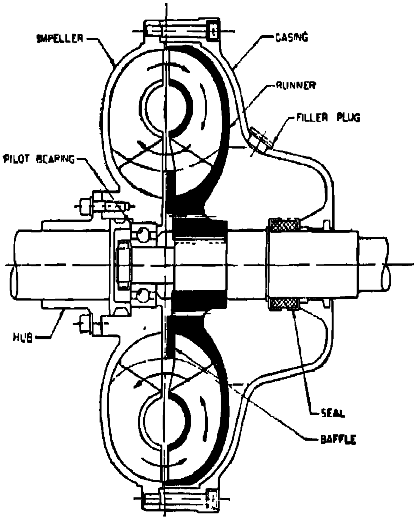

FIG. 17-2. Constant-filling fluid drive.

In the constant-filling design of fluid drive (Fig. 17-2), the rotating elements are initially filled with the proper charge of oil and sealed. The impeller rotates with the motor, and the runner rotates with the output shaft. The casing, which is attached to the impeller, retains the oil and supports the seal at the output shaft.

The uses of this fluid drive are best divided into (1) electric-motor applications and (2) internal-combustion-engine applications. In both cases, fluid drive serves as a flexible cushioned power connection. The ratings and sizes available are given in Tables 17-7 to 17-12, inclusive.

VARIABLE-FILLING–VARIABLE-SPEED DESIGN

Variable-speed fluid drives consist of the basic vaned rotors and a casing with an additional means of adding or removing oil from the vortex and thus varying the torque-transmitting capacity and/or the output speed. The usual means of varying oil filling and consequently speed are discussed below. The lever-controlled design is commonly applied to industrial drives ranging from 1 to 350 hp. The pump-controlled design is commonly applied to industrial drives ranging from 100 to 3,000 hp.

Lever-controlled Fluid Drive. *Principle of Controlling Speed.* The lever-controlled fluid drive is shown by Fig. 17-3. When running, the outer casing acts as a rotating reservoir, storing oil centrifugally against the rim. The movable scoop dips into the rotating reservoir and transfers oil to the working circuit, as required for the desired output speed and/or torque. A continuous circulation of oil takes place. The oil leaves the vortex by calibrated nozzles and circulates through the scoop tube, out of the manifold, through a cooler, and back via the manifold to the working vortex. When the motor is standing still, the outer casing has sufficient capacity to store the oil below the center line and below the labyrinth seal at the manifold.

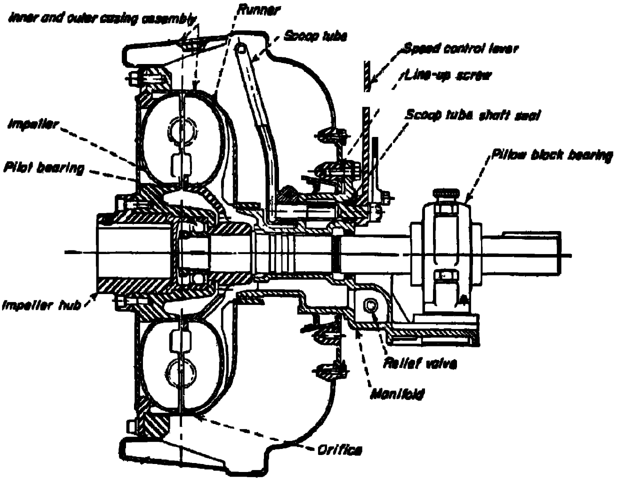

Fig. 17-3. Lever-controlled fluid drive.

The position of the operating lever determines speed. This lever may be manually operated or automatically operated by a diaphragm or piston. It can also be remotely operated by a damper motor and push button.

Arrangements. The arrangements commonly employed are shown by Fig. 17-4. For belt drives the manufacturers recommend the two bearing arrangement on the output shaft.

Applications. Applications and the principal functions of the fluid couplings in each case are listed below. In all cases, constant-speed motors are used.

Table 17-1. Variable-torque (Fan and Centrifugal Pump) Ratings

Fluid-drive size	Use with a-c motors of these hp ratings*							Over-hung wt, lb†	Center of grav-ity,‡
	1,760 rpm	1,450 rpm	1,170 rpm	970 rpm	870 rpm	690 rpm	585 rpm		
9¼	2– 7½	1½– 3	. ....	....	......	....	......	55	⅜
10½	10–15	5 – 7½	. . .	....	....	....	......	75	⅝
12	20–30	10 –15	5– 10	3 – 5	......	....	......	90	15/16
15	40–75	20 –50	15– 25	7½– 15	....	....	......	163	1 1/16
18	....	... .	30– 60	20 – 30	20– 25	....	......	266	1
21	....	....	75–125	40 – 60	30– 50	20– 30	......	416	1 5/16
24	....	....	150–200	75 –100	60–100	40– 50	......	595	⅞
27	....	....	.... . .	125 –200	125–150	60– 75	40– 60	835	⅞
30	... ...	....	....	.. ...	. ...	100–150	75–100	1,151	1 5/16

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* Strip winders, test stands, and similar special-duty machines may require special consideration.

† Including oil.

‡ The center of gravity is given in inches beyond the end of the motor shaft.

Note: The overhung weight of the fluid drive is carried by the motor shaft. For this reason it is advisable to use standard motors with full-size shaft and bearings to carry this weight.

FANS AND CENTRIFUGAL PUMPS—stepless variable speed, no-load motor starting, low power consumption, low noise level, long life

RECIPROCATING PUMPS—stepless variable speed in a compact a-c unit

TEXTILE SPINNING FRAMES—smooth starting, stepless speed control, compactness

DRAWBENCHES—shockless load pickup, controlled acceleration, high efficiency, simplification of control

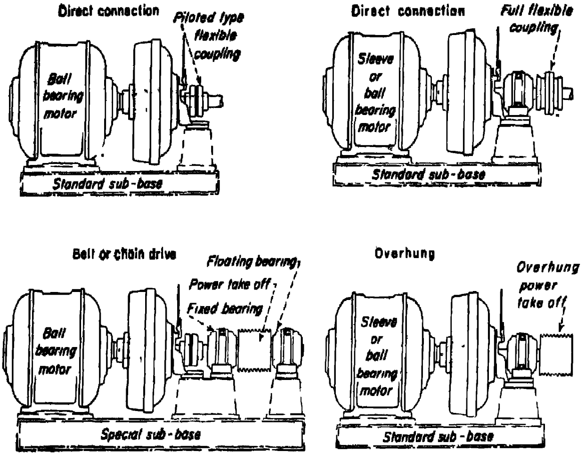


FIG. 17-4. Lever-controlled fluid drive—mounting arrangements.

CENTRIFUGALS, AGITATORS, AND MIXERS—controlled acceleration, low motor heating, no start-stop of the motor

ROTARY KILNS, DRIERS, CRUSHERS, ETC.—easy gradual starting, compact variable-speed unit

STRIP WINDING—smooth even tension, manual control sufficient

PAPER MACHINERY—stepless speed variation, responsive to process controls

Hydraulic Ratings. Ratings of the commercially available lever-controlled units are shown in Tables 17-1 and 17-2.

Table 17-2. Constant-torque Ratings

Fluid-drive size	Use with a-c motors of these hp ratings*							WR^2—impeller and casings†	WR^2 runner†
	1,760 rpm	1,450 rpm	1,170 rpm	970 rpm	870 rpm	690 rpm	585 rpm		
9¼	3- 7½	2- 3	3					8 1	0 8
10½	10-15	5- 7½	3 - 5					11	1 3
12	20-30	10-20	7½- 10	3 - 5				17	1 5
15	40-75	25-50	15 - 30	7½- 15				56	4 8
18		60-75	40 - 60	20 - 30	20- 30			140	12 2
21			75 -125	40 - 75	40- 60	20- 30		289	25 4
24			150 -200	100 -150	75-125	40- 60		530	46 0
27					150-200	75-125	40- 60	930	80 0
30						150-200	75-125	1620	158 0

* Strip winders, test stands, and similar special-duty machines may require special consideration

† Including oil.

NOTE: The overhung weight of the fluid drive is carried by the motor shaft. For this reason it is advisable to use standard motors with full-size shaft and bearings designed to carry this weight.

The capacity of the fluid circuit to transmit power varies as the cube of the impeller speed, the fifth power of the runner diameter, and directly as the slip (difference in speed between runner and impeller) in the lower ranges of slip.

It should be noted that the units are rated lower in some cases for fan and centrifugal-pump duty than for constant torque. This is because the safety factors employed in fan and pump selection (especially fans) tend to increase the actual load horsepower at maximum speed over the design condition, while the safety factors employed in constant-torque-drive selection create a margin of safety between the actual load and the motor rating.

Variable-torque Drives (Fans). For fan drives the speed can be varied from approximately 10 to 97 per cent speed when no stack draft is present and from about 20 to 97 per cent speed where there is a moderate amount of stack draft. The air horsepower at 20 per cent speed is 0.008 of full-speed horsepower (hp varies as rpm cubed) so that horsepower saving is no factor at low speed. The comparison curve (Fig. 17-5) shows the economy of variable-speed fan drive.

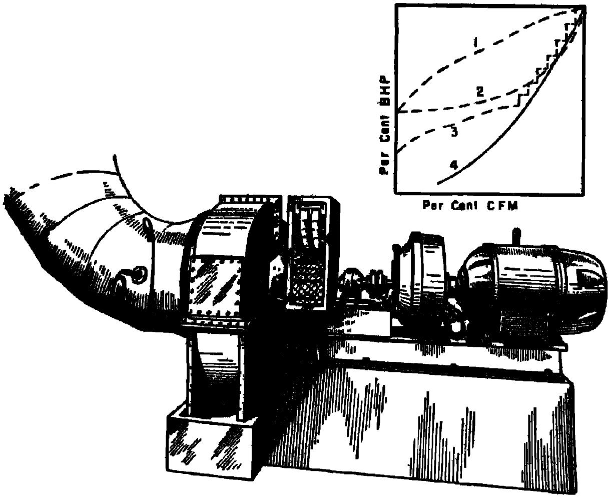


Fig 17-5 Lever-controlled fluid drive—typical fan application, and efficiency-comparison curves of fan-control devices Curve 1—outlet dampers; curve 2—inlet-vane control, curve 3—slip-ring motor and damper, curve 4—fluid drive.

Constant-torque Uses. Constant-torque loads may be divided into two classifications for simplicity.

Class 1—free-rolling loads with some momentum such as paper slitters, wire-drawing machines, centrifugals, etc.

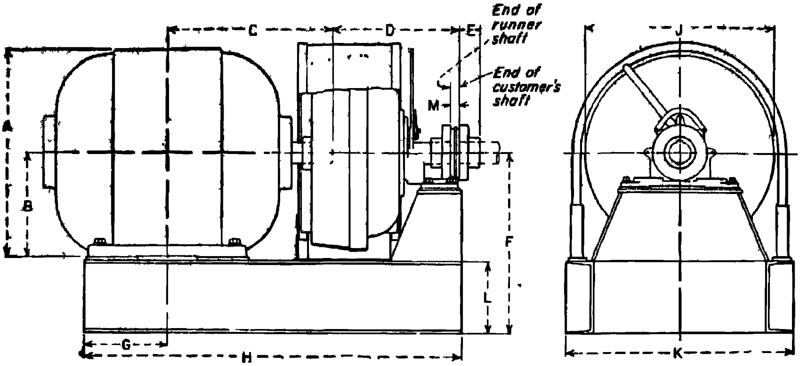
Class 2—loads with little or no momentum or subject to torque fluctuation, such as hammermills, kilns, conveyors, vacuum filters, etc.

It is natural that all loads do not fall clearly into one or the other of these classifications, but for the purpose of analyzing speed-control range they are cited. The control range of a fluid drive is affected by torque fluctuation to a small degree at high speed and to a greater degree at low speed. It is this characteristic which makes it especially suited to overload protection and tensioning drives.

Class 1 loads may be started from standstill and run through creeping speeds very gradually. They may be operated at one-third speed and above continuously (3:1 range).

Class 2 loads may be started from standstill, but because of the usual high break-away torque, accelerate more rapidly. Fluid drives for this duty are recommended for a 2:1 speed range continuously.

Table 17-8



Fluid-drive size	NEMA motor frame	A	B	C	D	E	F	G	H	J	K	L	M
9¼	224	11	5½	9⅞	9 17/32	1⅜	12	4 13/16	26⅛	14	19⅝	6	½
	225	11	5½	10¼			12	5 3/16	26⅞				
	254	12 15/32	6¼	11¾			12¾	5⅝	28 13/16				
10½	254	12 15/32	6¼	11¾	10 9/32	1 11/16	14¾	5⅝	30 9/16	15⅝	21	8	13/32
	284	14	7	13¼			15½	6⅜	32 9/16				
	324	15¾	8	15⅜			16½	6⅞	35 5/16				
	326	15¾	8	16⅛			16½	7⅞	36 15/16				
12	324	15¾	8	15⅜	11⅝	1 11/16	16½	6⅞	36⅛	17 21/32	22⅝	8	13/32
	326	15¾	8	16⅛			16½	7⅜	37⅞				
	364	17 9/16	9	17⅛			17½	7½	38½				
	365	17 9/16	9	17⅝			17½	8	39½				
15	364	17 9/16	9	17⅛	14 9/16	2⅜	19½	7½	40⅞	21⅞	27¼	10	¾
	365	17 9/16	9	17⅝			19½	8	41⅞				
	404	19 11/16	10	19⅛			20½	8¼	43⅝				
	405	19 11/16	10	19⅞			20½	9	45⅛				
	444	21⅞	11	21⅞			21½	9½	47⅝				
	445	21⅞	11	22⅞			21½	10½	49⅝				
	504	24⅞	12½	24⅝			23	10⅜	51				
18	364	17 9/16	9	17⅛	16⅝	2⅜	19½	7½	43 1/16	26 15/32	31⅞	10	¾
	365	17 9/16	9	17⅝			19½	8	44 1/16				
	404	19 11/16	10	19⅛			20½	8¼	45 13/16				
	405	19 11/16	10	19⅞			20½	9	47 5/16				
	444	21⅞	11	21⅞			21½	9½	49 13/16				
	445	21⅞	11	22⅞			21½	10½	51 13/16				
	504	24⅞	12½	24⅝			23	10⅜	53 5/16				
21	404	19 11/16	10	19⅛	18⅞	2⅝	20½	8¼	48¼	30 51/64	36 5/16	10	⅞
	405	19 11/16	10	19⅞			20½	9	49¾				
	444	21⅞	11	21⅞			21½	9½	52 5/16				
	445	21⅞	11	22⅞			21½	10½	54 5/16				
	504	24⅞	12½	24⅝			23	10⅜	55 11/16				
	505	24⅞	12½	26⅛			23	11⅜	58				
24	444	21⅞	11	21⅞	20 21/32	2⅞	23½	9½	54 5/16	35⅛	40⅝	12	1
	445	21⅞	11	22⅞			23½	10½	56 5/16				
	504	24⅞	12½	24⅝			25	10⅝	57 11/16				
	505	24⅞	12½	26⅛			25	11⅜	60 7/16				

Note: The motor selected should have full-size bearings designed for belt pull so that it can carry the overhung weight of the rotating parts. Do not use short-shaft motors without specific approval from the motor manufacturer.

Dimensions. Table 17-3 gives the approximate space required by a NEMA frame motor and a lever-controlled (variable-speed) fluid drive. Construction dimensions will vary slightly.

For dimensions of larger units apply to manufacturers.

Oil Cooler. Variable-speed fluid drives require a cooler to dissipate the heat from power absorbed at reduced speeds of the runner. The rate of oil flow to the cooler is determined by the nozzle diameter in the inner casing and may be set for the design duty. The limit of oil flow is the scoop-tube capacity since the scoop tube must be capable of handling all oil which comes through the nozzles.

Summary. To summarize the mechanical details of the lever-controlled fluid drive:

1. It is normally hub mounted on the driving motor.
2. It uses a light oil which both lubricates the pilot bearing and transmits power.
3. It can be connected to the driven machinery by flexible coupling, chain, or belt.
4. It includes an oil-to-water cooler as standard equipment.
5. It is adaptable to manual, remote, or automatic control.
6. It is built as a standard product for 1 to 350 hp.

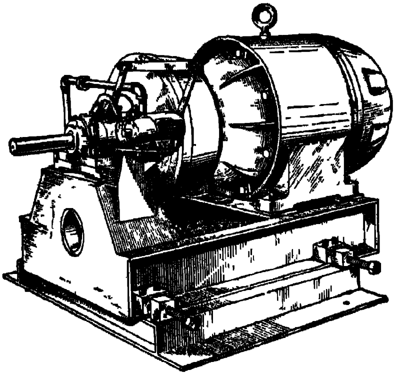


FIG. 17-6. Typical adjustable-speed unit of lever-controlled type.

Pump-controlled Fluid Drive. In general, the characteristics of the lever-controlled fluid drive apply to the pump-controlled unit. The primary differences appear in design.

1. The separate external sump or reservoir.
2. The use of a gear pump to add or remove oil from the working circuit.
3. The fixed scoop tube.
4. Larger size units handle up to 3,000 hp. A cross section (Fig. 17-7) shows the fundamental working parts.

Principle of Controlling Speed. The fixed scoop tube continually circulates oil through the external cooler circuit via the manifold. Speed control is effected by operating a small gear pump in one direction to add oil to the working circuit and, in reverse, to withdraw oil from the working circuit. When the pump is not running, the speed remains fixed. Calibrated nozzles at the periphery meter the oil flow from the vortex to the outer casing.

The pump-controlled unit is built for heavy duty—particularly for power-plant and utility service. Boiler feed-pump drives, mechanical draft, and waterworks pumps represent the bulk of the uses. Constant-torque machines falling within the ratings of these units are reciprocating pumps and compressors. The speed range for constant torque is limited by cooler selection.

Hydraulic Rating. The rating curve and dimensions appear in Tables 17-4, 17-5, and 17-6.

The horsepower transmitted by a full unit is proportional to the slip (difference between runner and impeller speed). The units are normally selected so that slip will not exceed 3 per cent, *i.e.*, the smallest unit that will fall within the 3 per cent slip rating is usually selected. The diagonal size lines in Table 17-4 indicate that hydraulic power increases as the cube of the speed and the fifth power of the diameter. (The size is the diameter of the rotors in inches.) The mechanical limits of commercially available units are listed in Tables 17-5 and 17-6.

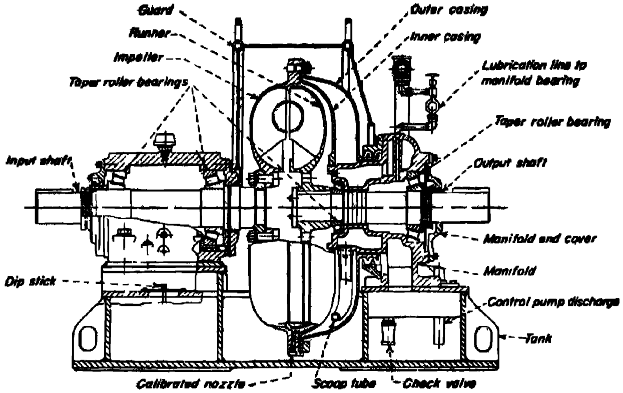
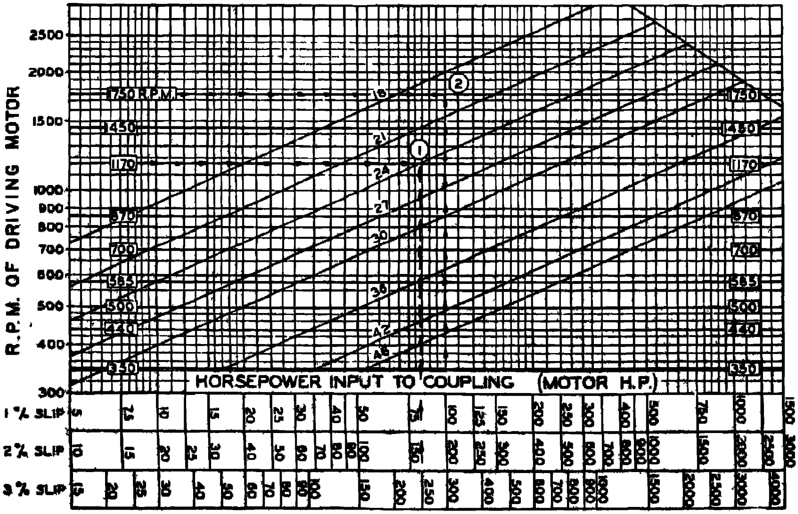


Fig. 17-7. Pump-controlled fluid drive.

The two examples following are shown in dotted lines on Table 17-4.

Example 1: 250 hp at 1,175 rpm—size 24 VS. The intersection of the speed (1,175 rpm) line and horsepower (250 hp at 3 per cent slip) line occurs at a size 24. A size 27 would

Table 17-4. Rating Chart—Pump-controlled Fluid Drive



satisfy this load at 2 per cent slip, but there is no need to use the larger size because the horsepower and rpm limits under the dimension and rating tables show that a size 24 is rated up to 325 hp at 1,550 rpm.

EXAMPLE 2: 300 hp at 1,760 rpm. The intersection of the 1,760-rpm line and the 300 hp at 3 per cent slip line falls between a size 18 and a size 21. Since the size 18 would have more than 3 per cent slip, it is necessary to use the size 21. The horsepower and rpm limits under the dimension and rating tables show that a VS fluid drive has this rating.

A typical installation of a variable-speed fluid drive of the pump-controlled type is shown by Fig. 17-8.

Arrangements. There are two basic arrangements of medium-duty (type VS) fluid drives. Arrangement 4 is a close-coupled design in which the impeller mounts directly on the driving motor. Arrangement 8 is an independent unit which can be set in place or lifted out without disturbing the driving motor on driven machine. A full flexible coupling is required between the motor and the arrangement 8 input shaft and between the output shaft and the driven machine. The ratings and dimensions of these units are given in Table 17-5.

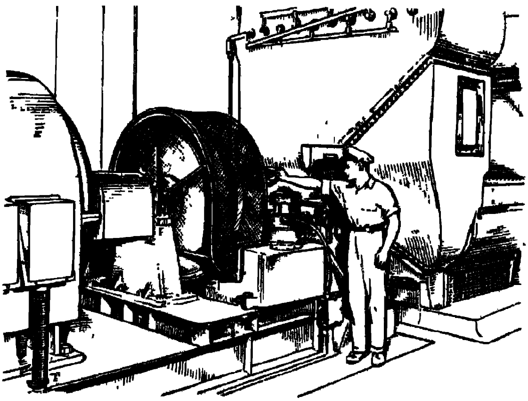


FIG. 17-8. Typical installation induced-draft fan drive.

Dimensions of heavy-duty (type PE) fluid drives are given in Table 17-6. This unit is similar to arrangement 8; it is an independent unit with its own bearings. Full flexible couplings are required to connect the input shaft to the motor and the output shaft to the load.

Duty Classifications for VS and PE Fluid Drives. *Constant-torque duty* ratings cover machines that require full torque output at reduced speeds. These loads are further classified as

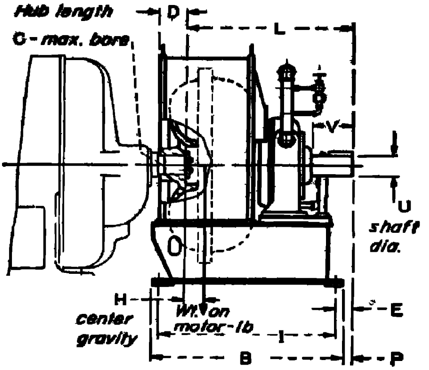
Smooth-rolling loads—steady loads with considerable rolling inertia such as reciprocating pumps, printing presses, pulp grinders, beaters, and lobe blowers.

Fluctuating-torque loads—constant-torque loads subject to torque fluctuation and shock such as ball mills, tube mills, crushers, hammermills, and multicylinder compressors.

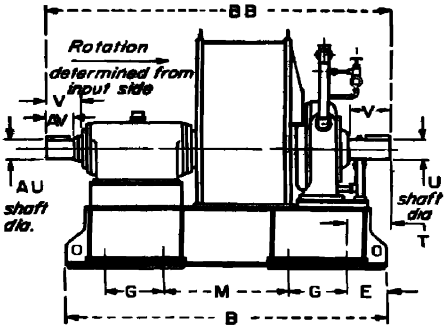
The oil-cooling requirements for these duties should be checked.

Variable-torque duty ratings cover fans, centrifugal pumps, centrifugal compressors, turbo blowers, pure liquid agitators, and similar loads. The horsepower required by these machines falls off rapidly at reduced speeds (usually as rpm^3) so that the average duty is light, and the peak duty is not in excess of motor horsepower.

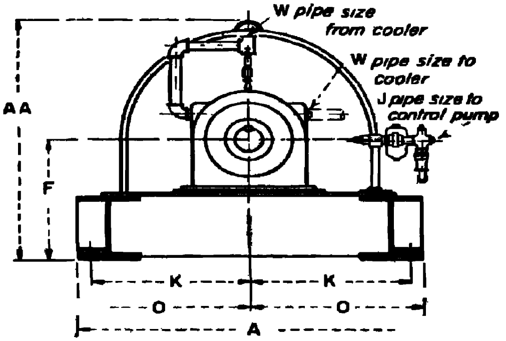
Maximum ratings are based on mechanical considerations only. Fluid drive should be selected from chart of Table 17-4 with initial slip not to exceed 3 per cent. Further limitations may be imposed by oil-circulating capacity.



Arrangement 4



Arrangement 8



Arrangements 4 and 8

FIG. 17-8a. Arrangements 4 and 8 for Table 17-5.

Table 17-5. Dimensions and Mechanical Ratings—Type VS

(For direct connection, see Fig 17-8a)

Ratings weights, thrust, WR^2

Fluid-drive size	Variable-torque duty*		Constant-torque duty				Oil capacity, gal	Data needed for arrangement 4 VS							
			Smooth-rolling loads		Fluctuating-torque loads			Hydraulic thrust, lb						Wt on motor, lb	H, center of gravity
	Max rpm	Hp per 100 rpm	Max rpm	Hp per 100 rpm	Max rpm	Hp per 100 rpm		1 800 rpm	1,500 rpm	1 200 rpm	900 rpm	720 rpm	600 rpm		
18	2,000	12	1 800	8 5	1 500	6 5	4	675	470	300		.		165	$1\frac{1}{8}$
21	1,800	18	1,800	13	1 200	10	7	1,250	870	555	312	.	.	250	$2\frac{1}{8}$
24	1,800	29	1,500	20	1,200	16	10	2,130	1,480	945	530	.	.	350	$2\frac{1}{8}$
27	1,400	37	1,200	26	900	20	15			1,520	855	548		470	$2\frac{1}{4}$
30	1 200	41	900	29	900	23	20			2,320	1 303	835	580	610	$2\frac{1}{2}$
36	1,000	80	900	57	720	44	33				2,710	1 740	1 205	1,020	$2\frac{7}{8}$
42	900	120	750	86	720	66	53				5,000	3,200	2 220	1,590	$3\frac{3}{8}$
48	750	158	600	113	600	88	80								

Dimensions WR^2

Fluid-drive size	AA		A		B		C	D	E		F		Primary WR^2, lb-ft²		Secondary WR^2, lb-ft²	
	Arrt. 4	Arrt 8	Arrt 4	Arrt 8	Arrt. 4	Arrt 8			Arrt 4	Arrt 8	Arrt. 4	Arrt 8	With oil	Dry	With oil	Dry
18	$28\frac{1}{8}$	$29\frac{1}{2}$	38	$35\frac{1}{2}$	$18\frac{15}{16}$	$35\frac{1}{2}$	$2\frac{7}{16}$	$3\frac{1}{4}$	$1\frac{7}{16}$	$4\frac{7}{8}$	$12\frac{1}{2}$	$13\frac{1}{2}$	146	124	16	10
21	$33\frac{1}{8}$	$32\frac{7}{8}$	42	$38\frac{1}{2}$	$20\frac{9}{16}$	$42\frac{5}{8}$	$2\frac{13}{16}$	$3\frac{5}{8}$	$2\frac{1}{2}$	$5\frac{5}{16}$	$14\frac{3}{4}$	$14\frac{1}{2}$	154	133	28	17
24	$37\frac{3}{8}$	38	46	45	$22\frac{1}{2}$	$43\frac{1}{2}$	$3\frac{1}{4}$	$4\frac{3}{8}$	$1\frac{11}{16}$	$5\frac{13}{16}$	17	$17\frac{5}{8}$	205	176	47	24
27	$40\frac{7}{8}$	$40\frac{1}{4}$	$50\frac{1}{2}$	$48\frac{1}{2}$	$23\frac{13}{16}$	$48\frac{1}{2}$	$3\frac{3}{8}$	$4\frac{3}{8}$	$2\frac{7}{16}$	$5\frac{13}{16}$	19	$18\frac{5}{8}$	330	295	99	48
30	$44\frac{3}{4}$	$44\frac{3}{4}$	60	$54\frac{3}{4}$	$25\frac{3}{4}$	$50\frac{1}{4}$	$3\frac{1}{2}$	$5\frac{9}{16}$	$3\frac{1}{16}$	$6\frac{1}{4}$	21	21	570	510	157	95
36	$52\frac{5}{8}$	$51\frac{7}{8}$	74	67	$28\frac{3}{8}$	$57\frac{3}{4}$	$4\frac{1}{8}$	$6\frac{1}{2}$	$3\frac{3}{4}$	$6\frac{7}{8}$	$24\frac{1}{2}$	$23\frac{3}{4}$	1,686	1 555	406	275
42	$60\frac{1}{4}$	$59\frac{1}{2}$	87	77	$32\frac{7}{8}$	$64\frac{1}{2}$	$4\frac{7}{8}$	$7\frac{5}{16}$	$3\frac{9}{16}$	$7\frac{7}{8}$	28	$27\frac{1}{4}$	3,250	2,910	875	632
48		$67\frac{3}{4}$		$81\frac{3}{4}$		76				$7\frac{7}{8}$		$33\frac{1}{4}$	5 400	4 770	1,775	1,145

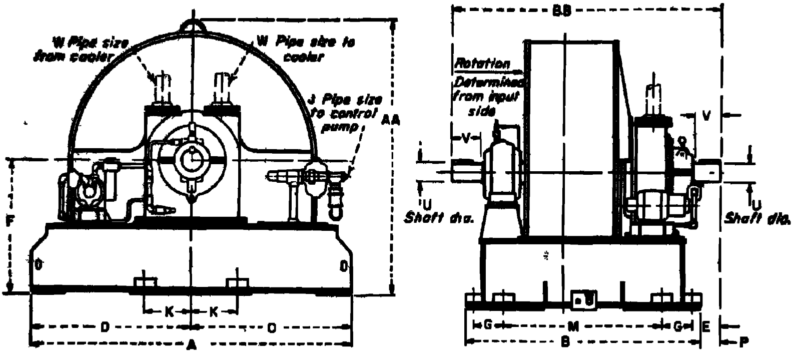
Fluid-drive size	G	I	J	K		L	M	O		P	T	U	AU	V	W	AV	BB
				Arrt 4	Arrt 8			Arrt 4	Arrt 8								
18	$6\frac{1}{4}$	$17\frac{1}{16}$	$\frac{1}{2}$	$15\frac{1}{4}$	$16\frac{1}{4}$	$18\frac{1}{2}$	$13\frac{1}{4}$	19	$17\frac{3}{4}$	$\frac{1}{2}$	$5\frac{1}{4}$	$2\frac{1}{4}$	$2\frac{1}{2}$	$4\frac{7}{8}$	1	$3\frac{13}{16}$	$39\frac{1}{8}$
21	$8\frac{1}{4}$	$18\frac{11}{16}$	$\frac{1}{2}$	$16\frac{1}{8}$	$17\frac{1}{8}$	$21\frac{5}{16}$	$15\frac{1}{2}$	21	$19\frac{1}{4}$	$1\frac{5}{8}$	$4\frac{9}{16}$	$2\frac{5}{8}$	3	6	$1\frac{1}{4}$	$4\frac{5}{16}$	$45\frac{3}{8}$
24	$7\frac{1}{2}$	$20\frac{3}{8}$	$\frac{3}{4}$	$18\frac{1}{4}$	$20\frac{3}{4}$	22	$16\frac{7}{8}$	23	$22\frac{1}{2}$	$1\frac{1}{16}$	$5\frac{1}{2}$	3	3	$5\frac{11}{16}$	$1\frac{1}{4}$	$4\frac{3}{8}$	47
27	$9\frac{1}{4}$	$21\frac{11}{16}$	$\frac{3}{4}$	$20\frac{1}{4}$	$22\frac{3}{4}$	$24\frac{1}{8}$	$18\frac{3}{8}$	$25\frac{1}{4}$	$24\frac{1}{4}$	$1\frac{3}{8}$	$4\frac{7}{16}$	$3\frac{3}{8}$	$3\frac{1}{2}$	$6\frac{15}{16}$	$1\frac{1}{2}$	$5\frac{1}{16}$	$51\frac{3}{4}$
30	9	$23\frac{1}{2}$	1	24	$24\frac{3}{4}$	$26\frac{11}{16}$	$19\frac{3}{4}$	30	$27\frac{3}{8}$	$1\frac{15}{16}$	$6\frac{5}{16}$	$3\frac{3}{4}$	$3\frac{13}{16}$	$7\frac{3}{16}$	$1\frac{1}{2}$	$6\frac{5}{16}$	$55\frac{7}{8}$
36	$9\frac{3}{4}$	$25\frac{7}{8}$	$1\frac{1}{4}$	29	$30\frac{3}{4}$	$30\frac{11}{16}$	$22\frac{1}{2}$	37	$33\frac{1}{2}$	$2\frac{1}{2}$	$7\frac{3}{16}$	$4\frac{5}{16}$	$4\frac{5}{16}$	$8\frac{1}{16}$	2	$6\frac{7}{16}$	$61\frac{15}{16}$
42	11	$30\frac{1}{8}$	$1\frac{1}{2}$	$34\frac{7}{8}$	35	$33\frac{11}{16}$	$26\frac{3}{4}$	$43\frac{1}{2}$	$38\frac{1}{2}$	$1\frac{15}{16}$	$5\frac{7}{16}$	$4\frac{1}{2}$	$4\frac{1}{2}$	$8\frac{13}{16}$	$2\frac{1}{2}$	$6\frac{7}{8}$	$68\frac{7}{16}$
48	$14\frac{3}{4}$		2		$37\frac{3}{8}$		$30\frac{3}{4}$		$40\frac{7}{8}$		$7\frac{1}{4}$	5	5	$8\frac{1}{2}$	3	$8\frac{1}{8}$	$82\frac{3}{8}$

* For definition of variable-torque and constant-torque duty, see page 1271

NOTE: Arrangement 4 VS fluid drive has no internal pilot bearing to take thrust, and therefore the motor bearings must be capable of taking the thrust given in Table 17-5. The motor manufacturer requires thrust, overhung weight, center of gravity, and mounting details in order to design the motor shaft and bearings correctly. The motor bearings in this case must be fixed

Table 17-6. Dimensions and Mechanical Ratings—Type PE

(For direct connection)



Ratings, WR^2

Fluid-drive size	Variable-torque duty		Constant-torque duty				Primary WR^2, lb-ft²		Secondary WR^2, lb-ft²		Oil capacity, gal
			Smooth-rolling loads		Fluctuating-torque loads						
	Max rpm	Hp per 100 rpm	Max rpm	Hp per 100 rpm	Max rpm	Hp per 100 rpm	With oil	Dry	With oil	Dry	
27	1,800	56	1,200	40	1,200	31	330	295	90	48	15
30	1,500	90	1,200	64	1,200	50	570	510	157	95	20
36	1,200	110	900	78	900	61	1 686	1,555	406	275	33
42	1,000	165	750	94	750	73	3 250	2,910	875	632	53
48	900	240	750	170	750	132	5,400	4,770	1,775	1,145	80

Dimensions

Fluid-drive size	AA	A	B	E	F	G	J	K	M	O	P	U	V	W	BB
27	42	53	39 5/16	5 1/2	20 3/4	6 7/16	3/4	8 1/4	23 1/2	26 1/2	4	3 3/8	5 1/16	2	46 1/4
30	45 1/2	57 1/2	44	6 7/16	22 9/16	6 7/16	1	8 7/8	28 1/8	28 3/4	4 15/16	4	6	2 1/2	52 15/16
36	53 1/2	64 3/4	51	6 1/8	26 1/4	7 9/16	1 1/4	9 3/4	32 1/16	32 3/8	4 1/4	4 1/2	6 3/4	2 1/2	57 5/8
42	61	73 1/2	53	8 7/8	29 3/4	6 3/4	1 1/2	10 9/16	35 1/2	36 3/4	6 7/8	5	8 13/16	3	65 5/16
48	68 1/2	81 3/4	60 3/4	9 7/16	34	7 1/4	2	11 1/2	42	40 7/8	7 5/16	5 3/4	8 15/16	3 1/2	73 1/16

Enclosed Design with Speed Controller. A new development is the enclosed design in which the rotors and bearings are completely encased within a stationary housing. This design embodies the following features:

1. The speed controller which makes possible the rapid and accurate response to demands of such control equipment as that used in boiler feed-water-level control.
2. Oil-film-type radial and thrust bearings supported in a split pillow block which is line-bored.
3. A heavy welded-steel stationary housing, split horizontally.
4. Integral shell- and tube-type oil coolers with removable tube bundle.
5. Forced-feed lubrication. All rotating parts and the oil piping are enclosed.

The principle of power transmission is the same for this unit as for the preceding units; the power is transmitted by a whirling vortex of oil. This vortex is set in motion by the impeller, and it imparts its energy to the runner with a smooth continuous flow of power.

Principle of the Speed Controller. There is a continuous fixed supply of oil circulated through the rotating parts by the oil pump. This oil enters the vortex and distributes itself centrifugally in the rotating parts, forming an annular ring of fluid. The speed controller skims the inner surface of this annular ring in the outer casing. Its position sets the depth (amount) of oil in the rotating parts and thus controls output speed and power.

The speed controller can be adjusted for very close accurate control. It is operated by a linkage connected to a shaft that extends through the housing. The automatic or remote manual control device is coupled to this shaft to adjust the speed controller.

The external and internal views of a 250- to 2,600-hp 3,600-rpm unit of this construction, intended for boiler feed-pump service are shown by Figs. 17-9 and 17-10. This unit is connected to the driving motor and to the driven machine by full flexible couplings.

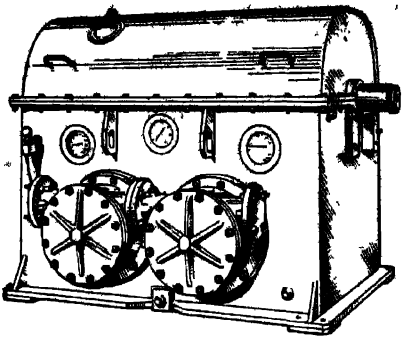


FIG. 17-9. Exterior of 3,600-rpm adjustable-speed unit.

CONSTANT-FILLING FLUID DRIVE

When a fluid drive is mounted on a general-purpose electric motor, certain new and improved characteristics result. The motor is able to start against any load and gain high speed as if started unloaded, and it cannot be stalled by overload. The motor torque can be limited to prevent shear-pin failure (for conveyors). The explanation for these characteristics is demonstrated by Fig. 17-11.

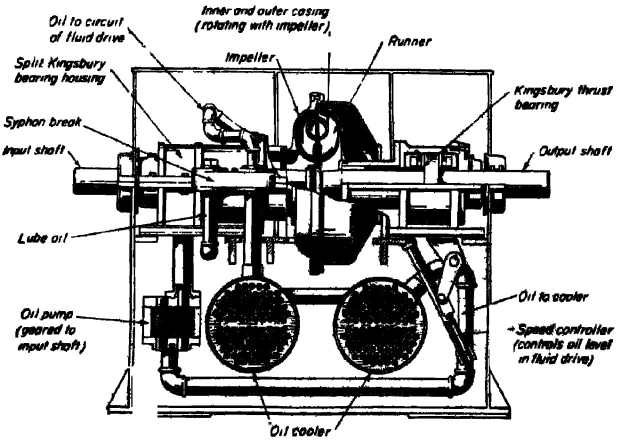

Oil to circuit of fluid drive
Inner and outer casing (rotating with impeller)
Runner
Split Kingsbury bearing housing
Impeller
Kingsbury thrust bearing
Syphon break
Input shaft
Output shaft
Lube oil
Oil to cooler
Oil pump (geared to input shaft)
Speed controller (controls oil level in fluid drive)
Oil cooler

FIG. 17-10. Section through 3,600-rpm adjustable-speed unit.

Principle of Operation. The dotted lines in Fig. 17-11 are the motor torque and motor current plotted against output speed (a general-purpose motor). The heavy line beginning at the origin is the torque required by the fluid drive with the output shaft standing still. It is seen that the motor has to gain speed before the oil in the fluid drive begins to circulate and to require power from the motor. For this reason, the motor starts virtually unloaded and gains a speed where its windings are well ventilated and where the power factor is good and current relatively low before it takes hold of the load.

Applications. The applications commonly employing fluid drive and the benefits derived in each case are listed below. In most cases, the control is simplified, and the rugged general-purpose motor is sufficient.

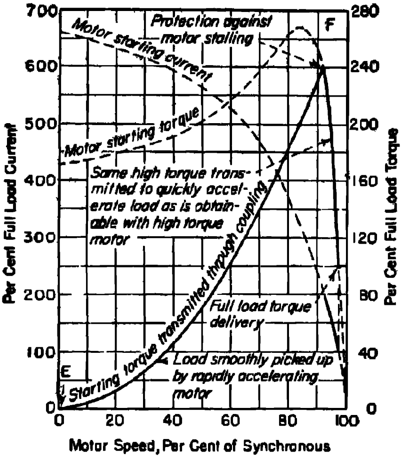


FIG. 17-11. Typical performance curve of constant-speed fluid drive and general-purpose electric motor.

AGITATORS AND MIXERS—no-load starting of the motor against a sticky mix

BALL MILLS (HEAVY STARTING LOADS)—no-load starting of the motor, low current consumption, and in some cases low demand peak (see Fig. 17-12).

CRUSHERS AND HAMMERMILLS, ETC.—shock absorption, prevention of motor-rotor troubles from jams, protection of crusher from motor impact on jams

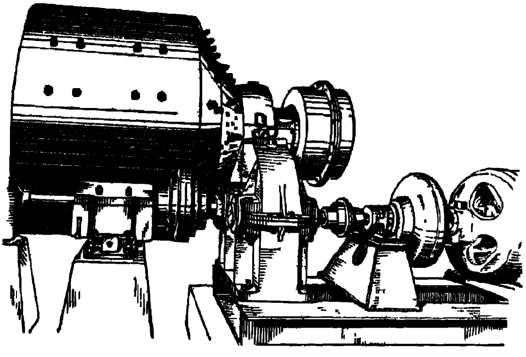


FIG. 17-12. Ball-mill application of constant-filling fluid drive.

FORGING PRESSES—hydraulic application of power

LOCK GATES—smooth starting, low peak stresses in the mechanism, shock absorption when logs get caught in the gates

CONVEYORS—Serves as a shear pin that does not need replacement, offsets bad effects of overmotoring, allows equal loading on multimotored conveyors, and gives smooth starting.

WINCHES AND CAPSTANS—simplifies control, allows the use of single-speed motors, smooth, even take-up on cable, reduces motor heating on heavy starts

Arrangements. Arrangements commonly employed are shown by Fig. 17-4. The direct-coupled arrangement can be rated for full power according to Table 17-7, but the two arrangements using a sheave on the runner shaft must be rated according to the allowable belt-pull reaction on the motor shaft.

Special motors, such as d-c motors and slip-ring motors, offer soft load pickup at low speeds and an extended speed range down to threading speeds owing to increased slip in the fluid drive (up to 100 per cent) at low motor speeds. Paper slitters, hoists, and winches sometime employ this characteristic of fluid drive.

Hydraulic Ratings. The ratings at various speeds are shown in Table 17-7. Since power varies as the cube of the speed (also the fifth power of the diameter), the most economical unit is a high-speed installation. For this reason, this drive is almost invariably mounted on the electric-motor shaft.

Table 17-7

Fluid-drive size	Use with a-c motors of these horsepower ratings*										
	1,750 rpm	1,620 rpm†	1,450 rpm	1,170 rpm	1,080 rpm†	970 rpm	870 rpm	810 rpm†	720 rpm	690 rpm	585 rpm
T-240	20–30	15–25	15–20	7½–10	5–7½	5	3	2–3			
T-270	**40–50**	**30–40**	25–30	15	10–15	7½–10	5–7½	5			
T-300	**60–75**	**50–60**	**40–50**	20–30	20–25	15	10	7½–10			
T-330	**100**		**60–75**	40–50	30–40	20–25	15–20	15			
T-360	**150**		**100**	60–75	50–60	30–40	25–30	20–25			
T-390			**150**	**100**	**75**	50–60	40–50	30–40			
T-420				**125–150**		75	60	50–60	30–40	30	
T-480				**200**		100–150	75–125	75	50–75	**40–60**	
T-540						**200**	150–200	....	100–125	**75–125**	60–75

* Rapid start-stop cycles require special consideration.

† High-slip motor full-load speed.

Where two ratings are shown for one size of fluid drive, any rating between these two can be used

NOTES: Ratings shown in lightface type are satisfactory for continuous operation at full load. The ratings shown in boldface type are for intermittent duty only Since there is no means of cooling the oil except by radiation, continuous operation at full load for these ratings will cause excessive oil-temperature rise and may necessitate use of cooling means. If the motor is only partially loaded or if the full load is applied for short periods of time only, the temperature rise will not be excessive. Severe start-and-stop cycles require special analysis of heat dissipation.

Dimensions and Mechanical Ratings. The dimensions of commercially available units are shown in Tables 17-8 and 17-9.

Internal-combustion-engine Applications of Constant-filling Type. The automobile has so thoroughly demonstrated fluid drive that not much in the way of performance in connection with internal-combustion engines is unexplained today.

The value of this drive can be summarized as (1) smooth load pickup, (2) extended engine speed range, (3) protection of machinery against overload, (4) the engine cannot be stalled, and (5) the engine torsional vibrations are dampened in the oil vortex.

The industrial applications cover switching locomotives, power shovels, cranes, logging machinery, tractors, road rollers, compressors, earth movers, trucks, and almost any engine drive. The ratings of commercially available units are given in Table 17-10.

EXAMPLE 1: 55-hp engine at 1,600 rpm driving a compressor at nearly constant rpm. Under the column for 1,600 rpm, the T-300 is rated at 57 hp. Therefore, this is the correct size

EXAMPLE 2: 275-hp engine at 1,120 rpm for main propulsion of a tugboat (variable-torque load). By interpolation between the 1,100- and 1,150-rpm columns, it is seen that a T-540 has the required capacity.

Hydraulic Ratings Based on Engine Torque. Another method of selection, most commonly used for all but constant-speed drives or marine drives, is on the basis of

Table 17-8. Dimensions, Ratings, Weights, WR^2

(See Arrangements 4A and 4B in Fig. 17-12a)

Fluid-drive size	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	*K*	*M*	*N*	*T*	*U*	*X*	*Y*	*Z*
T-240	13 1/8	3 15/16	2 1/4	3 7/8	4 1/2	9	11 5/8	1 7/8	2 5/8	3 3/8	1 11/16	13/32	162-R	1 7/8	1 11/16	2 3/4	4 5/8
T-270	14 5/8	3 15/16	2 1/4	3 5/8	4 3/4	9	11 5/8	1 13/16	2 5/8	3 3/8	2 3/8	3/4	40-S	1 7/8	2 5/16	4 1/16	6 15/16
T-300	16 3/8	4 5/8	2 1/2	4 1/8	5 5/8	9 3/8	12 7/16	2	2 9/16	3 7/8	2 5/8	3/4	40-S	2 1/8	2 5/16	4 1/16	6 15/16
T-330	17 7/8	5	2 7/8	4 5/16	5 7/8	10 13/16	13 11/16	2 1/16	2 7/8	4 1/2	2 5/8	7/8	50-S	2 1/8	2 11/16	4 13/16	8
T-360	19 1/2	5	2 7/8	4 5/16	6 3/8	10 13/16	13 11/16	2	2 7/8	4 1/2	2 5/8	7/8	50-S	2 3/8	2 11/16	4 13/16	8
T-390	21 9/16	5 7/8	3 3/8	5 1/8	7 1/16	12 7/16	16	2 1/4	3 9/16	5 1/4	2 7/8	1	60-S	2 7/8	3 1/8	5 9/16	8 3/4
T-420	22 11/16	5 7/8	3 3/8	5 5/16	7 3/8	12 11/16	16	2 1/2	3 5/16	5 1/4	2 7/8	1	60-S	2 7/8	3 1/8	5 9/16	8 3/4
T-480	25 7/8	7	3 7/8	5 11/16	9 7/16	15 5/16	18 7/8	2 11/16	3 9/16	6	3 1/4	1	70-S	3 3/8	3 1/2	6 1/8	9 3/4
T-540	28 5/8	7 3/4	4 5/8	7 5/16	11 3/8	18 3/4	22 13/16	4	4 1/16	6 3/4	3 3/4	1 1/4	80-S	3 5/8	4	7 3/16	11 9/16

Fluid-drive size	*AB*	*AC*	*AD*	*AE*	*AF*	*AH*	*AK*	*AL*	Overhung weight on motor	Primary WR^2	Secondary WR^2	Oil capacity, gal	Rated hp	Rated rpm
T-240	8 1/4	3 1/2	3 5/8	2 7/8	2 1/4	15/16	1/2 × 1/4	7 1/16	67	3.2	1.6	1	25	1,800
T-270	8 1/4	3 1/2	3 5/8	2 7/8	2 1/4	15/16	1/2 × 1/4	7 1/8	83	5.4	3.1	1 1/2	50	1,800
T-300	9	3 3/4	4 5/8	3	3 1/2	15/16	1/2 × 1/4	8 9/16	102	9.0	4.9	2.4	75	1,800
T-330	9	3 3/4	4 5/8	3	3 1/2	15/16	1/2 × 1/4	8 5/8	125	14	7.4	3	100	1,800
T-360	10	4	5 7/8	3 1/4	4 3/8	1	5/8 × 5/16	10 1/8	150	21	11	4	150	1,800
T-390	11 1/4	4 1/2	5 7/8	3 3/4	4 1/2	1 1/8	3/4 × 5/8	11 1/16	180	28	16.1	5	150	1,450
T-420	11 1/4	4 1/2	5 7/8	3 3/4	4 1/2	1 1/8	3/4 × 5/8	10 13/16	210	42	23	6 3/4	150	1,200
T-480	13 1/2	5	5 7/8	4 1/2	4 7/16	1 1/8	7/8 × 7/16	11 3/16	290	80	40	10	200	1,200
T-540	13 1/2	5	5 7/8	4 1/2	4 7/16	1 1/8	7/8 × 7/16	11 3/16	420	136	70	14 1/2	200	900

Table 17-9. Dimensions, Ratings, Weights, WR^2

(See Arrangement 4C in Fig 17-12a)

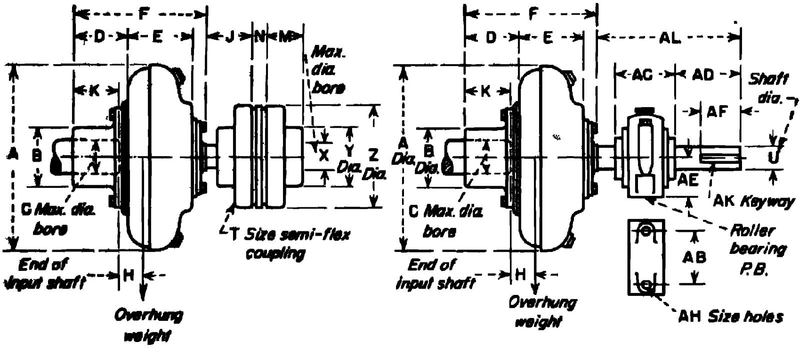
Fluid-drive size	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	*K*	*N*	*T*	*U*	*AA*	*AB*	*AC*	*AD*
T-240	13 1/8	3 15/16	2 1/4	3 7/8	4 1/2	9	10 7/8	1 7/8	1 7/8	3 5/8	13/32	162-R	1 11/16	12 1/4	7 1/2	3 5/8	3/16
T-270	14 5/8	3 15/16	2 1/4	3 5/8	4 3/4	9	11 5/8	1 13/16	2 5/8	3 5/8	3/4	40-S	2 3/16	13 1/4	9	3 3/4	5/16
T-300	16 5/8	4 5/8	2 1/2	4 1/8	5 5/8	9 7/8	12 7/16	2	2 9/16	3 7/8	3/4	40-S	2 3/16	14 5/8	9	3 3/4	5/16
T-330	17 7/8	5	2 7/8	4 9/16	5 7/8	10 13/16	13 11/16	2 1/16	2 7/8	4 1/2	7/8	50-S	2 3/16	18 11/16	9	3 3/4	5/16
T-360	19 1/2	5	2 7/8	4 5/16	6 3/8	10 13/16	13 11/16	2	2 7/8	4 1/2	7/8	50-S	2 3/16	10 1/16	9	3 3/4	5/16
T-390	21 5/16	5 7/8	3 3/8	5 1/8	7 1/16	12 7/16	16	2 1/4	3 9/16	5 1/4	1	60-S	2 7/16	18 11/16	10	4	5/16
T-420	22 11/16	5 7/8	3 3/8	5 5/16	7 3/8	12 11/16	16	2 1/2	3 5/16	5 1/4	1	60-S	2 7/16	18 11/16	10	4	5/16
T-480	25 7/8	7	3 7/8	5 11/16	9 7/16	15 5/16	18 7/8	2 11/16	3 9/16	6	1	70-S	2 15/16	22 9/16	11 1/4	4 1/2	5/16
T-540	28 5/8	7 3/4	4 3/8	7 5/16	11 5/8	18 3/4	22 13/16	4	4 1/16	6 3/4	1 1/4	80-S	3 1/16	26 9/16	13 1/2	5	5/16

Fluid-drive size	*AE*	*AF*	*AG*	*AH*	*AK*	*AL*	Overhung weight on motor	Primary WR^2	Secondary WR^2	Oil capacity, gal	Max hp	Max rpm	Max No. belts and size
T-240	2 5/8	5	6 9/16	3/4	3/8 × 3/16	18 3/4	67	3 2	1 6	1	25	1 800	3-B
T-270	3	6 5/8	7 5/16	15/16	1/2 × 1/4	20 5/8	83	5 4	3 1	1 1/2	50	1 800	6-C
T-300	3	6 5/8	7 5/16	15/16	1/2 × 1/4	21 5/8	102	9 0	4 9	2 4	75	1,800	5-D
T-330	3	9	8 7/8	15/16	1/2 × 1/4	26 1/4	125	14 0	7 4	3	75	1 500	6-D
T-360	3	9	8 7/8	15/16	1/2 × 1/4	26 1/4	150	21	11	4	100	1 500	8-D
T-390	3 1/4	9	9	1	5/8 × 5/16	26 5/8	180	28	16 1	5	100	1 200	8-D
T-420	3 1/4	9	9	1	5/8 × 5/16	26 5/8	210	42	23	6 3/4	125	1,200	8-D
T-480	3 3/4	9 1/4	12 5/8	1 1/8	3/4 × 3/8	31 5/8	290	80	40	10	125	900	10-D
T-540	4 1/2	13	12 1/8	1 1/8	7/8 × 7/16	35 7/8	420	136	70	14 1/2	200	900	12 D

NOTE: Dimensions in inches Do not use for construction. Belt pull = 150 per cent × (torque ÷ radius of sheave).

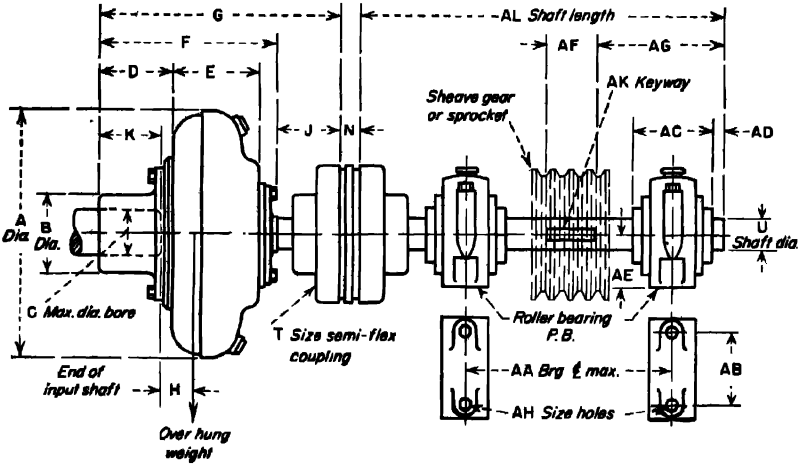
engine torque and the useful operating range of the engine. The method is shown by a typical engine curve (Fig. 17-13).

The engine torque output vs. speed is relatively constant. The drive should be so selected that it allows the engine to be pulled down under full load, only to a safe minimum engine rpm. This point determines the selection of the drive. On the basis of engine torque and minimum engine rpm, commercial sizes can be selected from Table 17-11.



Arrangement 4A

Arrangement 4B



Arrangement 4C

Fig. 17-12a. Arrangements 4A and 4B for Table 17-8 and Arrangement 4C for Table 17-9.

It will be noted that the operating slip is in the order of 5 per cent or less throughout the normal range of engine speeds and that at partial throttle the slip decreases. The square curve labeled "starting torque" is the torque developed at 100 per cent slip by this drive. It is this gradually rising torque which gives such excellent "feathering" action and which permits the engine to be cranked against the connected load.

Applications Based on Engine Torque. Applications usually selected by this method are fluctuating loads which require full engine torque at reduced speeds. Examples are vehicles, excavating equipment, and engine-driven hoists and winches.

Table 17-10. Hydraulic Ratings Based on Rated Engine Bhp

Fluid-drive size	Rated engine hp at governed speed												
	1,500 rpm	1,600 rpm	1,700 rpm	1,800 rpm	1,900 rpm	2,000 rpm	2,200 rpm	2,400 rpm	2,600 rpm	2,800 rpm	3,000 rpm	3,300 rpm	3,600 rpm
T-240	.	19	22	27	32	37	49	58	67	75	82	90	98
T-270	29	34	42	49	56	62	72	82	90	97	104	112	
T-300	48	57	66	75	81	88	100	110	120	129	137		
T-330	75	85	95	102	109	116	128	139	151				
T-360	112	124	134	145	154	162	178	194					
T-390	147	161	172	182	191	200							
T-420	192	210	222	235									

Fluid-drive size	Rated engine hp at governed speed												
	750 rpm	800 rpm	850 rpm	900 rpm	950 rpm	1,000 rpm	1,050 rpm	1,100 rpm	1,150 rpm	1,200 rpm	1,250 rpm	1,300 rpm	1,400 rpm
T-330	..	...	...	..	...	23	26	30	34	39	44	50	62
T-360		..	.	..	31	36	41	47	54	61	70	79	97
T-390		.		38	45	53	62	70	80	90	104	117	134
T-420		..	46	55	65	76	88	100	115	130	144	155	175
T-480		73	88	105	125	145	162	180	195	210	220	230	250
T-540	112	139	165	197	221	245	265	285	302	320	332	345	370

Applications usually selected by this method are (1) those which run at or near constant speed such as generator sets, reciprocating compressors, (2) variable-torque loads such as centrifugal pumps, turbo blowers, ship propulsion, and similar duties where torque falls off rapidly at reduced speeds. To select a fluid drive, look under the vertical columns for the governed speed, and find the horsepower rating. If it falls between two sizes, select the larger size fluid drive (from the column at left). For loads using full engine torque at low speeds, select on the basis of engine torque from Table 17-11.

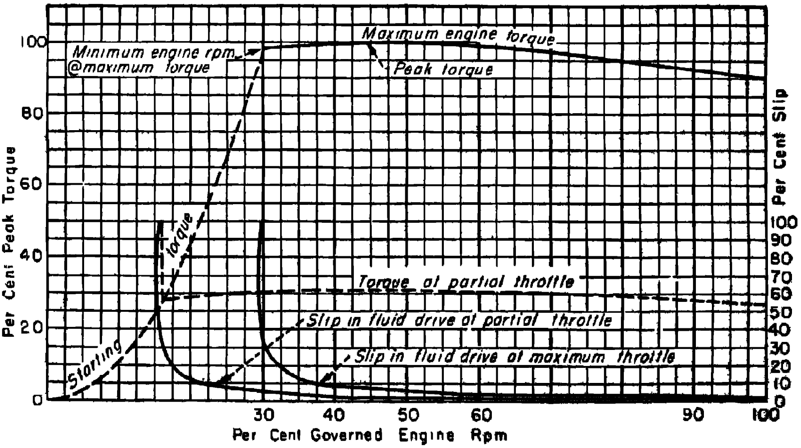


FIG. 17-13. Typical performance curve of fluid drive and internal-combustion engine.

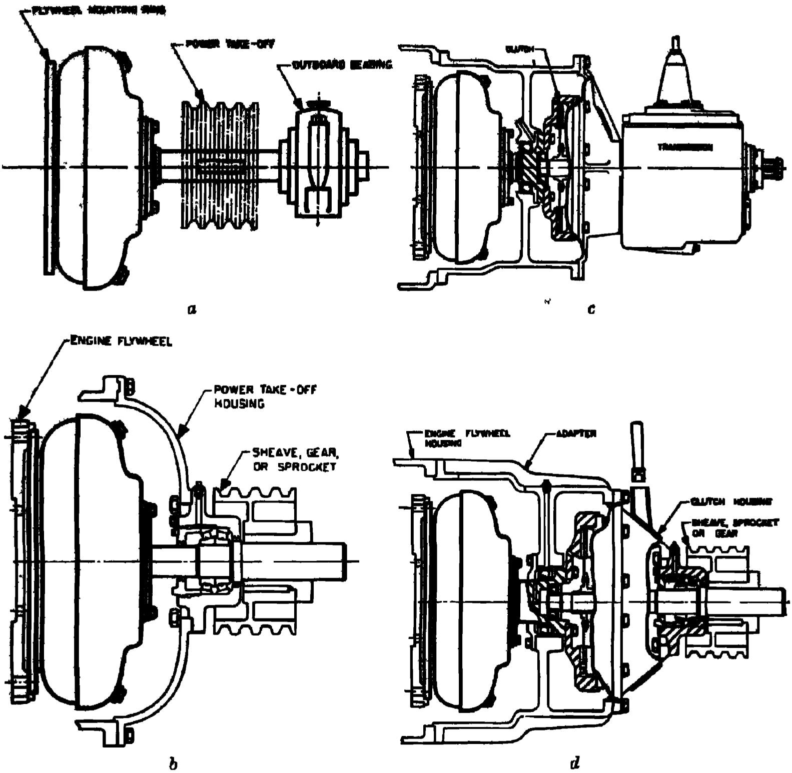


Fig. 17-14. Mounting arrangements—internal-combustion-engine applications of fluid drive.

Table 17-11. Minimum Engine Rpm = Per Cent Speed Factor × Governed Rpm

Type of engine	Per cent speed factors		
	Light duty—cranes, locomotives, trucks, winches	Medium duty—tractors, scrapers, reciprocating pumps	Heavy duty—power shovels, yarders, bulldozers, mine hoists
High-speed-wide-range engines which idle well at less than 20 per cent of governed speed....	35–40	30–40	
Medium-high-speed engines which idle well at less than 30 per cent of governed speed . ..	40–50	35–45	35–40
Slower speed continuous-duty engines which idle at a relatively high per cent of governed speed	.. .	40–50	40–45

Table 17-11 shows the per cent of governed speed at which rated torque must be developed. Table 17-12 shows the torque transmitting capacities of the various size fluid drives at minimum engine rpm.

The selection of a fluid drive by this method is based on (1) the rated torque to be transmitted and (2) the minimum engine rpm at this rated torque. The minimum engine rpm is the point at which the output shaft of the fluid drive stalls under *maximum engine torque*. (Figure 17-13 illustrates this graphically.) To select a fluid drive from the Table 17-11, find the desired minimum engine rpm at maximum torque from the per cent speed factor table. Enter the rating table in the speed column corresponding to the calculated minimum engine rpm, and select the fluid drive that will transmit the engine torque at that speed.

High minimum engine speeds should be avoided as they increase the fuel consumption of the engine and the working temperature of the fluid drive. Some field adjustment is available in the initial oil filling of the drive.

Table 17-12. Rated Torque at Minimum Engine Rpm

Fluid-drive size	Rated torque, ft-lb													
	350 rpm	400 rpm	450 rpm	500 rpm	550 rpm	600 rpm	650 rpm	700 rpm	750 rpm	800 rpm	850 rpm	900 rpm	950 rpm	1,000 rpm
T-240							47	54	62	70	80	90	100	112
T-270						72	86	98	113	128	146	164	184	
T-300					105	123	146	170	195	220	250	280		
T-330				137	164	195	230	266	305	347	395			
T-360			177	220	265	315	370	430	495	560				
T-390		205	260	320	385	460	540	625	720					
T-420	228	300	380	465	565	675	790	910						
T-480	445	590	740	910	1,100									
T-540	790	1 030	1 300											

Arrangements. The mounting arrangements are shown in Fig. 17-14. The engine can be cranked against the connected load with *fluid drive* so that a clutch is not necessary for starting the engine. However, a clutch is required wherever it is necessary to disconnect the engine from the load, as in shifting a transmission or in hoisting (crane operations). The friction clutch can be part of the engine power unit, as is the case with the unit-mounted transmission, or it can be incorporated in the power take-off pulleys or the hoist drum as is often done on oil rigs.

Reference

1. Hydraulic Piping, *Product Eng.*, *Bull.* 90010, November, 1941.

HYDRAULIC TRANSMISSION—POSITIVE DISPLACEMENT TYPE*

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Hydraulic transmissions of the positive-displacement type make use of energy carried in a body of fluid under pressure to perform work. The great use made of this type of transmission has been brought about by the ease with which this energy can be transmitted to points that are otherwise difficult of access and by the precision with which control can be accomplished.

HYDROSTATICS

Since the energy transmitted is primarily pressure energy, transmissions of this type are sometimes known as "hydrostatic transmissions" as contrasted with fluid couplings or torque converters which make use of the principles of hydrodynamics.

Pressure. Pressure is the stress within a fluid. Since a fluid has no tensile or shear strength, pressure is a purely compressive stress. The pressure of a fluid causes a force to be exerted on any solid surface with which it is in contact. This force is always exerted at right angles to each element of any such stationary surface.

Pressure can be produced by squeezing a volume of fluid confined in a vessel or, since the fluid has a definite density, by the elevation of a portion of a continuous body of fluid above another portion. The character of the pressures produced in either manner are identical and, in fact, in hydrostatic transmissions, portions of the fluid are under pressure produced by both methods simultaneously. However, in most transmissions the pressure produced by elevation is insignificant in comparison to the pressure produced by compression, and the net energy from elevation is nil since the transmission operates in a closed circuit.

Atmospheric Pressure. The atmosphere is a fluid that has density and elevation. Thus at any point there exists a definite atmospheric pressure similar in character to hydraulic pressure. While this pressure varies from time to time and from point to point, atmospheric pressure (except for aircraft) can be considered to be the standardized value of 14.7 psi, or 760 mm Hg. This suggests various methods of measuring pressure: (1) in units of force per unit of area, F/A; (2) in units of length or height of the fluid surface above the point considered; or (3) as a multiple of the pressure of a standard atmosphere.

Absolute Pressure. Absolute pressure is the stress above zero in the fluid. Thus, at any free surface the absolute pressure is considered to be equal to atmospheric pressure at that point. (This is slightly modified owing to surface tension, but this has no significance in the theory of transmissions.) Since hydraulic transmissions operate in a closed circuit, all effects of atmospheric pressure are canceled out, and gauge pressure, or the pressure above or below atmospheric, is used rather than absolute pressure. Thus, except that there is a limit to negative gauge pressures, dependent upon the value of atmospheric pressure, the existence of atmospheric pressure can be neglected if all pressures are considered to be gauge pressures.

Pressure Head. Fluid in a vessel or pipe that is under pressure owing to the elevation of another portion of the fluid is spoken of as having a head. Pressure head, then, follows as the pressure (measured in units of height of the given liquid) produced by this head. It is related (see Fig. 17-15) to the pressure as measured in units of force per unit of area through the density of the fluid thus:

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* Superior number refers to specific reference listed at the end of this subsection.

$$(P = F/A = \delta H)$$

where δ = density (weight per unit volume)

H = head in units of length or height of free surface above point considered

Suction Head. The gauge pressure is positive or negative depending upon whether the equivalent free surface is above or below the point to be considered. Pumps or similar apparatus frequently take their supply from bodies of fluid, the free surface of which is below the pump. The suction head is the negative pressure expressed in feet of head at the pump intake. With no flow through the suction line, this is equal to the distance below the pump inlet to the free surface.

The importance of suction head lies in the limit to its value. Since a fluid has no tensile strength, the pressure cannot be less than absolute zero. Thus the absolute limit of suction head is atmospheric pressure at the place in which the device is to be used. Most manufacturers will give a practical limit as to the suction head at which their equipment is capable of satisfactory operation. This is expressed in units of height, so if a significant change is made in atmospheric pressure, such as operation at high elevations above sea level, the limit must be modified accordingly.

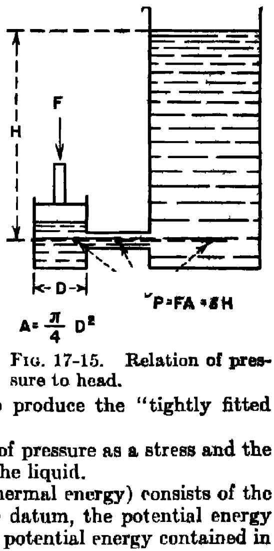


Fig. 17-15. Relation of pressure to head.

Pascal's Theorem. One of the great functions of hydrostatic transmissions is the multiplication of forces. Hydrostatic transmissions are unique in possessing the property of easily converting small forces into great ones or, conversely, short motions into long ones. This property was first recognized by Pascal, who enunciated it thus: "If a closed vessel of fluid have a hole of unit area closed by a closely fitted plunger, and a second hole of one hundred units area also closed by a closely fitted plunger, then one man exerting a force on the smaller plunger can equilibrate the force exerted by one hundred men against the other, and can overcome ninety-nine." While this is the principle of the hydraulic press, as well as many other hydraulic applications, no practical use could be made of it for over a century because, until the invention of the hydraulic packing by Bramah, no way was known to produce the "tightly fitted plunger."

This, of course, follows naturally from the concepts of pressure as a stress and the equality of pressure in all directions at a point within the liquid.

Energy. The energy in a fluid at rest (excluding thermal energy) consists of the sum of the potential energy due to its elevation above datum, the potential energy due to the pressure to which it has been raised, and the potential energy contained in the fluid as elastic energy. A fluid transmission is normally a closed-circuit device, and the fluid used for power transmission passes around the circuit and eventually reaches the point from which it started. Since fluid fills the circuit, each element reacts against the next, and the energy of elevation, so far as the transmission is concerned, is canceled out.

Pressure Energy. The potential energy of pressure was for many years the basis of practically all hydraulic applications. Hydraulic energy was stored in weighted accumulators and used as required. The potential energy thus stored was equivalent to the product of the quantity of fluid which can be discharged by such an accumulator and the pressure which is raised by the accumulator. This is illustrated by Fig. 17-16.

Elastic Energy. While liquids are frequently defined as incompressible fluids, and they are so when contrasted to vapors or gases which are compressible fluids, they all possess a considerable degree of elasticity. The hydraulic fluids used in fluid transmissions have an elasticity which is most conveniently expressed as a percentage change in volume as pressure is changed by a given amount. The fluids used have an elasticity, under working conditions, on the order of ½ per cent per 1,000 psi pressure change. This percentage decreases slightly when pressures in excess of 1,000 psi are used, becoming about ¼ per cent per 1,000 lb pressure change at pressures on the order of 50,000 psi. While the energy stored in this form is not ordinarily a large quantity, it should be noted that this elasticity is roughly 100 times that of steel, and since the normal pressures used are much less than the stresses applied to steel, the energy is stored in the manner of a slightly compressed spring. This property is important in considering the rigidity of a transmission system or in designing the valving to release pressures on a quantity of liquid. This is discussed in greater detail (see Cylinders and Valves).

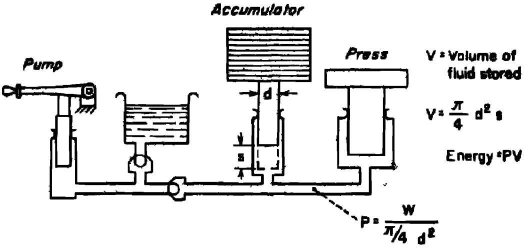

Accumulator
Pump
Press
d
s
V = Volume of fluid stored
$V = \frac{\pi}{4} d^2 s$
Energy = PV
$P = \frac{W}{\pi/4\ d^2}$

FIG. 17-16. Potential energy in an accumulator circuit

HYDRODYNAMICS

While the basic principles utilized by positive-displacement transmissions are those of hydrostatics, it is necessary that motion of the fluid take place in order to obtain work from the transmission. The velocity imparted to the fluid produces dynamic effects which modify the purely hydrostatic principles.

Any mass in motion possesses a kinetic energy equivalent to $\frac{1}{2}(W/g)V^2$ in which W/g is the mass and V the velocity.

Velocity Head. If the moving mass is a fluid, it gains this kinetic energy at the expense of the potential energy it possessed while at rest. The potential energy converted to kinetic energy is equivalent to the weight of the fluid element multiplied by the pressure head (vertical distance it could fall) required to impart the velocity to the fluid.

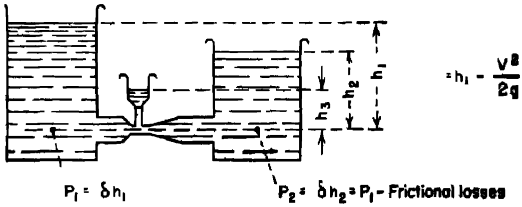


FIG. 17-17. Venturi effect.

Since these quantities must be equal,

$$Wh = WV^2/2g \qquad \text{or} \qquad h = V^2/2g$$

Since $V^2/2g$ is equal to the head required to produce the velocity, this quantity is named the velocity head.

Venturi Effect. Since the velocity head in a moving fluid is produced at the expense of the pressure head, the static pressure at a point in a moving fluid is less than the static pressure in the body of fluid supplying the moving steam.

This is of considerable importance in analyzing the dynamic effects of fluids at high velocities, especially during the critical range of opening or closing valves, etc.

It will be noted from Fig. 17-17 that the kinetic energy of velocity head is reconverted to the potential energy of pressure head when the moving mass is brought to rest. Thus the pressure at one point in a hydraulic circuit may be at a lower pressure than at adjacent points on either side.

Fluid Friction. Whenever a particle moves in relation to an adjacent particle, there is an energy conversion due to friction. This is true regardless of the physical state of the particle—solid, liquid, or gaseous. Therefore, in a moving fluid there is a decrease in pressure equivalent to the amount of energy converted into heat. This subject is covered quantitatively under Piping, Valves, and Fittings (Section 11).

Fluid Friction in Hydraulic Piping. The usual method of sizing hydraulic piping is the rule-of-thumb method based on the velocity of the fluid in the pipe. This is ordinarily adequate for the high-pressure piping in the majority of hydraulic installations, as the lines are short, and the losses are low compared to the pressures to be used. Pipe-line velocities are considered to be within the limits of best practice if held to 12 fps for the smaller pipes (1 in. or smaller) and 15 fps for larger pipes. These limits are exceeded for momentary restrictions or for portions of a cycle in which head loss is of little importance, but 20 fps is seldom exceeded since the danger of dynamic effects in valves, heating of the fluid, and hydraulic shock increases rapidly with velocity.

The rule-of-thumb method should be used with great caution, if at all, for sizing low-pressure or suction lines. When conditions are good, the lines straight and short, and viscosities are not high, velocities of 5 to 7 fps are sometimes used. However, a preferable method is to determine the actual head loss. The head loss in a pipe line is a proportional function of a fluid-friction factor, the equivalent length of the pipe, the density of the fluid, the square of the velocity, and is inversely proportional to the actual inside diameter of the pipe. The fluid-friction factor is a function of Reynolds number (diameter times velocity divided by kinematic viscosity), and the equivalent length takes into account the effect of bends, elbows, valves, etc.[1]

The most convenient method of solving this rather complex equation is by means of curves or alignment charts. Fig. 17-17*a* shows a set of curves made up for one specific viscosity condition (200 SSU). Fig. 17-17*b* is an alignment chart showing equivalent length of fittings.

While the head loss through piping is subject to variation with varying viscosities, it is characteristic of a sharp-edged orifice of proper proportions that it is almost independent of viscosity variations if the critical velocity is exceeded. Fortunately, the critical velocity is within most normal limits except for very small orifices, very small flow, or very high viscosities. The orifice should be no larger than one-quarter the diameter of the pipes on either side, which should be straight and uniform for at least 20 diameters on either side. The length of the orifice should be less than 0.4 its diameter to be considered sharp-edged. When these conditions are met, the pressure drop through the orifice can be approximated by

$$P = 0.015V^2$$

where P = pressure drop, psi

V = velocity through the orifice, fps

This feature of independence of viscosity through normal working range makes the orifice a useful control device where its pressure characteristics permit.

Water. Clear water is used in certain hydraulic presses and similar apparatus. In apparatus for which it is designed, it serves its purpose, being easily available and cheap. However, it must be used with care in climates in which the temperature falls below the freezing point; and the equipment is subject to rusting, especially if idle.

Water Emulsions. Soluble oils are frequently used in conjunction with water to lessen the danger of rust and corrosion. They also provide a degree of lubrication for rubbing surfaces. The dangers of freezing, however, are still present with this fluid, and it has not sufficient lubrication for use in high-speed rotary pumps and like equipment.

Oil. The properties of oils are discussed under Lubrication. Aside from selecting an oil of proper viscosity range, the factors that are important for transmission service are a high viscosity index, good stability for long life, an emulsibility rating that will

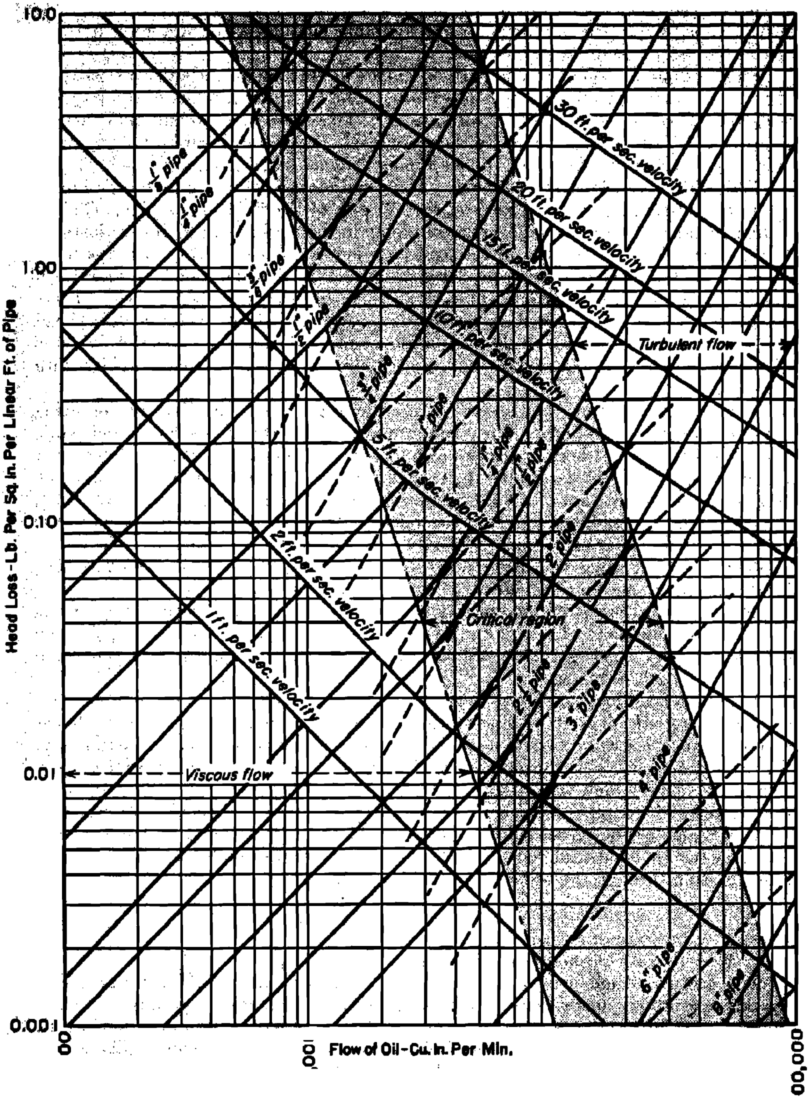
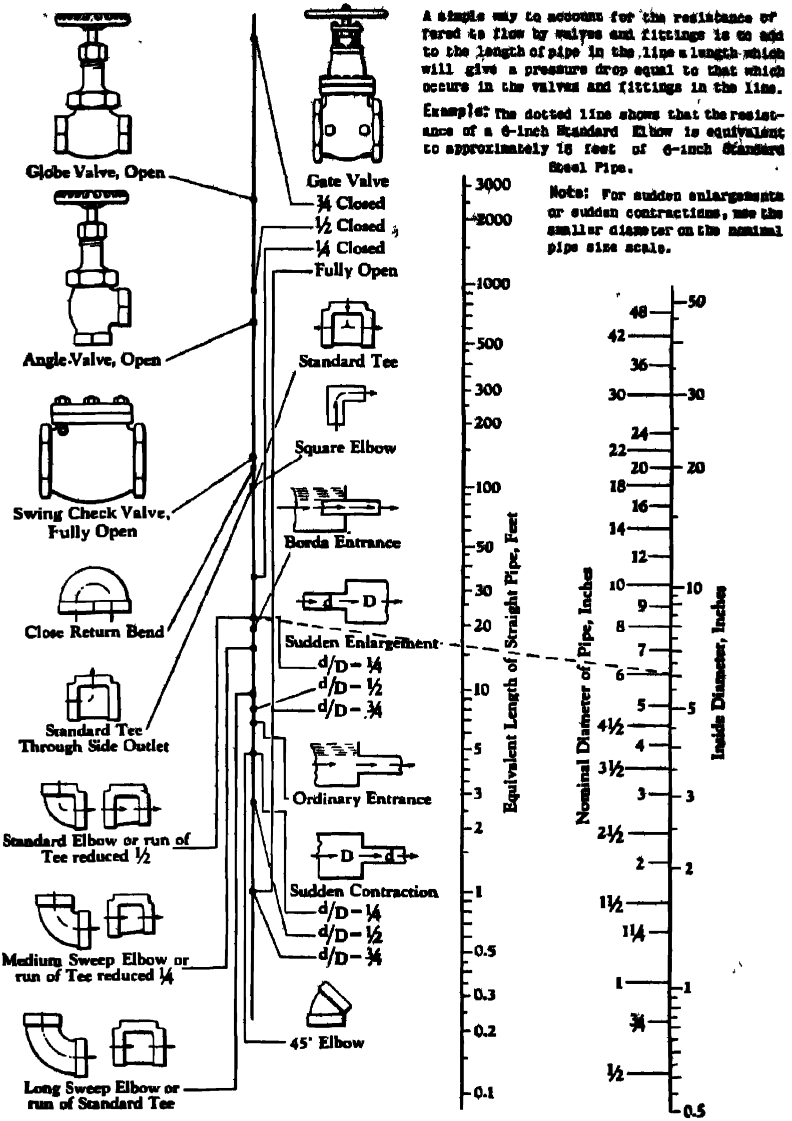


FIG. 17-17*a*. Head loss in pipes for oil at 200 SSU.

assure rapid release of entrained air, and a neutralization number that indicates that it will be free from corrosion elements damaging to the equipment with which it is used.

Oil Maintenance. Just as the initial selection of the oil has an important bearing on the performance and life of many systems, so the maintenance of the properties of the oil is of importance if performance is to be maintained and the expected long life realized. Contaminated oil is the foe of wearing surfaces and a major cause of



A simple way to account for the resistance offered to flow by valves and fittings is to add to the length of pipe in the line a length which will give a pressure drop equal to that which occurs in the valves and fittings in the line.

Example: The dotted line shows that the resistance of a 6-inch Standard Elbow is equivalent to approximately 16 feet of 6-inch Standard Steel Pipe.

Note: For sudden enlargements or sudden contractions, use the smaller diameter on the nominal pipe size scale.

FIG. 17-17b. Equivalent length of pipe fittings. (*Courtesy of Crane Co.*)

sticking of closely fitted valves and controls. Every effort spent in "good housekeeping" about a hydraulic system will be repaid by a longer, more trouble-free life of the equipment. Care should be exercised especially during the initial setup of the equipment or when the units or piping are opened for repair that all open holes are protected against the entrance of dirt or water and that the oil for filling is handled in clean containers and protected against contamination during the process.

If hydraulic systems are to be operated in dusty atmospheres, the breathers should be protected by air filters or equivalent protectors. In excessively humid atmospheres, it may be necessary to pipe the breathers into a drier, cleaner room to prevent condensation contamination.

In spite of all precautions, many systems become contaminated from breakdown of the oil and the entrance of external contaminants in spite of the protective devices. Oil filters are of great value in such cases and often save many times their cost in preventing shutdown and repairs. The manufacturer's recommendations as to the proper type of filter and its location in the circuit should be followed.

Oil oxidizes and breaks down when agitated in contact with the air. This process is greatly accelerated with increasing temperatures. Heat exchangers or coolers are of great benefit in those systems which operate at high temperatures. The installation of water coils in an oil reservoir is generally very inefficient and often dangerous because of condensation on uncovered portions of the coil. Generally a commercial heat exchanger will save enough water to justify its cost over a water coil.

When cooling water is difficult to obtain, use can be made of oil-to-air heat exchangers, a standard unit heater being used for the purpose.

HYDRAULIC POWER EQUIPMENT

The elements forming hydraulic-transmission circuits are pumps, motors, control devices, auxiliary equipment, and instruments. An understanding of the characteristics and limitations of the elements used is essential to an understanding of the circuit.

Positive-displacement Pumps. Of the several hundred general classifications of pumps listed in a mechanical catalogue, only a relatively few are suitable for power-transmission purposes. However, these few types are offered in a considerable variety of designs. There may, therefore, be individual exceptions to some of the following statements concerning characteristics and limitations of any particular type of unit. For power transmissions, pumps are made to be either a constant-displacement type or a variable-displacement type. All principal types of construction have modifications which permit their use either as constant- or variable-displacement pumps. However, some types are essentially better adapted to be constant than variable.

In order to achieve a successful system, it is necessary not only to design the suitable circuit or connections between units but also to select circuit elements having characteristics that most nearly match the requirements of the work to be performed. In comparing the applicability of various types of pumps, the following characteristics should be weighed, and the pump selected that will provide the best over-all performance at the least cost. Among the most desirable performance characteristics are

1. Capacity to deliver the required amount of fluid
2. Pressure rating sufficient to perform the work
3. A high-pressure overload capacity
4. Low slip, so the output will be maintained over wide ranges of load
5. Ruggedness to stand abuse
6. Minimum size
7. Input speed to match the prime mover
8. Dependability to ensure continuity of service
9. Accessibility to facilitate repair
10. Self-priming ability
11. Ease of control for those types having controls
12. Freedom from pulsations in output
13. Quietness
14. High efficiency
15. Cost

Crank and Piston Pumps. (See Fig. 17-18.) The crank and piston pump is the type first developed. It is essentially the hand piston pump modified by the addition of a crankshaft and connecting rod to permit the application of power. Pumps of this type are adaptable to the use of any fluid that can be pumped. They are rugged and dependable and can be easily understood and serviced by any mechanic. They are capable of being manufactured in any practical capacity or pressure rating. They are, however, of considerable weight and bulk in comparison to their capacity, usually require a considerable reduction in speed between prime mover and crankshaft, and frequently require considerable maintenance to keep the packings from leaking, the bearings tight, and the valves in condition. Most of these pumps impart a noticeable pulsation in line flow and pressure.

The principal design modifications are in the number of plungers—single- or double-acting plungers, horizontal or vertical arrangement—and in details of valve design. A modification to produce variable displacement is available which utilizes a variable-ratio element between the crankshaft and the connecting rod.

When well maintained, the volumetric efficiency of these pumps is quite high, but the mechanical efficiency is often rather low, bringing the over-all efficiency on the average to perhaps 60 to 75 per cent.

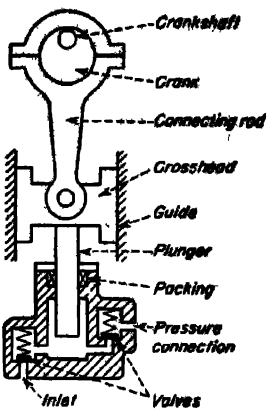


FIG. 17-18. Crank and piston pump.

Gear Pumps. (See Fig. 17-19.) The gear pump is a simple and inexpensive pump. It depends for its action on a pair of gears closely fitted in a housing. Fluid enters the tooth spaces on the inlet side as the teeth disengage, is carried around to the outlet side by the tooth spaces, the teeth forming a seal against the housing, and is forced out of the tooth space by the engagement of the teeth on the pressure side.

Gear pumps are ordinarily limited to medium- or low-pressure service, although some precision pumps are made for pressures up to 2,000 psi. They can be made to pump many different fluids. They are small in relation to their capacity and, when pumping a light lubricating oil, can be made to run at high speeds which permit direct coupling to a standard electric motor. Gear pumps are usually self-priming up to moderate suction heads. The ordinary gear pump imparts a rather severe high-frequency pulsation to the output, and this is one of the principal reasons it is usually rather noisy. Many variations in tooth design and other arrangements have been made to reduce this factor and to reduce trapping of the fluid in the gear spaces. This trapping, which is quite severe with standard involute gear teeth, results in considerable loss in efficiency.

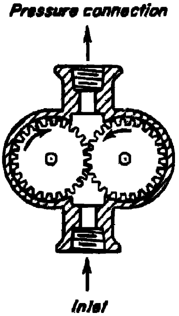


FIG. 17-19. Simple gear pump.

Many modifications in the design are available, among them the use of helical or herringbone teeth; internal gears; special tooth [illegible] some requiring the use of timing gears; and the use of [illegible] wide-face gears to form the "screw" pump. One modification uses a [illegible] form, together with drilled holes to the face and root of each tooth connected with a ported shaft to obtain freedom from trapping and [illegible] of variable output. By rotating the shaft, the oil trapped between the teeth can be discharged into either the pressure or the suction line, thus subtracting from the amount carried around by the teeth and providing a means of varying the delivery. Figure 17-20 illustrates this principle.

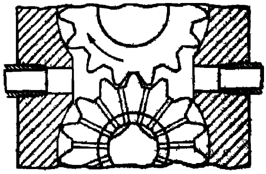


FIG. 17-20. Nontrapping type of gear pump.

Radial-piston Rotary Plunger Pumps. The radial-piston rotary plunger pump was designed primarily to provide an accurate and easily controlled source of variable flow. The pumping action depends upon eccentric action to reciprocate pistons in a rotor as both rotate relative to a ported valve element. The degree of eccentricity can usually be varied by any one of a variety of controls, although pumps of fixed eccentricity are also produced. Figure 17-21 illustrates the principles of this type of design.

Pumps of this type are available in capacity ranges from one to several hundred gpm and pressure ratings from 1,000 to 5,000 psi. Since they were produced primarily to provide a controlled output, their slip is low, and they are well and accurately made, thus providing long life. The input speed is generally suitable for direct coupling to standard motors, and the size is not excessive for industrial purposes. The large number of pistons cuts down the pulsations in the output, and the noise level is tolerable in most cases. Some of the smaller pumps can be considered practically noiseless. This type of pump shares with the axial-piston rotary pump the distinction of having the highest efficiencies obtainable in pumps.

Since these pumps are generally produced in the variable-displacement type, they come equipped with stroke control devices. These may be manually operated or arranged for automatic or remote control responsive to electric switches, line pressure, or the motion of pilot valves.

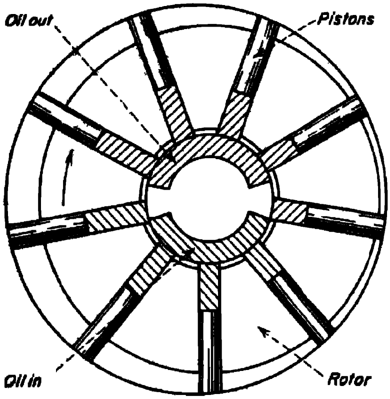


FIG. 17-21. Radial-piston rotary plunger pump.

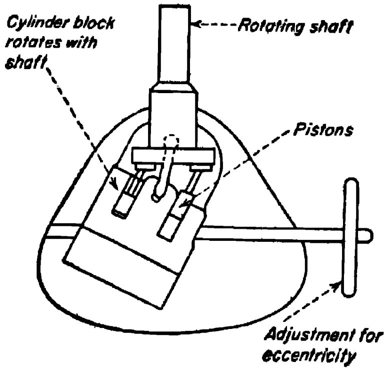

Cylinder block rotates with shaft
Rotating shaft
Pistons
Adjustment for eccentricity

FIG. 17-22. Axial-piston rotary plunger pump.

A design modification makes use of an elliptical cam track to provide two piston impulses per revolution, thus obtaining a hydraulically balanced unit. Displacement variations are obtained by rotating two adjacent cam tracks relative to each other and to the valve ports so that one set of pistons adds to or subtracts from the other.

Axial-piston Rotary Plunger Pumps. Axial-piston rotary plunger pumps obtain their pumping action by the angularity between the axes of a cylinder barrel containing the pistons grouped about its axis and a reaction member controlling the longitudinal position of these pistons. Figure 17-22 illustrates this principle.

Characteristics of these pumps and the uses for which they are especially suitable are very similar to the radial-piston pumps described above.

Vane Pumps. Vane pumps obtain their pumping action from vanes which push the fluid trapped between them across the seal between the suction and pressure ports. The shape of the housing containing the rotor and vanes is such that the vanes are retracted within the rotor while passing the pressure port, so that very little fluid is carried from the pressure to the suction port. Figure 17-23 illustrates a type of vane pump having a cam-shaped housing, or vane track, and two suction and two pressure ports. Each vane thus performs two pumping functions each revolution, and the hydraulic forces on the rotor are balanced. Many designs of vane pumps have been produced. However, the ones having the chief differences from the above are those providing variable displacement. One design uses a circular vane track with variable eccentricity. Another has sliding members and articulated guides to vary the shape of the vane track.

Vane pumps have been limited in capacity by the maximum rubbing speed permissible on the vane tips and by the port area limitations. Pressure has been limited by the strength of the vane and rotor. The maximum capacity of present models is limited to about 55 gpm and 1,000 psi per rotor. These pumps are characterized by low slip, small size, and ability to run at standard motor speeds. They are self-priming and have very smooth output characteristics. They are quiet in operation and have a high efficiency.

While they are somewhat sensitive to pressure overloads and are somewhat short lived under continuous full-load conditions, they may be repaired or replaced easily and also at a low cost.

Motors. Hydraulic motors are of three general types, depending upon the type of work to be done by them. The linear motor, or cylinder, is preferred for operations requiring motion in a straight line or through a small arc, such as presses, lifts, machine-tool slides, pushers and transfers, etc. Rotary motors find their greatest use in driving the shafts of printing presses, conveyors, testing machines, mixers, or any other device in which the speed must be changed while keeping the input speed constant or where a constant output speed is required with a variable-speed input.

Oscillating types of hydraulic motors have been produced for special purposes. However, their use is not widespread, and no designs of this type have been standardized.

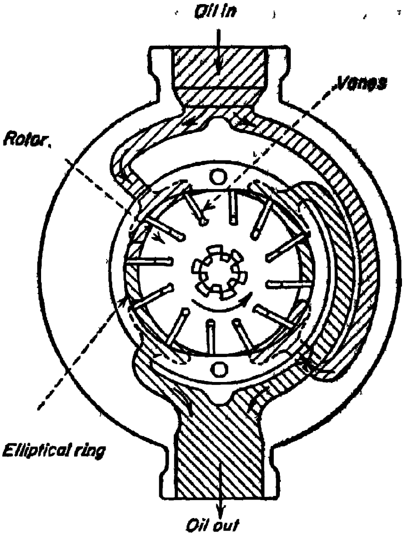


Fig. 17-23. Constant displacement, balanced vane pump.

Hydraulic Cylinders. Hydraulic cylinders are the oldest, simplest, and the most frequently used of the hydraulic-transmission motors. They are available in many standard types and sizes, and special sizes and types are easily and quickly made when needed. Many cylinders are made integral with the machines they operate. Cylinders may be single acting (delivering power in one direction only) or double acting (capable of exerting force in both directions); differential (having unequal areas exposed to the pressure fluid in opposite directions) or nondifferential (with equal areas on each side of the piston); plain or cushioned; and may be had in a variety of mountings or with practically any type of pipe connection (see Fig. 17-24).

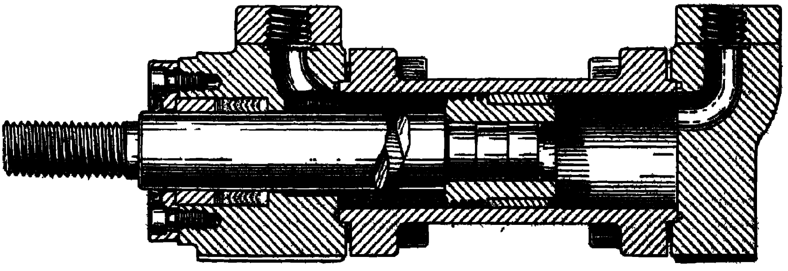


Fig. 17-24. Double-acting hydraulic cylinder.

A cylinder consists essentially of a tube, which may be rough for single-acting cylinders of the most usual type; a smooth ram; a ram-packing gland; a ram packing; a piston and piston packing (not essential for single-acting cylinders); mounting devices; and pipe connections.

The cylinder tube may be of any suitable material and may be open at both ends requiring a head to close it or closed at one end. It is essential that the tube be liquid-tight and have dimensions such that the ram assembly can slide freely. It must, of course, have sufficient strength to withstand the greatest pressures to which it will be subjected. The tube or the heads attached to it carry the mounting devices. These may be lugs, feet, flanges, eyes, trunnions, or recesses for a saddle.

The ram must be smooth, straight, and of uniform diameter so that it can easily slide within the tube. In a single-acting cylinder the diameter of the ram and the pressure used determine the force which the cylinder will exert. In a double-acting cylinder the difference between piston area and the ram area determines the working area in the pulling direction, while the full-bore area may be used for pushing. Valving can be arranged so that the pushing and pulling areas may be connected, permitting selection of either full-bore area or ram area only to provide two tonnages and two rates of speed. The operation with the two areas connected is referred to as "differential advance." The ram must have sufficient strength to withstand the forces and be of sufficient rigidity so that it remains in alignment at all times so that the packing will not be worn.

The packings are perhaps the most critical items in the cylinder. The usual type of ram packing is a material that is impervious to the fluid, can be formed to fit the ram closely, and has sufficient pliability to conform to minor imperfections in the ram surface as it slides through. They are therefore generally made of such materials as leather, treated hemp, asbestos, or a moulded composition or rubber. A gland is often used to apply a pressure on the packing. The adjustment of the gland is critical, since insufficient pressure will permit leakage, while excessive pressure causes excessive friction, heat, and possible packing failure. Several types of self-adjusting packings are available. These are formed in the shape of cups, Vs or Us, so that the pressure against them expands them against the walls of the ram to seal the fluid. When properly installed, these require no adjustment throughout their life.

Self-adjusting packings are also used as piston packings. Expanding metallic piston rings are another frequently used piston packing. While these are subject to some leakage, they have a longer life and require no adjustment or other attention.

Cylinders are easily understood and easily maintained. Their application requires only the selection of a size that will give the required force and stroke. Their limitations are few. They must apply their power in a straight line, and they must not exceed a length at which the ram will deflect excessively. In actual practice this deflection must be kept quite small to avoid excessive packing wear.

For some applications the elasticity of the oil column limits the practical length of the cylinder. Movements requiring rigid coupling, such as machine-tool feeds operating at slow speeds, should be strictly limited as to length. The difficulty appears as jump, chatter, or gallop of the moving member and is caused primarily by variations in the working load. These load variations may be characteristic of the work being done or may arise as a result of the difference between static and moving friction between the moving parts. The characteristics of this jump are greatly affected by the character of the load, the type of guiding, the mass of the moving parts, lubrication, alignment, adjustment of packings, etc., and by the type of circuit used to suppress the jump. A first approximation as to the amplitude and frequency of the jump can be obtained by estimating the working pressure and applying the compressibility factor of ½ per cent per 1,000 psi to the length of the oil column. The maximum stroke length can be used as an approximate length for the oil column if the connecting pipes are not unusually long or large in comparison to the cylinder dimensions. This figure expresses the amplitude of the jump. This length divided by the average speed gives the period, and the frequency is the reciprocal of the period. When the load is subject to sudden periodic variations (as it frequently is in machine-tool feeds when the cutting tooth leaves the cut) when friction is low and the mass large, the length of jump may actually be double the amount calculated by this method. If no provision is made for suppression of the jump by controlling the back pressure, it may even be more than double. When the length of jump is sufficiently short and the frequency is sufficiently high, the momentum causes the jumps to run together, and a smooth feed results. This condition is produced by making S low, P low, or V high. Since S and V are generally established by the requirements of the work, the only solution to a problem of jumpy feed is to increase the cylinder bore or to replace the cylinder with a screw feed (see Fig. 17-25).

Positive-displacement Rotary Motors. Any pump, except those which utilize check valves as flow-controlling means, will theoretically operate as a motor. Thus, the ordinary crank and plunger pump cannot be reversed, as the check valves prevent the flow of pressure fluid into it. However, crank and piston motors or hydraulic engines can and have been built.

The suitability of a hydraulic device for use as a motor is affected chiefly by its slip, or leakage, rate and the relationship of its power losses to the generated power, or its mechanical efficiency.

Since the energy input to a motor to produce one revolution is the product of the volume of fluid required to produce one revolution, or the displacement per revolution, and the pressure, while the energy output is the product of the distance traveled by a point at a given radius from the shaft and the force exerted at this point, the mechanical efficiency is

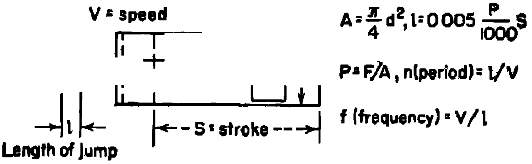


FIG 17-25. Elasticity effects in cylinder feeds.

$$e = 2\pi RF/PV = 2\pi\tau/PV$$

Thus

$$\tau = PVe/2\pi$$

where τ = torque, in -lb = RF
P = pressure, psi
V = displacement per revolution, cu in.
e = mechanical efficiency, per cent

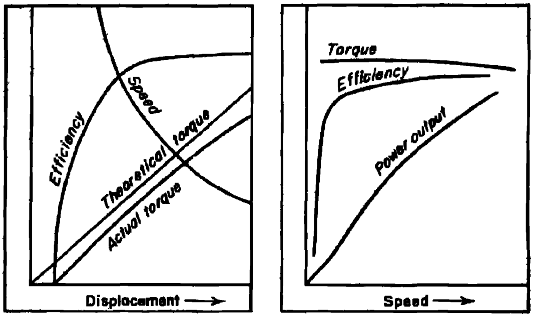
Hydraulic motors are usually applied to drive a machine at a controlled speed or at a controlled torque. They are occasionally used for driving machines in hazardous places, to avoid placing electric motors in the danger area, to provide a suitable means for accelerating very heavy inertia loads, or to provide regenerative dynamic braking of such masses over the entire range to zero speed.

Since hydraulic motors are essentially similar to their corresponding pumps, they can be produced either in the constant-displacement or the variable-displacement types. Furthermore, the loading characteristics of a hydraulic machine limit the pressure to a given value regardless of the speed. Therefore, the torque characteristic of a constant-displacement motor is constant throughout the speed range, and the horsepower characteristic is proportional to the speed. In comparing the performance of this type of motor with other prime movers, it is essential to determine the comparative overload characteristic. Since pressures in a hydraulic system can be easily and accurately controlled, the spread between working loads and overload, which is, in other types of equipment, frequently quite wide to allow for unforeseen peak overloads, may be reduced, since damage to the equipment will not be a result of such overloads. It is therefore essential that the overload and starting characteristics of these motors be checked against a carefully determined requirement of the equipment to be driven.

Speed control of hydraulic motors is usually obtained by controlling the volume of fluid permitted to enter the motor. This may be done either by controlling the stroke of a variable-displacement pump supplying the motor or by means of a flow-control valve in the supply line.

Since speed control is effected by flow control, the effect of slip or internal leakage is to cause the speed to decrease with increasing loads. The effect of slip should be considered over the full working range of speeds. Slip is generally but little affected by speed, and while a variation of a few rpm may be tolerable at 1,000 rpm, it may become a large proportion of the total speed at 50 or 100 rpm and may even cause the drive to stall when full load is applied.

Torque control is achieved either by variation of pressure or by varying the motor displacement. If a variable-displacement motor is used and a constant input volume and pressure maintained, the torque decreases, and the speed increases in approximate proportion as the displacement is reduced. Thus the power output of the motor is maintained approximately constant with changing speed. The approximation lies in the change of efficiency with change of displacement. The efficiency is reduced quite rapidly as the displacement approaches zero. The rate of change of efficiency is a characteristic of the design of each motor but, in general, the speed range obtainable by varying the displacement is strictly limited, probably to between 3:1 and 5:1 (see Fig. 17-26).



Variable displacement motor with constant volume input

Constant displacement motor

FIG. 17-26. Hydraulic motor characteristics.

Crank- and Piston-type Motors. Crank- and piston-type motors are seldom used. They are difficult to construct in multicylinder designs, and thus their torque fluctuates throughout the revolution, and they are large, heavy, and slow-speed machines, thus rather limited in speed range.

Gear-type Motors. Most gear pumps have such low efficiency that they will not run as motors. Except for a few special designs, this type of machine is not suited to operate as a motor. One type is equipped with hydraulic balance ports and is designed for use as a motor. These will operate satisfactorily over a limited range of speeds but at slow speeds become unsteady.

Vane-type Motors. While vane-type motors are well suited for running as motors, most available designs will not start themselves. In most of these units the vanes are held in contact with the vane track by pressure. However, no pressure is generated unless they are initially in contact with the vane track. Centrifugal force is relied upon to cause this initial contact. Therefore a stalled vane motor merely by-passes all pressure past the ends of its vanes. There are, however, designs of vane motors which have auxiliary means for obtaining this initial contact, and these motors have good characteristics.

Piston-type Rotary Motors. At present the preferred type of hydraulic motor is the rotary piston type, both the radial-piston and the axial-piston types being available. These have similar characteristics, much like those of the comparable pump, being capable of satisfactory operation over a wide speed range owing to low slip and high efficiency.

Hydraulic Control Devices. In hydraulic circuits it is necessary to control the direction of flow to the motors, the pressure in any portion of the circuit, the rate of flow to a motor, and to select the portion of the circuit to which pressure is to be directed. Controls can be applied directly to variable-displacement pumps to change direction or amount of flow or to control the pressure. Other controls can be built, either singly or in combination, in valves to be applied in the pipe lines.

Pump Controls. Radial- or axial-piston rotary pumps are capable of being controlled by varying their eccentricity or angularity. Manual or automatic controls can be applied to reverse the direction of flow from the pump, to control the volume of flow from the pump, or to limit and control the pressure generated by the pump. These controls can be furnished to be manually operated, either through screws, cams, levers, etc., or through servomotors powered by hydraulic or other means. They may be power driven or automatic through electric motors or solenoids, hydraulic pilot valves, or automatic pressure regulators.

Pressure-control Valves. Pressure-limiting valves should be applied in every positive-displacement-pump circuit to prevent damage from overpressure. These valves may be of the simple spring-loaded ball or plunger type or more elaborate reverse-flow or piloted type (see Fig. 17-27).

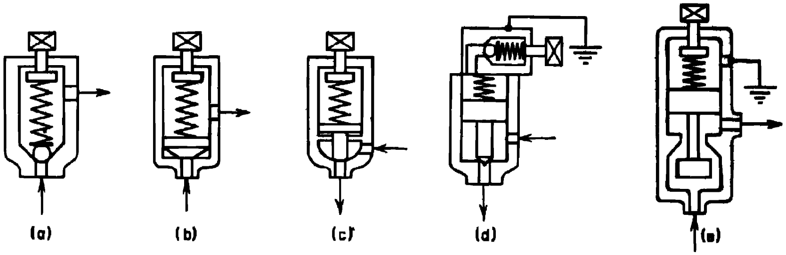


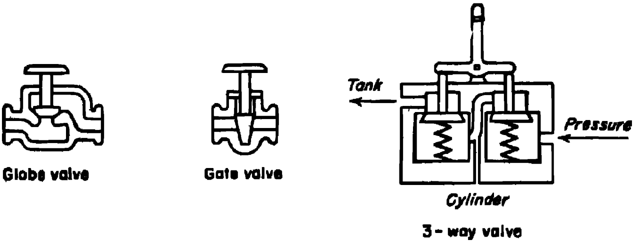
FIG. 17-27. Pressure-control (relief) valves.

The desirable functions to be obtained from a pressure-control valve are (1) accuracy, the maintaining of uniform pressures for long periods or repeating the same setting uniformly time after time; (2) stability, freedom from whistles, chatter, or other disturbances; (3) freedom from peaks or overshooting under suddenly applied loads; (4) close regulation or uniformity of operation under widely varying flow conditions; (5) wide range of pressure adjustment; and (6) sensitivity in the pressure-adjusting means. Of course, all these are not required for every application. For instance, many valves, once set, are not again adjusted. Also, safety valves which are strictly emergency controls need not have a high degree of stability.

The ball-type relief valve *a* is accurate and dependable for very small flow or for use as a safety valve. However, it is incapable of operation continuously at more than a very small flow or low pressure, as it is unstable and will chatter or whistle, introducing destructive oscillation into the line. The plunger type *b*, having a guided plunger, is more stable as the plunger has only one degree of freedom. A valve of this type is often satisfactory for light or medium duty as a safety valve or for performing occasional control functions. However, it is also inherently unstable and subject to whistle or chatter under unfavorable conditions. The reverse-flow type *c* is stabilized by the addition of a pressure-filled dashpot above the pressure chamber. It has proved accurate and sensitive over wide ranges of flow and pressure and has a relatively low peak with suddenly applied pressure when the dashpot is properly fitted. The pilot-operated type *d* is also stable, accurate, and sensitive. It may be subject to high peaks when the pilot valve is small compared to the flow in the circuit. It can be remotely controlled if desired by separation of main valve and pilot valve, and since the spring on the main plunger is light and since it is assisted by the pilot pressure, the valve is small. This type of valve can be connected to a venting valve if desired. This drains the control chamber and opens the valve to practically free flow.

The valve shown at *e* is a type of pressure-reducing valve. When the downstream pressure is increased above the setting, the plunger restricts the flow to prevent the downstream pressure from rising further. If the controlled flow is to be stopped completely, an artificial bleed or additional relief must be incorporated, as it is unwise to rely upon a single seal to seal tightly under all conditions.

Channel-selecting and Directional-control Valves. Valves that are used to select the part of the circuit to which pressure is to be applied or to reverse the direction of oil flow are of two general types. Seated or poppet-type valves, shown by Fig. 17-28, are those such as globe valves, check valves, or the more complex three- or four-way valves having several plungers operated by a common lever. These valves all make use of the heavy pressure available from screws or cams to seat the disk tightly or use the pressure differential itself to accomplish this seating. These valves seal tightly when the seats are in good condition. However, the disks are subject to erosion or wire drawing if operated in the partially closed position and are affected by particles of foreign matter which may lodge between the seat and disk.



Globe valve

Gate valve

3-way valve

FIG. 17-28. Seated valves.

Spool-type valves (Fig. 17-29) rely upon a good fit and finish to limit the leakage past the fits when closed. In a well-made valve, this leakage can be held to almost negligible amounts for most purposes. Valves of this type are hydrostatically balanced and thus move with little effort. Their movement is small so they can be switched rapidly. They can be built to provide innumerable functions, *e.g.*,

Two-way normally open
Two-way normally closed
Three-way open center
Three-way closed center
Four-way open center
Four-way closed center
Four-way open center with one port blocked
Three- and four-way
Five-way
Six-way, etc.

Spool-type valves can be provided with many operators such as hand levers, return springs, or spring offset, centering springs, hydraulic operation by the main pressure or by pilot pressure, solenoid operation, or operation through servomotors.

A special valve of this type may be called a sequence valve. It is essentially a spool-type two-way valve, spring offset to the closed position and utilizing the line pressure to open it. The operation of a valve of this type is not affected by pressure on the downstream side. Thus it will hold a portion of the circuit in isolation from the pump until a given amount of work has been done as indicated by a rise in pressure. Owing to this pressure rise the plunger opens sufficiently to permit fluid to pass to the isolated section without dropping pressure on the upstream side. When the load on the downstream side exceeds the pressure setting of the valve, the valve will open wide and will have no more effect on the circuit until the pressure again drops.

Flow-control Valves. The rate of flow through a pipe can be controlled by restricting the area through which the pressure can flow (see Fig. 17-30). This can be accomplished by a fixed restriction such as an orifice, by an adjustable restriction as provided by a needle valve or a throttle, or by an automatically self-adjusting valve providing compensation for load variations.

A fixed sharp-edged orifice, if not over half the diameter of a straight pipe in which it is placed, is relatively insensitive to changes in viscosity of the fluid when the rate of flow is above the critical. It does, however, have a varying flow characteristic with changes in the pressure differential. A restriction in the form of a pipe of reduced diameter or an orifice having a length of more than four diameters is affected both by changes in viscosity and pressure differential. Various types of adjustable flow-control devices, ranging from needle valves to V-ported throttle valves, have characteristics ranging between the orifice and the pipe.

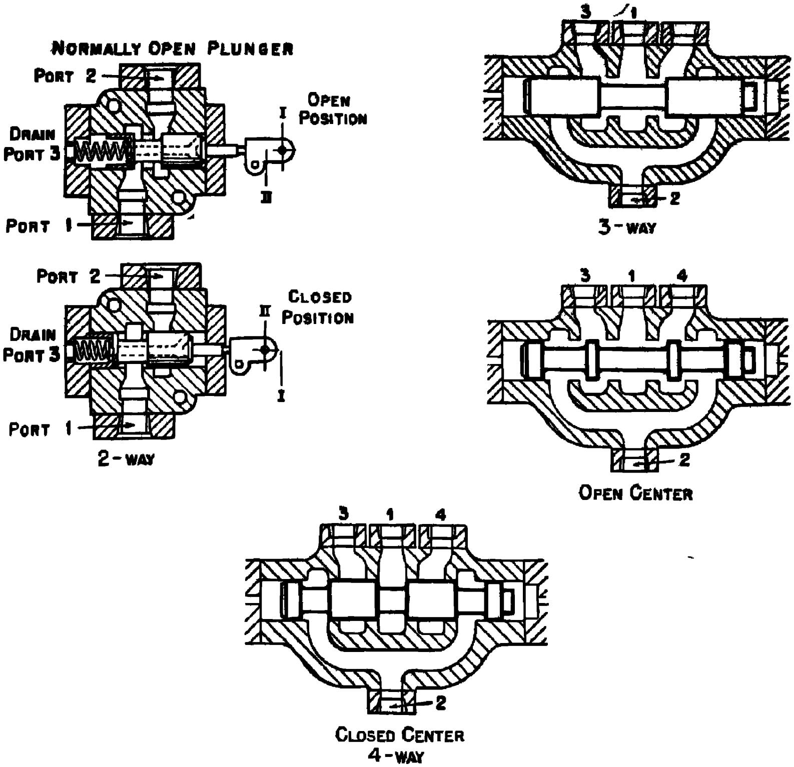


FIG. 17-29 Spool-type valves

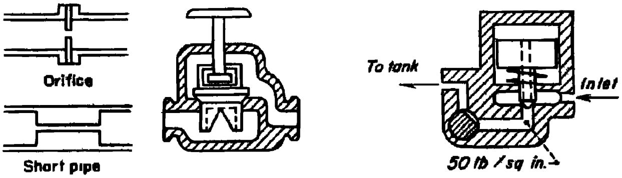
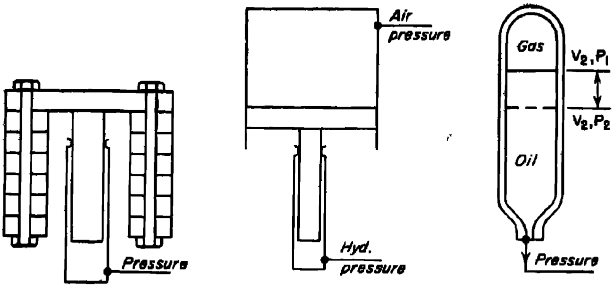


FIG. 17-30. Rate-of-flow controls.

Self-adjusting compensating flow controls consist essentially of an adjustable orifice having a good temperature characteristic, following a pressure-reducing valve.

A valve of this type is placed in the outflow line of the motor so that the orifice will discharge into the reservoir. Thus, since the pressure-reducing valve supplies a constant pressure to the orifice, the flow through the orifice can be regulated and can be independent of temperature and load.

Miscellaneous Controls. A variety of miscellaneous controls such as pressure knockouts, pressure switches, unloading valves, etc., can be obtained. However, these are, generally speaking, modifications of the basic types described. Many special-purpose controls can be designed by combining several basic valves into single units.



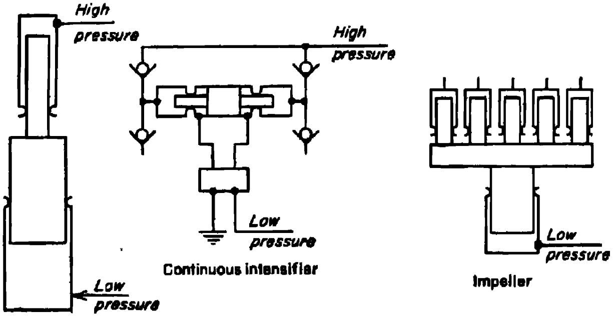
Weighted accumulator

Hydropneumatic accumulator

$$\frac{P_2}{P_1} = \frac{V_1}{P_2}$$

FIG. 17-31. Hydraulic accumulator.

Accumulators and Intensifiers. Only a few of the hydraulic machines now being made use accumulators (see Fig. 17-31). They were originally of great importance since a constant-displacement pump delivers fluid at a constant rate, while the conditions of use dictate its consumption at a variable rate. The accumulator was used to store excess energy during periods when the load is little and to make up the excess requirement when the load is greater than can be obtained from the pump alone.



Continuous intensifier

Impeller

Single stroke intensifier

FIG. 17-32. Hydraulic intensifier.

Accumulators utilize the energy stored in an elevated mass or in a compressed spring or compressed gas to drive a piston or ram which develops pressure on the fluid. Some systems utilize the energy of compressed gas directly upon the surface of the fluid in a confined vessel or through a diaphragm, bladder, or float to provide a storage of fluid under pressure. Accumulators are used chiefly to supply systems that periodically require a great amount of power for short time intervals.

An intensifier is essentially a direct-acting pump delivering high-pressure fluid by the application of low-pressure fluid to connected rams having a differential area (see Fig. 17-32). They are useful to obtain a nonstandard pressure from standard pumps or to provide a stage of high force at slow speed following one of low force at high speed. An occasional use is made of a multiple intensifier (or impeller), consisting of a number of small rams connected to one large ram, to split a single flow into a number of equal, or proportional, flows.

An intensifier can be either a single-stroke type or a continuous automatic pump. If single stroke, it must contain sufficient capacity to perform the required function in one operation and can usually be cut into the circuit by simple valving. If continuous, it is controlled by an automatic valve which causes the intensifier stroke to repeat itself as long as energy is supplied to it.

HYDRAULIC CIRCUITS

While hydraulic principles have been applied to the solution of innumerable problems, the following illustrations will indicate the methods used in the design of a hydraulic circuit. In designing or evaluating any hydraulic circuit, a check list may be of value. The following, while not necessarily complete, will indicate the most important items to consider:

1. Operation.

Will the system operate?

Will it go through a required cycle without stalling or repeating, etc.?

Is the sequence correct?

Is it sufficiently interlocked?

Is there a provision for return fluid where necessary?

Is the pump suction adequate?

Will air be cleared?

Will the pipes drain out during idle periods causing a malfunction on starting?

Is the unit protected against extremes of temperature?

2. Safety and Emergency Operation.

Can the unit be stopped instantly at any time?

How is it started normally?

Can it be damaged by starting after an emergency stop?

How is it recycled?

What "inching" functions are required?

Will any element fall unexpectedly from lack of positive support?

Will creeping take place when idle?

What will happen if a fuse blows?

Will it take care of itself if left unattended?

3. Performance.

Will the equipment do the job?

Has it sufficient force, torque, speed, range, etc.?

Has sufficient allowance been made for lost time?

Has proper allowance been made for off-standard operation, such as operation on 50- or 25-cycle current or the use of standard gear ratios or sheave diameters rather than theoretical values?

Has allowance been made for change in characteristics due to wear?

Has allowance been made for temperature change?

Are all elements and connections of adequate size?

4. Maintenance.

Do the units chosen have the degree of dependability warranted by the importance of their work?

Is their arrangement such that they are accessible for replacement or repair?

Is the oil system protected by adequate filters, coolers, cleaners, breathers, or temperature cutouts?

Is the oil reservoir provided with drainage facilities and a hand hole for cleaning?

Can the oil level be checked conveniently?

Is the equipment "foolproof"?

Is it rugged enough to stand the rough treatment which most industrial equipment sometimes gets?

5. Appearance.

Does the equipment "fit" the job?

Can it be arranged to produce a balanced, harmonious machine?

Are connections and elements simple, and can they be pleasantly arranged?

6. Cost.

Is the over-all cost, including installation, justified?

Pressing Circuits. Pressing is the earliest field of hydraulics and one in which it still excels. It is especially desirable where it is necessary to control the force, length of stroke, or speed of application. The simple press, powered by a pump or an accumulator, has been described. One of the most common problems in this type of equipment is to produce a high-speed approach while the press is idle, then to provide a large force upon the work at a slow speed, required by the nature of the work or to keep the power input within reasonable limits. The two generally used methods are (1) to use auxiliary low pressure for the idle stroke, obtained from a separate pump, a low-pressure accumulator line, or perhaps compressed air, with an automatic switch over to a lower volume high-pressure pump; and (2) the use of gravity or small-size kicker cylinders to cause the platen to approach rapidly with an automatic valve to cut the main cylinder off from connection to the tank to connection to pressure at contact with the work.

Another problem frequently encountered is the obtaining of precision in distance traveled, especially under varying loads. The most accurate and dependable method of accomplishing this is to bring the moving member solidly against a fixed or adjustable positive stop. This generally requires a slowdown before bumping to avoid shock. This is accomplished through operation of pump controls, speed-control valves, or cushions in the cylinder. Other methods are to approach the point of reversal at a speed that will permit precision by quickly reversing a valve; to apply follow-up links or servomotors; or to use cam-operated valves to bring the moving member to rest.

Reciprocating Circuits. Reciprocating circuits comprise the group of planers, grinders, honing machines, transfer conveyors, etc. The reciprocation can be obtained by valving or by reversing the flow from a variable-displacement pump. The main problem is to obtain maximum speed of movement of a heavy mass at uniform speed for the longest possible stroke, then to obtain a smooth shockless reversal in the minimum possible stroke and time. When properly designed, a pilot-controlled valve will provide excellent reversals. However, extreme precision in the point of reversal is not obtained consistently.

Positioning Circuits. Hydraulic cylinders can be equipped with devices to provide accurate positioning of heavy masses against large forces. These systems are illustrated by such systems as ship steering gears or gun-aiming devices. These circuits depend upon differential follow-up devices of many sorts to obtain precision in position. A signal from the controlled element is matched against the signal from the master to set the pump control or control valve to the proper position.

Feed or Speed Controls. Feed control is obtained by adjustment of pump displacement or through flow regulators described under Flow-control Valves. Generally, rapid traverse (obtained from a separate pump) and direction of feed are controlled by auxiliary valving controlled manually, by dogs or cams, or electrically.

Torque Controls. Torque control is utilized in controlling helper drives and sometimes to limit acceleration. Torque control is easily obtained by limiting the pressure applied to the motor by any of the means described under Pressure Controls.

Acceleration Controls. In cases that require a long period of acceleration, it is generally preferable to use an electric pilot motor to change the displacement of a pump at a predetermined rate or to apply a preset pressure, actually a torque control, to limit the rate of acceleration.

Reference

1. Hydraulic Piping, *Product Eng.*, November, 1941.

SECTION 18

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LUBRICATION*

BY ALLEN F. BREWER

Technologist, Engineering, Technical, and Research Division, The Texas Company

PETROLEUM CRUDE OIL

There are two basic types of petroleum crude oil from which lubricants are derived, *viz.*, paraffin and naphthene. Formerly, the base was sometimes regarded as a criterion of performance. Today, however, petroleum technology regards methods of refinement as being of more importance than the base or source when it is desired to predict the performance in service of any oil refined therefrom.

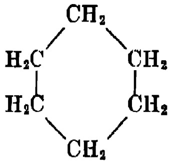
Broadly speaking, base terms are those which have been adopted by the chemist to indicate the nature of the hydrocarbons which predominate in the make-up of an oil. Petroleum research has designated four classifications, *viz.*,

1. Naphthenes
2. Paraffins
3. Asphaltenes
4. Aromatics

All are present in virtually any type of crude petroleum. Usually some one group predominates, according to the geological locality from which the oil has been obtained. For example, oils produced from Pennsylvania, New York, and West Virginia fields contain largely paraffin and naphthenic hydrocarbons. In the mid-continent area, the mixed-base or naphthene-paraffin-asphaltic hydrocarbons predominate. Ohio, Kentucky, and Indiana fields produce such oils, as do also the Kansas, Oklahoma, and North Texas areas. The Gulf coast and California crudes in turn consist of naphthene, asphaltene-paraffin combinations, with the former predominating.

Regardless of these combinations, complex compounds of carbon and hydrogen are involved, in percentages ranging from 83 to 87 for carbon and from 11 to 14 for hydrogen. Some sulfur, oxygen, and nitrogen may also be present according to the source of the crude, but they will usually be considerably below 5 per cent.

In the make-up of the four classifications the so-called saturated and unsaturated hydrocarbons are of primary concern. The former predominate as naphthenes and paraffins. The naphthenes correspond to the formula C_nH_{2n}, being known as closed-ring groups, according to the number of carbon atoms. A typical naphthene is cyclohexane.



Some of the better known paraffin derivatives of petroleum are

Methane, CH_4
Ethane, C_2H_6
Propane, C_3H_8
Butane, C_4H_{10}
Heptane, C_7H_{16}
Octane, C_8H_{18}

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* Superior numbers refer to specific references listed at the end of this subsection.

Among the better known aromatics are

Benzene, C_6H_6
Toluene, $C_6H_5CH_3$

THE REFINING PROCEDURE

The segregation of crudes has become a vital part of refinery operation, for subsequent refinery processing must be planned according to the type of refined products that can be economically produced. By controlled distillation a variety of light products can be fractionated from the naphtha, gasoline, and kerosene stocks through the lighter lubricating-oil range to those heavier distilled oils which are so advantageous for diesel-engine lubrication.

Distillation, however, does not remove all the objectionable unsaturates or readily oxidized hydrocarbons which are present in virtually all crudes. They must be further treated with acids, alkalies, and solvents. Then, to improve the low-temperature characteristics where necessary, the lubricating stock is subjected to dewaxing—removal of those paraffins which are solids at temperatures of 90°F or higher in the pure state.

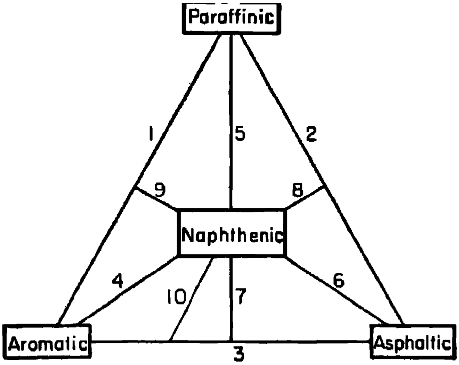

FIG. 18-1.

The object throughout is to produce a series of refined products, regardless of base, which will be free from gum-forming constituents, resistant to oxidation, capable of flowing freely at low temperatures, and heat resisting at high temperatures. Very often the knowledge of how raw stocks will respond to the necessary refining procedure is more important than the source or basic hydrocarbon make-up of the crude. The finished products will be most stable if composed of the so-called saturated paraffin or naphthene hydrocarbons.

Gruse[1] has presented a most enlightening analysis of the arrangement of these hydrocarbon components in petroleum oils. Based on the premise that "all crude oils are mixtures of various types of hydrocarbons," he shows, as indicated in Fig. 18-1, that the naphthenes are of chief importance as they are present in all crude oils and furthermore that the paraffins are almost as widely distributed. To some extent this would discount the preference that has developed for oils by base or source in favor of a more intelligent consideration of their method of refinement for the purpose involved. He further states:

> The four main constituents can be placed on a diagram as shown by Fig. 18-1, and mixtures of the constituents can be expressed by lines joining the constituents. For example, line 1 describes a mixture of paraffins and aromatics; line 2, a mixture of paraffins and asphalt; and line 3, a mixture of aromatics and asphalt; no point on these represents crude oil. Lines 4, 5, and 6 represent mixtures of naphthenes with aromatics, paraffins, and asphalts, respectively; points on these lines will represent real crudes. Line 7 repre-

sents a mixture of naphthenes with aromatics and asphaltic compounds, and lines 8 and 9 have similar obvious significance. Line 10 represents, as does line 7, a mixture of naphthenes with aromatics and asphalt, but a mixture in which the aromatics predominate over the asphaltic constituents; crudes on line 10 would be more aromatic than those on line 7.

This diagram can be read one step further to learn that line 5 denotes a so-called Pennsylvania oil; that line 6 shows a Gulf coast stock; line 8, an oil from the mid-continent area. Thus, naphthenes occur in all, with the paraffins and aromatic or asphaltic compounds included in varying amounts according to their source.

It is usually most important to control the aromatic and asphaltic content in oils refined for lubricating purposes. This is accomplished by proper choice of the crude stock and the use of methods of solvent treatment, especially where the resultant oils are to be used for automotive or diesel engines, the steam turbine, or refrigerating compressor.

The paraffin component is not so readily controlled unless improved methods of dewaxing are employed. It was once thought that the paraffin hydrocarbons were of more importance than they really are, perhaps owing to the fact that the earlier lubricating stocks were obtained from the predominatingly paraffin fields. Today selected methods of solvent refining can be applied to control the ultimate tests to a remarkable degree, in many cases giving a naphthene-base oil many of the physical characteristics of a paraffin.

In turn, by solvent dewaxing, the pour test of paraffin oil can be made comparable with that of a naphthene-base product. For internal-combustion-engine service, the method of refinement is highly important. It is especially essential to control the carbon-residue content and the unsaturate content. The former may be the cause of objectionable deposits which later may be agglomerated by gummy residues, resulting from oxidation of the unsaturates.

TESTS TO DETERMINE LUBRICATING-OIL CHARACTERISTICS

Effective lubrication is attainable by considering the controlling factors and by coordinating these factors as far as is possible. Lubrication begins with the selection of lubricants. In the process of coordination the characteristics are studied with due consideration of the machinery to be lubricated, the conditions under which the lubricants must function, and the means by which they are circulated or applied.

To select lubricants most intelligently, it is necessary to be informed on their physical characteristics and the significance each characteristic bears to the service involved. Laboratory tests are employed to determine these characteristics and also to obtain some idea of how an oil may perform in service.

Gravity. Gravity is of significance as a measure of the relative unit weight of certain petroleum products. It is normally of no value as an index of lubricating ability. In fact, the gravity, as indicated by the Baumé or API scales, is of use only as an indication of the crude base from which such products have been derived. Those from paraffin-base crudes will show a lighter gravity than those from naphthene crudes.

Viscosity. Viscosity is that property possessed by all liquids which serves as a measure of their ability to flow and an indication of the extent to which internal or fluid friction must be considered. The heavier an oil, the higher will be its viscosity, owing to the resistance which the particles or molecules will offer to one another when in sliding contact at a given temperature.

The meaning of viscosity and the means by which it is customarily determined must be understood in order to conform to the lubrication recommendations of machinery builders and attain most effective lubrication with minimum fluid friction. Use of an oil of too low a viscosity or one too light in body may lead to metal-to-metal contact and wear if the oil film is too thin to carry the prevailing loads. Use of too heavy an oil may lead to power losses due to the inherent sluggishness of such an oil.

Viscosity is measured in terms of time by noting the number of seconds required for a certain quantity of oil to flow through an orifice of standardized size at a certain definite temperature. The ASTM has adopted the Saybolt Universal viscosimeter for commercial purpose, with 60 cc as the usual volume of oil involved. Lubricating

Table 18-1. Abridged Viscosity Index*

Viscosity at 100°F SSU	Viscosity at 210°F, SSU																				
	45	50	55	60	65	70	75	80	85	90	95	100	105	110	115	120	125	130	135	140	145
150	130																				
200	73	133																			
250	17	103	135																		
300		73	115	135																	
350		43	96	121	135																
400		13	76	107	125	135															
450			57	93	114	127	135														
500			37	79	103	118	128	135													
550			18	65	92	109	121	129	135	139											
600			0	51	81	101	114	123	129	134	138										
650				37	71	92	107	117	124	130	134	137	139								
700				23	60	83	100	111	119	125	130	134	136	139							
750				9	49	75	92	105	114	121	126	130	133	136	138	139					
800					38	66	85	99	109	116	122	127	130	133	135	137	138				
850					27	57	78	93	104	112	118	123	127	130	133	135	136	138	139		
900					16	49	71	87	99	108	115	120	124	128	130	132	134	136	137	138	139
950					6	40	64	81	94	103	111	116	121	125	128	130	132	134	135	136	137
1,000						31	57	75	88	99	107	113	118	122	125	128	130	132	134	135	136
1,050						23	50	69	83	94	103	110	115	119	123	126	128	130	132	133	134
1,100						14	43	63	78	90	99	106	112	117	120	123	126	128	130	132	133
1,150						5	35	57	73	85	95	103	109	114	118	121	124	126	128	130	131
1,200							28	51	68	81	91	99	106	111	115	119	122	124	127	128	130
1,250							21	45	63	77	87	96	103	108	113	117	120	123	125	127	128
1,300							14	39	58	72	83	92	100	106	110	114	118	121	123	125	127
1,350							7	33	53	68	80	89	97	103	108	112	116	119	121	124	125
1,400							0	27	48	63	76	86	93	100	105	110	114	117	120	122	124
1,450								21	42	59	72	82	90	97	103	108	112	115	118	120	122
1,500								15	37	54	68	79	87	95	100	105	110	113	116	119	121
1,550								9	32	50	64	75	84	92	98	103	108	111	114	117	119
1,600								3	27	46	60	72	81	89	96	101	105	109	113	116	118
1,650									22	41	56	68	78	86	93	99	103	107	111	114	116
1,700									17	37	52	65	75	84	91	96	101	106	109	112	115
1,750									12	32	49	61	72	81	88	94	99	104	107	111	114
1,800									7	28	45	58	69	78	86	92	97	102	106	109	112
1,850									1	23	41	55	66	75	83	90	95	100	104	108	111
1,900										19	37	51	63	73	81	87	93	98	102	106	109
1,950										15	33	48	60	70	78	85	91	96	101	104	108
2,000										10	29	44	57	67	76	83	89	94	99	103	106
2,100										1	21	37	51	62	71	78	85	90	95	99	103
2,200											14	31	45	56	66	74	81	87	92	96	100
2,300											6	24	38	51	61	69	77	83	88	93	97
2,400												17	32	45	56	65	73	79	85	90	94
2,500												10	26	40	51	60	68	75	81	87	91
2,600												3	20	34	46	56	64	72	78	83	88
2,700													14	29	41	51	60	68	74	80	85
2,800													8	23	36	47	56	64	71	77	82
2,900													2	18	31	42	52	60	68	74	79
3,000														12	26	38	48	57	64	71	76
3,200														1	16	29	40	49	57	64	70
3,400															6	20	31	41	50	58	64
3,600																11	23	34	43	51	59
3,800																2	15	26	36	45	53
4,000																	7	19	29	39	47
4,200																		11	22	32	41
4,400																		4	15	26	35
4,600																			9	19	29
4,800																			2	13	23

* Courtesy of *Lubrication;* The Texas Company.

NOTE: The difficulty of obtaining accurate Saybolt Universal values for viscosities lower than 45 SSU at 210°F is so great that the viscosity-index table is not extended below that limit. The accuracy of the viscosity index may be increased by the use of the kinematic viscosimeter at 210°F for oils having viscosities between 40 and 75 SSU at 210°F since the conversion of kinematic viscosity in this range at 210°F to Saybolt viscosity is more reliable than the actual Saybolt determination. To show which viscosity-index values are in this critical range, dotted lines have been used to set it apart from the rest of the table.

oils for plant service are generally measured at 100 and/or 210°F. The viscosity of lighter oils (turbine grade, for example) is measured at 100°F. Heavier products such as steam-cylinder oils or gear lubricants are tested at 210°F. Accordingly, reference to an oil as being of 300 or 500 SSU viscosity at 100°F means that 300 or 500 sec, respectively, would be required for the standard volume to flow through the

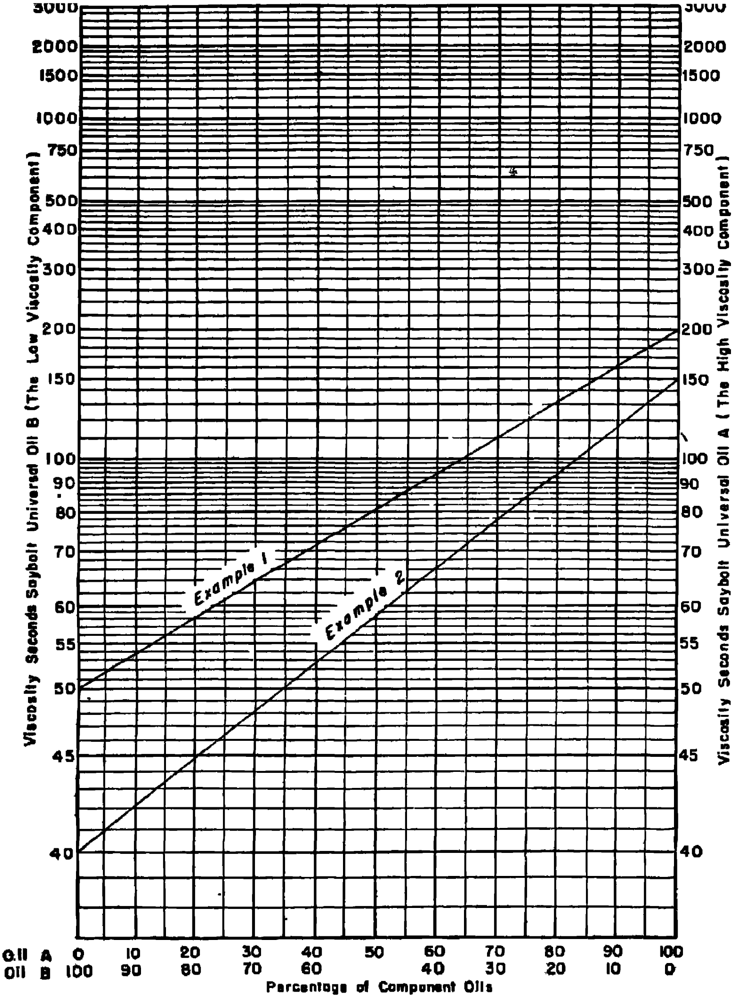

FIG. 18-2. Viscosity blending chart. (*Courtesy of Lubrication, The Texas Company.*)

Saybolt Universal viscosimeter at that temperature. A viscosity blending chart is shown by Fig. 18-2 with instructions for its use given in Fig. 18-2*a*. Data for estimating viscosity index is shown in Table 18-1.

Flash Point. The flash point of an oil is that temperature at which inflammable vapors will be given off to a sufficient degree to ignite momentarily when exposed to a naked flame. Flash point is of importance when an oil may have to be exposed to high temperatures, *e.g.*, when considering whether an adequate lubricating film will be maintained on engine cylinder walls without undue loss by vaporization. Lubricating oils for internal-combustion engines will have a flash-point range of from 325 to 425°F, according to the base of the oil and its viscosity. Naphthenic-base oils will show a somewhat lower flash point than those of paraffin base of the same viscosity. Flash point also increases with the viscosity of oils from the same source. Accordingly, a 500-viscosity product will in general show a somewhat higher flash point than a 200- or 300-viscosity oil of the same base.

The flash point always should be sufficiently high to ensure that there will not be any hazard under storage due to vaporization of the more highly volatile constituents. Good quality lubricating oils of reputable manufacture will adequately meet this requirement since all will range above 325°F, which in turn is far above any extreme temperature that might occur under storage.

HOW TO USE THE VISCOSITY BLENDING CHART

In order to adapt the viscosity-temperature chart as a blending guide, we have used that portion of the chart between 0 and 100°F, covering a viscosity range of 37 to 3,000—the range ordinarily encountered. Let the left vertical represent 100 per cent of the lighter oil, or Oil B. Let the right vertical, represent 100 per cent of the heavier oil, or Oil A. By plotting the known viscosities of any two oils at the same temperature on these respective verticals and connecting them by a straight line, points along this connecting diagonal show the viscosities at the given temperature according to the proportion of the respective oils in the blend.

Example 1. How much oil of 50 SSU at 100°F (Oil B) must be blended with a 200 SSU at 100°F (Oil A) to give a 100 second viscosity oil?

Draw a line between the 50 and 200 viscosity points on the respective zero lines. Note where this line crosses the 100 viscosity horizontal. The answer is 36 per cent of 50 viscosity oil (Oil B) and 64 per cent of 200 viscosity oil (Oil A).

Example 2. What is the viscosity of a blend of 20 per cent of a 40 SSU at 100°F oil (Oil B), and 80 per cent of a 150 SSU at 100°F oil (Oil A)?

Draw a line connecting 40 viscosity on the Oil B line, and 150 viscosity on the Oil A line. This line crosses the 80 per cent line (Oil A) at 94, indicating 94 SSU at 100°F, to be the viscosity of the desired blend.

FIG. 18-2*a*.

It is comparatively easy to make the flash test with the proper kind of apparatus. However, it may be materially influenced by such factors as the shape and size of the receptacle used, the rate of heating, the presence of air currents, the atmospheric pressure, the size of the flame used to make the test, and the distance this flame is held above the surface of the oil in the test cup.

The flash point has no relation to the spontaneous-ignition point; this will be very much higher than the flash point.

Fire Point. The fire point is also a temperature measurement. This characteristic is of interest primarily because of its connection with the flash point.

The fire point is determined by increasing the temperature of the oil in the flash-testing apparatus until the resultant vapors ignite and burn continuously when the test flame is introduced. The temperature of the oil at which this takes place is regarded as the "fire point" of the lubricant under test.

Pour Test. The pour test is a measure of the lowest temperature at which an oil will pour or flow under the conditions of the ASTM procedure. This test is of interest to plant operators who must handle oils under relatively cold conditions, especially where intermittent running may be necessary. In an engine operating at low atmospheric temperatures and equipped with mechanical force-feed lubricators, the pour test of the oil used would be of much importance. Any tendency towards congealment in the lubricator or in the crankcase of an enclosed engine (especially when shut down) might cause the pumps to become inoperative, with the result that lack of lubrication, scored cylinders, and burned-out bearings might readily occur. It is perfectly practicable to obtain lubricants of the highest quality which will show a pour test of 0°F or slightly above, according to their viscosity. Naphthenic-base oils in general show a lower pour test over the same viscosity range than paraffin-base products unless these latter have been subjected to special dewaxing treatment.

Carbon Residue. The development of an excess of carbon residue will be detrimental to the attainment of maximum efficiency in internal-combustion engines owing to the tendency which this material has to deposit on piston heads, in combustion chambers, on valve seats, and around and under the rings. Free carbon is chiefly the result of breaking down or decomposition of lubricating oil exposed to the combustion process.

All engine deposits are not always free carbon, however, although this latter element may be most obvious when such accumulations have to be cleaned out. Dirty air is a prevalent cause of carbonaceous deposits, for finely divided particles of dust will combine with carbon residues, particularly if they are of a gummy nature.

Means should be provided, therefore, to reduce the extent of such deposits before they have a chance to form. As far as the lubricant is concerned, this is best done by selecting oils that show low carbon-residue tests and then using such oils (in engine cylinders especially) in as nearly the right amounts as possible to reduce the volume subjected to the possible decomposing effects of the operating temperatures. From a mechanical viewpoint the best preventive (in engine and compressor service) is to use an air cleaner of proved ability and thereby remove the dust particles before they have a chance to combine with any free carbon to cause gummy residual matter.

Emulsification. Emulsification is used in connection with the tendency of an oil to go into such intimate mixture with water as to form a definite emulsion. It applies especially to liquids that are insoluble or only slightly soluble in each other, the mixture being so stable and the liquids so mutually held in suspension that natural separation takes place at a very slow rate. This rate of separation can be retarded by addition of certain fixed or fatty oils to straight mineral oils, as is customary with lubricants for steam cylinders where saturated steam is used. In this way emulsification is accelerated and the resultant emulsion rendered more permanent.

Emulsification is of advantage wherever a lubricating oil must be used in the presence of moisture, especially if sludge formation need not be seriously regarded. Otherwise emulsification must be guarded against, for it is normally a prelude to oxidation and permanent sludge formation. Sludge must always be seriously considered in connection with oil circulating systems or wherever an accumulation of nonlubricating gummy matter may interfere with complete circulation of the oil.

Demulsibility. Whereas emulsification is a measure of the tendency of an oil to go into more or less permanent mixture with water, demulsibility is a measure of the readiness with which separation will take place. Demulsibility is important in any turbine or diesel operation where the oil must be continuously circulated through comparatively small oil piping and through low bearing clearances. Of equal importance, however, is the matter of preventing entry of water through leaky cooling-water connections or abnormal condensation and providing means for water separation and removal from the lubricating system. The conventional oil purifier or filter will normally perform in a satisfactory manner.

The demulsibility of an oil will depend upon the degree of refinement of the oil. The more highly or accurately refined an oil, the better will it separate from water.

This holds true primarily for the lower viscosity range of lubricants, for in oils of the same degree of refinement, as the body is increased, the readiness with which an oil will separate from water will be proportionately reduced.

Sulfur. Sulfur is objectionable in a diesel-engine lubricating oil or fuel only when it is in such form as to be corrosive. In a diesel fuel, sulfur content up to 2.0 per cent is allowable in large slow-speed engines. Sulfur becomes most serious after combustion when there is possibility of contact with water, thus causing sulfurous and sulfuric acids which are corrosive.

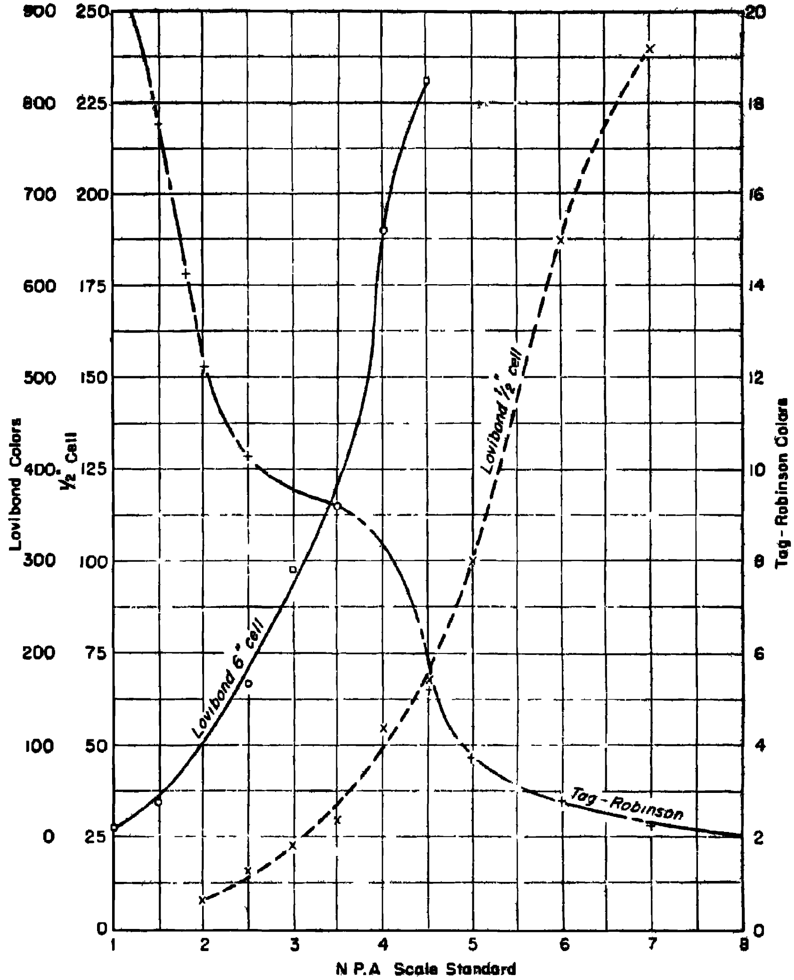

Fig. 18-3. Color conversion chart.

In determining the sulfur content of petroleum products, oxidation and analysis of the oxidized materials are customary. Sulfur in diesel fuels is often regarded as more serious than it really is. Normally, sulfur compounds in petroleum fuel oils have a high Btu value and are completely burned. Sulfur dioxide, as the product of such combustion, is noncorrosive unless there is possibility of contact with condensed water.

Saponification Number. Saponification is a chemical reaction involving the action of an alkali upon a fat or fatty acid, the resultant combination being termed a soap. Saponification will not occur in a well-refined petroleum oil except where fatty acids may result owing to oxidation. However, the principle of saponification is the basis for the manufacture of greases by which a fatty oil or acid is treated with caustic soda or potash or other alkaline metal compounds.

The tendency an oil may have to saponify is determined by measuring the equivalent amount of caustic potash necessary to react with or saponify 1 gram of the oil under test. In terms of milligrams of caustic, the resultant figure is called the saponification number of the oil.

Any lubricating oil for internal-combustion-engine cylinder or crankcase lubrication should show as low a saponification number as possible. When dealing with a compounded oil, however, the presence of a small amount of fatty oil may affect the saponification number proportionally. Some authorities also hold that this increase in the tendency towards saponification will have a like effect upon the tendency towards oxidation and gum formation. They urge that the rate of increase in the saponification number of the used oil be closely watched.

Acidity and Neutralization Number. Organic acids may be present in fatty oils as well as in petroleum or straight mineral oils. The nature of these acids will differ, however, according to their carbon content. In petroleum oils such acids are normally weak and noncorrosive. They should be minimum in a well-refined new oil, but even the best of such oils will show a slight increase in acidity with usage under oxidizing conditions involving heat, moisture, and air. Their noncorrosive nature eliminates one cause for worry. However with increase in acidity, the tendency towards sludge formation will be increased.

Acidity is measured by neutralization; there is, therefore, a direct relationship between the two terms. The "neutralization number," as an index of the acidity, is taken as the weight in milligrams of potassium hydroxide which is necessary to neutralize 1 gram of the oil under test. Since mineral acidity may be more serious from a corrosive angle than organic acidity, the former must be nil. Fortunately, this can also be predetermined by neutralization. Usually the determination of mineral acidity is made only on used oils to determine freedom from or extent of contamination. Unused oils should always be free from any mineral acidity and are so refined.

The Value of Color. Color is simply a visual measure of refinement; it is a function of the refining process; it does not assure the adaptability of any finished petroleum product for a specific use. It can be determined in a number of ways (see Fig. 18-3 for conversion).

The color of refined petroleum lubricating oils may vary all the way from a virtual water white to a deep red or green. Spindle oils are usually of a pale straw hue, although this is by no means a measure of the lubricating qualities. The lighter the color, the better the possibility of absence of undesirable foreign material. However, the color has no bearing upon the ultimate usage.

The color of a petroleum distillate is also to some extent indicative of the viscosity; lighter bodied oils are more nearly of the yellow-straw tinge; the heavier the body, the darker the oil for the same degree of refinement.

GREASES

Base. The term base is indicative of the type of soap used in the manufacture of a grease. In plant service the chief concern is with products of soda or lime base or the mixed-base products designed primarily for ball or roller bearings. There are also specialty greases containing other metallic soaps such as aluminum, lithium, barium, or lead, which are intended for more specific usage.

The Dropping Point. Heat causes a grease to soften. The temperature at which it passes from the semisolid to a so-called liquid state is called the dropping point. It can be defined as the temperature at which a drop of grease disengages itself from the remainder.

In the ASTM test (method D-566-42), the temperature of a grease sample contained in a cup is raised at a standard rate. When the grease has become sufficiently fluid so that one drop falls through a hole in the bottom of the cup, the temperature is noted, and this temperature is termed the dropping point of the grease. Since semifluid greases extrude from the cup at room temperature, this test is not applicable to them, nor is there a standard method for determining their dropping point. Dropping point is controlled by the percentage of soap present, the nature of the fatty oil, and the type of alkali. This latter will usually have the most pronounced effect, greases of soda base or soda-lime base having a higher dropping point than those of straight lime base.

The dropping point of a grease does not indicate the maximum temperature at which it can be used. Ordinary cup greases, for example, in which the oil and soap are held together or stabilized by means of water, have dropping points around 200 to 225°F, yet very often they will not stand up at temperatures continuously above about 175°F since continued subjection to heat of even this intensity causes vaporization of the water, and eventually the grease separates, and soap and oil result. Yet, there are some greases that can be used for short periods of time even above their dropping points provided the bearing housing is sufficiently tight to prevent leakage of the fluid grease.

The dropping point of a grease should be used primarily as an indication of uniformity of manufacture and as a means of identifying greases. The best way to determine high-temperature performance of a grease is to test it in the laboratory under simulated field conditions or by actual service in the units it is designed to lubricate.

Penetration. Penetration is a measure of the consistency and texture of a grease. Numerical measurement of this property is made by allowing a cone-shaped weight of standardized size to drop into a sample of the product at a certain definite temperature. The distance to which this cone penetrates in hundredths of a centimeter is termed the penetration of the grease under test. Obviously, the harder the grease, the lower will be the penetration or the harder the consistency.

PLAIN BEARING DESIGN AND CONSTRUCTION

Alignment. Alignment is a most important factor in the maintenance of efficient plain bearing operation. Any misalignment will immediately lead to abnormal wear, varying in intensity according to what might be termed the angle of error. Under such a condition the shaft will not bear uniformly over the bearing surface but at the ends with abnormal pressure. This may readily lead to excessive wear at these points, overheating, and even wiping of the bearing metal.

The longer the bearing with respect to the diameter, the greater may this wear become under even very slight conditions of misalignment. In view of this, bearing designers endeavor to reduce the length wherever possible and approach a length-to-diameter (LD) ratio of around 2:1, or even 1:1 in some cases. This becomes especially beneficial on power-transmission equipment.

Oil Grooving. When soft metals predominated in bearing design, oil grooving was a generally accepted procedure. This still holds true, for proper grooving is a definite assistant to lubrication of the tin- or lead-base bearing metals.

With harder bearing alloys, mirror finishes, increased machine speeds, and perfection of flood and pressure oiling systems, oil grooving is regarded as less important. In fact, in automotive- and aviation-engine design, rod and pin bearings may carry no grooving at all on the theory that development of the oil film under constant pressure will maintain this film in conjunction with the natural pumping action which is developed at high speeds. Furthermore, elimination of grooving increases the effective bearing area, permitting reduction of over-all bearing dimensions.

Oil grooves, where necessary, are cut in the low-pressure surfaces of the bearing to aid in developing complete circulation of oil and maintenance of a sufficiently protective film at the high-pressure area. They constitute a series of depressions in the bearing surface, connected or so located with respect to each other and the point of maximum bearing pressure as to ensure not only complete and ready distribution of lubricant throughout the bearing clearance but also a pathway for collection and subsequent removal of any nonlubricating material which might otherwise interfere with free flow of the oil film (see Fig. 18-4).

Not infrequently, too little importance is attached to oil grooving in bearings where it should be applied. Some engineers have even felt that it was necessary only to drill an oil hole or two and that the film would spread by some means. This is literally true in the precision bearing designed for circulated flood lubrication, but it is not to be depended upon in rough-service bearings where an oil cup or compression grease cup may be the means of lubrication. The rotating shaft always develops some pumping action to draw lubricant into the clearance, but unless wedge formation is facilitated by chamfering at the point of oil inlet, this action may be very slight. Then boundary lubrication is approached. Hot-running bearings with increase in power consumption and ultimate wiping of the bearing metal are indications of such a condition.

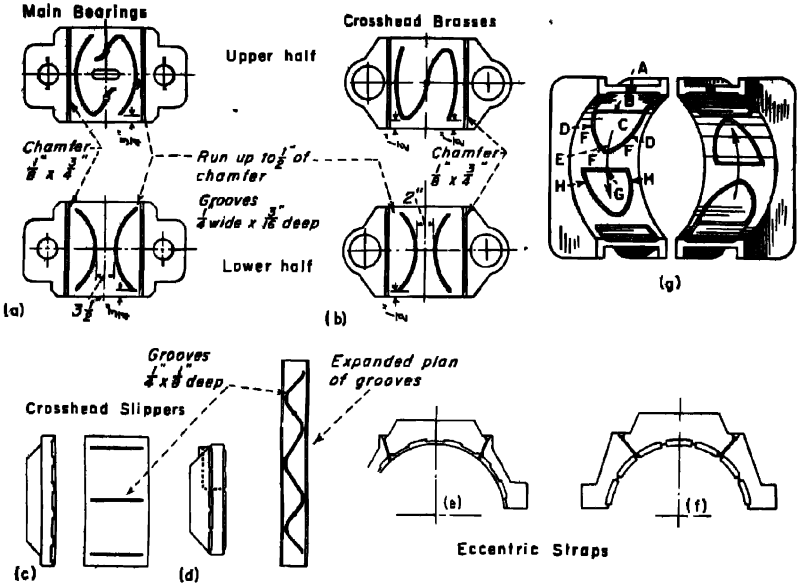
Main Bearings
Upper half
Crosshead Brasses
Chamfer $\frac{1}{8}" \times \frac{3}{4}"$
Run up to $\frac{1}{2}"$ of chamfer
Chamfer $\frac{1}{8}" \times \frac{3}{4}"$
2"
Grooves $\frac{1}{4}"$ wide x $\frac{3}{16}"$ deep
Lower half
(a)
$3\frac{1}{2}"$
(b)
A
B
C
D
D
E
F
F
F
G
H
H
(g)
Grooves $\frac{1}{4}" \times \frac{1}{8}"$ deep
Crosshead Slippers
Expanded plan of grooves
(e)
(f)
Eccentric Straps
(c)
(d)

Fig. 18-4. Typical bearing grooving for marine engines.

Bearing oil grooves should not just be cut to a pattern to suit the individual. Some definite designs have been applied so successfully in practice as to render them virtually standard. Then they can best serve as a means of distributing lubricant lengthwise along the bearing so that the clearance is uniformly filled. The lubricating film is maintained by rotation or sliding of the moving element developing the necessary pumping action. The groove is merely an inlet receptacle for receiving and distributing the lubricant. Where properly designed, the groove can also aid in keeping lubricant within the clearance, preventing excessive leakage at the ends of the bearing. Such a design facilitates circulation of the oil alternately from the center to the outer ends of the bearing and back again. How intricate to make the grooving design would depend upon the means of lubrication. With periodic lubrication, involving sight or drip-feed oil cups, oil grooving and oil retention are more important than where flood lubrication and oil circulation are available.

Planning the Grooves. Oil grooving is most effectual where it is of comparatively simple design and devoid of sharp curves. The more usual designs, as shown by Fig. 18-4, have been developed through practical experience and probably many thousands of hours in engine, compressor, or shaft service. Any or all might function effectively under similar operating conditions provided the lubricating system is dependable. The all-important factor is to develop lubrication to such a degree that friction and power consumption will be reduced to the minimum. The means employed is more or less secondary, although the effective bearing area must not be reduced by too extensive grooving. Grooving should be applied only to those bearing areas which are in the low-pressure zone. In a two-part bearing this will mean that only the top should be grooved. In a four-part bearing the segments to consider would depend upon the direction of rotation of the journal or shaft. Normally the bottom part should be plain or ungrooved.

The purpose is to be in accord with the theory of lubrication which dictates that oil must flow from a low-pressure area to a zone of higher pressure. Grooving of the high-pressure zone might result in the grooves becoming runways to lead oil away from the clearance. Ultimately starved lubrication and actual metallic contact might occur, especially under high load conditions. Oil leakage at bearing ends is often an indication of faulty grooving or grooving extending into the high-pressure area, especially if bearing temperatures increase at the same time. No bearing should be cut away or have the continuity of its surface altered by grooves within 30 deg either way of the point of maximum pressure; otherwise uniformity in the oil film may be impaired.

Cutting or Formation. Having decided upon the bearing area which should be grooved and the types of grooves to use, it is then advisable to make a rough calculation of the dimensions. These will depend upon the diameter of the shaft or journal which is to be carried by the bearing.

The width of a groove is calculated by multiplying the diameter in inches by 0.01 and adding 0.10 in. as a factor of safety.

The depth of the groove is taken as one-half the calculated width.

An oil groove should never be extended too close to the end of a bearing surface, otherwise a channel might be provided which would allow the oil to escape from the bearing clearance to be prematurely lost by drip or leakage. The approximate distance a groove should be from the bearing edge varies according to the diameter, as shown in Table 18-2.

Bearing-metal Thickness. It is obvious that the bearing metal should always be thick enough to allow for the calculated depth of groove without danger of cutting through to the bearing shell. Therefore, it is advisable to note the approximate thickness of bearing metal available before the grooves are cut or the edges chamfered.

It must be realized, when planning a system of oil grooving and calculating the dimensions, that the available bearing area is reduced by these grooves. Under normal conditions, where they are cut in the top half or low-pressure area, actual metal-to-metal contact may not be as likely as in the high-pressure area where the load is carried. But as some pressure is carried over the entire bearing, especially under higher speed conditions, even the low-pressure area should not be reduced any more than necessary by excessive grooving.

Table 18-2.

Bearing diam, in.	*Distance of oil groove from either end, in.*
1¾ or less	¼ min
1¾–2	⅜
2–6	½
6 and larger	1

The actual cutting of an oil groove must be done with the utmost care in order that the finished groove may conform as closely as possible with the calculated dimensions and the intended location with respect to the curvature and length of the bearing. Uneven cutting or irregular edges may lead to clogging if nonlubricating material gains entry. More or less nonlubricating solid foreign matter such as dirt or metallic particles will be present in almost any oil after service for any length of time. Furthermore, if higher operating temperatures prevail, certain types of lubricants may have a tendency to form gummy or carbonlike deposits of oxidized oil or separated soap from grease. As material of this nature is also more or less inert, it will reduce the circulating ability of an oil and accumulate at high spots in the oil grooves to serve as a dam and further impede free and complete circulation of the lubricant. Once begun, accumulation of nonlubricating deposits is often cumulative to such an extent that ultimately bearing failure will result through starved lubrication at the pressure area.

Oil grooves can be cut by hand or by machine. With hand cutting there may be possibility of some variation from the calculated dimensions, although the fineness of the work will depend upon the skill of the mechanic and the care in handling the cutting tools. Rough surfaces must be eliminated regardless of the method of cutting, for as already indicated, rough spots will not only impede free oil circulation, but they may also act as catchalls for nonlubricating foreign matter.

Roughness in an oil groove may be indicated by loose particles of bearing metal. If adjacent to the main surface of the bearing, the occurrence of metal-to-metal contact might lead to breaking or chipping off of certain of these metallic particles, causing contamination of the lubricant and possible scoring of the shaft or bearing surface, especially where bearing alloys containing a considerable amount of harder metals are involved.

CIRCULATING OILING SYSTEMS

Circulation of fluid oils under uniform pressure conditions is helpful in reducing operating temperatures through the cooling effect of an excess of fresh cool oil. Heat is one of the contributing causes of oil oxidation, especially if the oil is exposed to air and metallic particles. Oxidation is evidenced by sludge accumulations and increase in the neutralization number of the oil by the formation of organic acids.

Under comparatively low speeds, the peripheral speed, as it affects the ability of the oil to free itself from entrained water, is important. This becomes more apparent with heavier oils as these (after usage) do not free themselves from water so readily as do light oils. Viscosity is also a factor as it may be related to the pressure developed by the circulating system, the pressure exerted upon the bearings, and the facilities for reconditioning. One of the premises on which the theory of lubrication is based is that oil viscosity can be reduced as the speed of the rotating parts is increased. Therefore, from a speed viewpoint only, it would be right to presume that lighter bodied lubricating oils should be used in high-speed circulating systems. Certainly they would serve as more effective bearing coolants in removing surface heat by conduction. But the matter of bearing pressure also must be considered on many modern machines. This has increased with increase in speed, and heavier loads need greater body in the lubricating film. This requires prior consideration of load-carrying capacity over heat-transfer ability, for the latter can be controlled by increasing the volume of oil circulated through the system.

As a result, a variety of combinations has been developed ranging in simplicity from the unit type of gravity-feed system to those multiple-power devices whereby an entire series of sleeve-type or roller bearings can be served by one oil under the same constant pressure.

The latter has been accepted as a most efficient method of bearing and gear lubrication with the perfection of the sleeve-type roll-neck bearing, the heavy-duty roller bearing, and the oiltight gear housing. The principle is to circulate cool, clean oil under uniform pressures from a central reservoir to bearings and drive gears to provide proper lubrication and to remove heat from the friction surfaces. This keeps the operating temperatures within reasonable limits.

Pressure Oiling by Gravity. In the operation of a gravity system, the supply pump delivers oil from the settling or sump tank to an overhead supply tank which is mounted high enough to give ample pressure to the gears and bearings. An oil overflow line from the overhead tank to the sump tank returns all surplus oil delivered by the pump. From this overhead tank a supply line, through an assembly of lines, delivers oil by gravity to the various points of lubrication. The oil then returns by gravity to the sump tank located below the operating floor.

Gravity circulation reduces the intricacy in construction and the necessity for costly automatic controls, but it is limited in regard to available pressure owing to plant conditions. Where this pressure is low, it may influence the choice of the viscosity of the oil.

Full-pressure Lubrication. With full-pressure lubrication, a supply pump delivers oil at a predetermined pressure from the settling tank to a pressure tank in which an air cushion is provided at the top. Usually two pumps are provided, one of which may serve as a spare. This latter cuts in automatically by means of electrically operated pressure switches when the pressure in the pressure tank drops below a predetermined minimum owing to failure of the operating pump or whenever the latter is unable to maintain the desired pressure. If both pumps fail or are unable to maintain sufficient pressure, a third pressure switch, which is set just below the minimum operating pressure required, sounds an alarm.

The expansive action of the air cushion in the pressure tank forces the oil through the main supply line and the numerous branch lines. These are equipped with orifice plates or valves to provide each bearing or gear unit with its proper share of lubricant. Frequently each of the various points of lubrication is equipped with a pressure switch and light signal to warn the operator should the oil flow be interrupted. The return oil from the gears or bearings flows back to the settling tank by gravity; here it is subjected to filtration and purification to fit it for recirculation.

Pressure Control. Oil pressure in a pressure lubricating system must be carefully controlled within the limits established by the machine designer. Usually from 5 to 15 lb will prevail. Excessive increase in the pressure under which oil is delivered to bearings may cause seal leakage. Then loss of oil and possible entry of water would more than offset the cooling benefits of the increased volume of oil delivered. Most effectual lubrication in a circulating system results when the oil is kept as free as possible from water.

Essential Equipment.* *Oil Cooler.* An oil cooler of ample size is provided to lower the temperature of the oil leaving the pressure tank. For best results an ample supply of cooling water must be provided for the cooler, and the latter must be kept clean. Thermometers inserted in the oil lines entering and leaving the cooler serve as indicators of the efficiency of the cooler.

Oil Filters. Pressure filters installed between the pumps and the pressure tank remove any foreign matter that might remain in the oil drawn from the settling tank. Filters of the latest design have a self-cleaning feature which requires very infrequent dismantling for the removal of the collected foreign matter from the filtering medium.

Centrifugal Purifiers. Many circulating systems will also include one or more centrifuges which are usually operated intermittently for an hour or so each day. Where large quantities of water may get into the system, however, the centrifuges are sometimes run continuously until the source of the water is located and the leak corrected.

Settling Tanks. With the possible exception of the circulating pumps, the settling tanks are the most useful and necessary features of the circulating system. Where there is any chance of considerable water getting into the system, it is advisable to install two settling tanks so that, when the oil in one tank shows signs of emulsification, the contents can be taken out of service for reconditioning. This is done by heating the oil, by means of steam coils located in the bottom of the tank, to a temperature sufficient to reduce the viscosity and enable the entrained water to settle to the bottom whence it can be drawn off through the drain valves.

A well-designed settling tank will have several large cleanout valves located along the front and as near the bottom as possible. It is also desirable to slope the bottom of the tank toward the front. Then by periodically opening the drain valves, the accumulation of water and sludge in the tank can be held to a minimum. Floating suctions are also often installed to allow the pumps to take the cleanest oil which is found near the surface of the oil in the tank.

Any settling tank should be of ample size to allow the oil to flow through at minimum velocity. Thus the oil entering at one end has an opportunity to drop out a considerable amount of its entrained moisture before it again reaches the pump suction at the other end. The tank should be kept as nearly full of oil as possible (due consideration being given to the expansion of the latter when heated for reconditioning purposes) because a full tank reduces the exposed surface on the inside of the tank and prevents excessive sweating. It also aids in the cooling of the oil and allows more time for the oil to drop out its moisture.

It is good practice to alternate the service of such tanks, thus allowing a batch of oil to be in service 1 week and then rest for a week. This is a safety factor against formation of emulsions.

Protecting the Oil in Service. Lubricating oils for circulating systems must be most carefully protected to assure continued lubricating ability. Sludge is an ever-present possibility if bad water conditions prevail. This is why bearing and machine manufacturers include such effective methods of sealing against entry of water and solid foreign matter, which also is a deterrent to good lubrication.

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* For details of construction see manufacturers' catalogues.

Regardless of these precautions, however, water will often gain entry. The only recourse, therefore, is to remove it before it can lead to the formation of emulsions and certain types of sludge. Here the reconditioning system and oil filter play a most important part. In a relatively water-free system, however, sludge can be present in the absence of an emulsion, especially in an oil readily susceptible to oxidation. Furthermore, sludges can be either water-soluble or oil-soluble.

Emulsions are largely prevented by continuous removal of water. Along with sludges they can be eliminated by centrifugal purification. Filtration, in turn, will remove solid contaminants and oxidizable material. In any filtering element the uniformity of the filtering mediums must be maintained. Variation in pressure drop across the filter mediums due to plugging is not desirable.

Water traps and settling tanks in the circulating system have been found to aid materially in removing a considerable amount of water before it has time to affect the oil. All points where water may gain entry into such a lubricating system should be carefully sealed as added insurance. This can be accomplished sometimes by caulking surface joints with oakum and covering with molten lead.

With bearings, however, attention should be directed to their original design in order to provide for some recognized method of sealing which will not impair the free rotation of the shafts. The roller-bearing designer has achieved outstanding success in working with industry. Some of the newer machines are so completely equipped with sealed roller bearings and oil filtration that the color of the oil in circulation is scarcely changed after months of operation.

Methods of Sealing. Modern sealing methods have also been carefully studied to determine their adaptability to the oil-lubricated sleeve-type bearing. The perfected design of this bearing will rarely permit the entry of an excess of actually solid foreign matter. Water leakage, however, may occur, or there may be some accumulation due to condensation. Seal investigation has, therefore, been directed towards reducing this water content. Brass rings, the expanded leather cup, and the compression type of rubber seal have all proved their adaptability and are widely used today for the protection of such bearings.

BALL- AND ROLLER-BEARING LUBRICATION

Ball and roller bearings, commonly called antifriction bearings, are used on motors, line shafting, and other industrial machinery where protected lubrication is necessary. They can be oiled, prepacked with grease, or provided with fittings for pressure grease lubrication.

Particular makes of ball bearings will differ according to the design of the housing and type of lubricant seal used.

In the roller bearing the design of the rollers will be an added characteristic. Rollers generally are either solid or flexible. The latter are always cylindrical in shape, the distance between the inner and outer raceways being the same over the entire length. Solid rollers, however, may be either cylindrical, tapered, or barrel-shaped.

In any case the rolling elements of antifriction bearings will be carried or housed in much the same manner, *i.e.*, in suitable containers comprising raceways and cages. Between these latter are located the balls or rollers. They are kept in their proper position with respect to the races and to each other by the separator, cage, or retainer. In a typical design the inner race will fit on the shaft or journal, the outer being held by the frame or other rigid part of the bearing housing. Rotation of the shaft sets up rotary motion between the rolling elements and the inner and outer surfaces of the raceways.

Ball and roller bearings involve rolling contact. Rolling contact is assured when the balls or rollers are held as nearly as possible in perfect alignment with the shaft. Then deflections under load are reduced. Some designers feel that it is impossible to eliminate entirely such deflections; they believe some sliding motion will always occur. Much of the friction developed is between the rolling members and the retainer or cage.

This is not serious since other conditions of operation may vary widely, as, for example, the speed of operation, the load, or the temperature. Any one of these in its normal variation between minimum and maximum production periods or summer and winter ranges may have considerable effect upon the performance of a lubricant within such a bearing.

For example, the operating speeds must be carefully considered. They influence the choice of oil or grease. The higher the speed, the lower should be the viscosity or body of the oil; 80 to 100 sec Saybolt Universal at the operating temperature of the bearing is the usual minimum limit.

The reason for this is that, as speed of operation is increased, the more rapidly will the lubricant in such a bearing be churned about, especially if the bearing has been overfilled.

Churning causes increase in internal or fluid friction within the lubricant and leads to increase in temperature. This is why oils of low to medium viscosity are preferred for the lubrication of high-speed antifriction bearings or as a component of greases designed for similar service.

Rolling motion in a ball or roller bearing must be maintained as perfectly as possible. If any one of the balls or rollers stops rolling, it will slide. This will be bad for their contact surfaces as well as for the raceways. This can be prevented by designing the bearing seal so that it will work with the lubricant to keep out abrasive dust, lint, water, steam, or acids.

Function of Lubrication. The general purposes of lubrication in such a bearing are

1. To prevent rusting
2. To facilitate as easy rolling as possible
3. To reduce friction where balls or rollers contact the cages or retainers
4. To serve as a seal to prevent entry of foreign matter

Easy rolling is attained by lubricating the balls or rollers and the retainer. To accomplish this, however, all the surfaces (which are of a highly polished nature) must be in as perfect condition as practicable.

Generally, as light a lubricant should be used as can be successfully retained in such a bearing, considering the operating temperatures, speeds, and pressures.

Types of Lubricants. When and where to use oil or grease for the lubrication of a ball or roller bearing will depend upon

1. The operating temperature range
2. The method of sealing
3. The range of operating speeds
4. The load conditions
5. The means provided for relubrication
6. The service conditions
7. The bearing dimensions
8. The possibility of contamination with liquids, solids, or vapors

These conditions naturally will vary widely, and their combination may be quite diverse. Some specific one, however, will usually stand out and suggest the type of lubricant that will give best protection of the bearing.

Oil Lubrication. Petroleum lubricating oils suitable for ball or roller bearings will vary over a wide range of viscosity—from virtual water fluidity to products as thick as syrup at room temperatures. One universal requirement always prevails: they must resist oxidation. Provided a petroleum oil meets this requirement, it can be relied upon to function dependably for long periods of hard service when kept free from contamination.

The choice of the most suitable viscosity will depend upon the bearing seals, the prevailing speed conditions, and the maximum operating temperatures. Generally, high speeds and average plant temperatures will require very fluid oils to reduce the internal or fluid friction as much as possible. In high-temperature service where speeds normally are lower, the oil viscosity must be increased. Specially refined cylinder oils are often used in such service.

Desirable Properties of Grease. Grease for ball and roller bearings should be as free as possible from acid-forming tendencies in order to ensure best protection of the highly polished metallic surfaces. The presence of any material that might lead to oxidation and corrosive reaction or to decomposition or settling is also objectionable.

A properly designed ball or roller bearing will require relubrication only about every 3 or 4 months. Often this period can be extended if the housing holds sufficient lubricant and an effective seal is maintained. *But never pack the bearing too full.*

What Is Grease? Lubricating greases are mixtures of petroleum oil and some kind of soap.

The petroleum oils used in grease making may vary from light pale oils of approximately 100 SSU viscosity at 100°F to heavy cylinder stocks. In some special cases even extreme-pressure characteristics may be required. The oils used should be of the proper viscosity for the service intended and should be well refined so as to resist oxidation and gumming.

The soap components should be just as carefully selected as the petroleum oil and should be made from nonoxidizing and noncorrosive fats which also are free from any foreign matter which might reduce the lubricating qualities. Antioxidation additives are today included to increase the stability of ball- or roller-bearing greases. The finished greases should not contain any fillers or abrasives which would affect the highly polished metallic surfaces of the bearings.

Types of Greases. Greases are divided into different classes according to their base or type of soap. These include calcium, sodium, mixed-base, lithium, aluminum, potassium, and lead-soap greases. Calcium–sodium (known as mixed base) greases predominate in the lubrication of antifriction bearings for industrial machinery.

Calcium- or lime-soap greases are made with a variety of mineral oils. They should be used only for average temperatures, up to, say, a maximum of 150°F; above this temperature the water used in their make-up evaporates gradually, then the grease may decompose, causing the oil and soap to separate. Recently, however, heat-stable lime-soap greases have appeared on the market. Such products do not separate when used at temperatures up to their dropping points; consequently their usefulness is greatly extended. They do not require water for stabilization.

Sodium-soap greases are even more resistant to heat and so can be used under higher temperature conditions.

A mixed-base grease contains a combination of calcium and sodium soap; such greases usually have the good features of both except that they do not have the fibrous characteristics of most sodium-soap greases. They are more like butter and require but little power on starting and running.

When carefully selected materials are used, mixed-base greases resist deteriorations and can be used to pack bearings that are to be stored for a long time or in service where the lubricant is used for long periods without relubrication. These greases can be used at moderately high temperatures, up to, say, 200 to 225°F, for their stability is uniform over a wide temperature range. When a grease is to be used continually at temperatures above 200°F, a frequent cleaning and relubrication routine should be followed.

Ball-bearing Requirements. The amount of lubricant used is very important as it affects internal friction and power consumption. Leading manufacturers advise that where oil is employed *the housing should generally be filled to a level just sufficient to submerge half the surface of the lowest ball when the bearing is idle.*

The viscosity or body of the lubricant becomes a decided factor under high speeds owing to excessive churning and development of heat through internal friction. Under extremely high speeds wick-feed or sight-feed drip oil lubrication has satisfactorily reduced churning and heat development.

When grease is used, however, a somewhat greater amount is permissible, the housing being charged from one-quarter to one-half full. Use of greater quantity may lead to drag, temperature rise, and abnormal power consumption.

Grease, owing to its seal-forming properties, is useful for any type of service where there is danger of contamination with dirt and moisture. Grease also can be very much more effectively retained in a nonoiltight housing. Conversely, dirt or grit that finds its way into a grease-lubricated bearing does not settle out but usually becomes well mixed with the grease, forming an abrasive.

Roller-bearing Service. When a roller bearing is to be lubricated with oil, much the same conditions apply as stated in connection with ball bearings. Where pressures or temperatures may be high, better lubrication will often result if somewhat heavier oils are used.

Heavier oils for roller-bearing lubrication should be chosen with the utmost care after careful study of the bearing temperatures, for it is very possible to overestimate the conditions of operation; then excessive drag may develop.

Where such bearings may be exposed to consistently high temperatures, pressure circulation of a comparatively fluid oil has been found to promote lower actual bearing temperatures. Where the oil acts directly as a coolant, it may in turn be cooled before recirculation.

In the grease-lubricated bearing, but little cooling effect is developed by the grease itself. If, however, grease has been prepared to meet the operating conditions, it will promote a uniform range of bearing temperatures by its ability to reduce friction, especially if it does not develop abnormal internal friction within itself.

Protecting Lubrication. A well-designed ball or roller bearing usually contains a good seal to prevent leakage of lubricant. Grease, in itself, will form a good seal if it is heavy enough, but it is best to rely on mechanical seals, for heavy greases may cause high power consumption. For this reason greases which are easily worked and which can be used under a wide temperature range are best suited to motor bearings and other bearings.

It is also important to study sealing devices wherever the bearings must operate under high temperatures. As indicated, the effect of such temperatures is to reduce the viscosity or consistency of a lubricant. The extent to which this will ocur depends upon the original body of the product.

Seal Design. In the earlier developments of bearing seals, felt- and leather-washer and grease-groove seals were widely used. They were comparatively inexpensive and simple in design. These advantages cause them still to be used by many bearing manufacturers for certain types of service.

Since their development, however, other types of seals have come into usage including metallic rings, similar to piston rings, and metallic springs for the purpose of maintaining the adjustment of felt, leather, rawhide, and cork against the shafting. Composition rings have also proved of value as an oil seal or to protect a grease-lubricated bearing against entry of water.

The chief objective in the selection of a suitable seal is to obtain a design that will wear least in service; otherwise the purpose may be defeated. Washer-type felt or leather seals will have this tendency, although the rate of wear will depend upon the quality of the material. Where adjusting springs are used in connection with similar material automatically to keep the seal in close contact with the shaft, this will counteract wear; also it will help the seal to keep in the lubricant, although the pressure developed may sometimes promote shaft wear.

Protecting the Seal. Control of the lubricant in the bearing is most important. For each ounce of fresh grease applied, almost an equal amount of used grease disappears—usually past the bearing seals to accumulate or drip onto some nearby part of the machine.

If the bearing operates at higher temperatures, this may be very serious, especially when the greases used may thin down excessively. The result will be the same as when grease is applied to the bearing under sufficient pressure to force it past the seal.

Cleaning and Flushing. Many ball and roller bearings have a plug at the bottom for draining. When cleaning is necessary, the plug is unscrewed, the bearing flushed, the plug replaced, and the whole relubricated.

In an oil-lubricated bearing it is practicable to provide an arrangement of nipples and pipe fittings terminating in a sight gauge glass. In this way not only is cleaning made easier, but also the oil level in the bearing can be observed.

Maximum protection of any antifriction bearing is assured only when the lubricating system is *free from foreign matter.* In certain types of operation, this can be more easily maintained than in others. The extent to which dust or dirt may be present in the air or thrown onto the bearing will be a factor.

Bearings should be flushed and cleaned at periodic intervals, say about once or twice a year for the average plant motor. A light slushing oil should be used for this purpose. Gasoline or other solvents should not be used, for these would tend to dissolve the mineral-oil content of any grease used, leaving a residue of insoluble soap.

The frequency will depend upon the design of the bearing, the type of seal, the extent to which dust, dirt, or water may be present, and the nature of the lubricant. In some plants it may be necessary to flush certain ball bearings every 3 or 4 months.

Pressure Grease-gun Operation. The grease gun is one of the most valuable tools in plant maintenance. However, if the pressure gun is not carefully handled, too much grease can be forced into a bearing housing. Also, dirt can be forced in if the operator forgets to wipe each fitting before using the gun.

Overlubrication may cause overheating especially if the grease is too heavy. Such overheating is brought about by excessive internal friction within the lubricant itself.

With certain types of grease abnormal increase in temperature may make the grease so fluid or may change its make-up so that oil separates from the soap in the grease. This reduces the lubricating value of the grease and causes loss of oil by leakage if the bearing is not well sealed.

INDUSTRIAL GEAR LUBRICATION

The ideal gear lubricant must possess certain very definite properties such as

1. Sufficient adhesiveness to ensure that it will remain on the teeth and resist the action of centrifugal force
2. Such a viscosity as to form a suitable film regardless of tooth pressures or temperatures
3. No tendency to harden or become brittle and chip off in event of exposure to abnormally low temperatures
4. Sufficient "oiliness" or lubricating ability to reduce to a minimum the friction that occurs between the teeth
5. Sufficiently low cost to enable it to be purchased and applied at a reasonable expenditure in comparison with the value of the service rendered

Spur, Bevel, Helical Gears, Etc. In any gear installation the pressures usually encountered will be high; they may be especially severe on the bearing surfaces inasmuch as a relatively small area of contact is involved. The pressure developed by this contact is constantly shifting as the teeth are subjected to varying directions of load application. If rolling contact alone prevailed, the intensity of such a load momentarily on any part of a gear tooth could be disregarded. But sliding motion as well is practically always present; this explains the abnormal wear that occurs where gears are not properly lubricated or where the tooth surfaces are subjected to a continual abrasive action.

There are numerous gear lubricants on the market which have been designed to meet these requirements. They will vary from fluid mineral oils to intricate compounds, some of which may contain "fillers." Fillers serve but one purpose in a lubricant: to increase the viscosity and weight. They possess no appreciable lubricating value. Therefore fillers should not be used as they are deceiving and may lead an uninformed user to feel that his lubricant is suitable, when in reality, it is affording comparatively poor protection against wear. The necessary adhesive characteristics may also be inadequate. As a result, when used under high-speed conditions or wherever the gears may come into contact with water, acids, alkalies, or chemical fumes, the gear teeth may lose their protective film of lubricant owing either to its being thrown off by centrifugal force or being washed from the wearing surfaces.

Worm Gears. Except where the speeds are very low and the loading relatively light, worm gears are generally enclosed and bath lubricated. Open worm gears must be lubricated according to the speed and size. Normally they require a lubricant that will adhere to the teeth and yet be fairly fluid. Enclosed worm gears, particularly where a hardened-steel worm operates with a bronze gear, are usually lubricated with compounded steam-cylinder oils or noncorrosive lead-soap lubricants. These rather heavy-bodied oils are necessary in worm gearing to resist the squeezing-out pressure exerted by the sliding tooth contact and to provide the protective film between the teeth which is depended upon to minimize the friction of the sliding motion inherent in worm gearing. At the same time they cushion the impacts between the teeth resulting from gear inaccuracies or unevenness of the load being transmitted.

In the average bath-lubricated worm-gear installation, either the worm or the gear is permitted to dip into the lubricant according to the size and arrangement of the gearing, thus providing a sufficient amount of lubricant on the teeth to afford a protective film but not enough to cause "drag."

Extreme-pressure Lubricants in Industry. The advantage of the extreme-pressure (EP) type of oil for industrial gears has been definitely proved. Iron, steel, paper, and rubber are but some of the leading industries that have benefited. In brief, there is a definite field for the EP oil of the right viscosity, as indicated in Table 18-3.

Table 18-3

Service	*Viscosity range, SSU, at 210°F*
For high-speed high-pressure gears	50– 75
For smaller units of medium speed where gears and bearings must be lubricated with the same lubricant	70– 115
For large units of medium speed where gears and bearings must be lubricated with the same lubricant	130– 150
For gears of large pinion stands where a separate system, using lighter oil, is used for the bearings	230– 275
For heavy-duty mill pinions and reduction-gear sets	400–1,000

Conditions of Operation. Gears may be either encased or exposed according to the nature of their service and the extent to which they require guarding or protection against abrasive matter or dust. Exposed gears may often present lubrication difficulties. Normally they will require a lubricant that will adhere tenaciously to the teeth and resist the action of centrifugal force, speed, and temperature changes.

Where tooth pressures are not abnormal and the teeth are not subjected to heavy shocks, the lubricant viscosity should be chosen accordingly. If the lubricant is too heavy, it will defeat certain of the advantages of lubrication by imposing more drag on the gears and increasing their power consumption. As tooth pressures increase, however, the viscosity of the lubricant should be somewhat higher in order to ensure that the film on the teeth will be able to resist the squeezing action of their surfaces, especially when sliding motion takes place. This condition warrants a lubricant of increased film strength.

The presence of considerable dust in the air as a result of materials processed, as in the cement or rubber industry, may cause exposed gear lubricants to become veritable grinding compounds if they are too adhesive to be regularly displaced by fresh lubricant. Here it may be necessary to compromise and use a much lower viscosity oil than would otherwise be desirable.

The heavy pressures that are customary on certain types of steel-mill pinions, rubber-plant mixers, or cement-mill kilns may squeeze out ordinary lubricants, permitting actual tooth contact and thus calling for the use of a mild EP lubricant.

The frequency of relubrication of exposed gears depends entirely upon the service conditions. They must never run dry, neither should they be overlubricated so that a sloppy condition will prevail. Experience will dictate the proper viscosity just as it will the best method of application. A grade suited to clean operation may not maintain an efficient lubricating film under dirty conditions.

Where heavy bearing pressures prevail, these, in turn, may be transmitted to the gear teeth to some extent. Sometimes adjusting screws are employed to regulate the pressure to suit the work. Widely varying duty may, therefore, be demanded of some reduction and driving gears. Where slower speeds are customary, the possibility of overheating or throwing of the lubricant is reduced, yet wear may be appreciable if a proper film does not coat the teeth completely.

Sliding motion is most likely to occur when the teeth first come into contact and as they become disengaged, the points of the respective elements developing a scraping action on the sides of the adjacent teeth. Both wiping action and squeezing may occur at these times. Wiping or scraping off of the lubricant is most apt to occur where the teeth are sliding; squeezing out is possible as maximum pressures are encountered. Wear will occur more rapidly at the points of approach and recess of the teeth owing to the sliding friction involved, whereas the intermediate tooth surfaces adjacent to the pitch line, being subjected to rolling action, will resist wear more effectively.

Methods of Gear Lubrication. Gear lubricants must be applied according to the design of the installation. We are most generally familiar with automotive practice where the gears in the transmission and differential are bath lubricated. Contrast this with industrial production, *e.g.*, the gears that drive the rolls on certain rubber machines which may be either bath or hand lubricated; the gearing on excavating machinery which is largely hand oiled; and turbine reduction gears which are lubricated by force feed. This latter method of lubrication is also now widely applied to enclosed-type steel-mill reduction-gear sets.

So a wide variety of conditions exists, each of which may require special consideration according to the design of the machine, the size and number of the gears, and the products being handled. Certain of the latter will be of a perishable nature such as foodstuffs, paper, and textiles. Gearing on the attendant machinery must be so lubricated that there will be practically no chance of the gear lubricant splashing, dripping, or being thrown onto these products.

Analysis of industrial machinery presents five distinct conditions pertinent to gear lubrication:

1. Where the gears are enclosed in a suitable casing, out of contact with the bearing lubricant, running either in a bath of oil or being lubricated by force feed
2. Where gears run in an enclosed case, the gear lubricant at the same time serving the bearings, as is true in the majority of industrial gear drives
3. Where gears are entirely exposed, the lubricant being applied by hand
4. Where the gears run exposed but with their lower portions encased so that bath lubrication is possible.
5. Where gears are open and exposed to such dirty conditions as to render practically any lubrication ineffectual

Where and when to use any specific grade of lubricant is, therefore, often quite a problem. It is disconcerting to attempt to heed all the modern arguments in favor of industrial-plant efficiency and at the same time have a set of gears that may refuse to function properly with ordinary methods of lubrication. Such conditions are not entirely hopeless even though perhaps the air may be filled with dust or the gears subjected to hot water or acids, etc. The petroleum industry has solved this problem by developing specialty lubricants refined so as to stick and lubricate even when totally submerged in practically boiling water.

There are available similar lubricants for pressure conditions as already mentioned. These along with the wide range of straight mineral gear lubricants will serve virtually any condition of operation or means of application. It requires only judicious study of these conditions along with the physical characteristics of the lubricants and the application of the gearing.

Clean Conditions Promote Good Lubrication. Where pans are furnished for bath lubrication, they should be cleaned out at regular intervals since any dust that may have gained entry may destroy the adhesiveness of the gear lubricant, tending to form a pasty mass which the gears cannot pick up.

Where the gears dip in a bath of lubricant, this bath should be only so deep that the lowest teeth are just covered. This will give ample lubrication and keep the consumption at a minimum.

When the lubricant is applied by hand, it should be heated, if necessary, to reduce it to the consistency desired and painted on with a brush or poured on from a dipper as the gears are slowly revolved towards each other. Applying it in a fine stream or by spraying is most economical and effective.

Gear lubricants should be applied to open gearing relatively frequently and in small quantities rather than in considerable volume and at longer intervals. Thus there will be less wastage and less danger of gobs of lubricant being thrown off by centrifugal force to contaminate the products passing through the machine. In case the lubricant becomes laden with foreign material and packs at the roots of the teeth or between the gear and its shield, this cake often can be softened and removed by the use of kerosene or some petroleum solvent. When there is trouble, see Table 18-4.

Cautions—Gear Lubrication. Check alignment of driving- and driven-shaft connections.

Inspect after first half hour after starting. Evidence of improper selection of lubricant often shows up during this period, or evidence of misalignment may be indicated.

Table 18-4

Condition	Most prevalent on	Cause	To correct it
Abrasive wear or scoring	Spur, helical, bevel gears	Misalignment or rough surfaces; improper tooth contact; sliding under heavy load; too low oil viscosity; low starting temperatures	Increase oil viscosity; use oil with mild noncorrosive EP additive; use means for preheating to raise the starting temperature
Galling	Spiral-bevel, hypoid gears	Oil-film rupture, high surface temperatures	Use oil with mild noncorrosive EP additive; use means for cooling to reduce temperatures
Pitting	Any or all types of gears	Occurs with rolling as well as with combined rolling and sliding with oil too low in viscosity. More prevalent on rough-surface finishes or where local tooth overload prevails	Use an EP oil; increase the oil viscosity; try to get gears of better surface finish; increase the surface hardness or metal toughness. Improve tooth alignment and load uniformity by shimming under bearings or adding outboard bearings to overhung pinions
Burning	Any or all types of gears	Overload or lack of lubrication	Run under conditions for which gear was designed. Lubricate

Watch the oil level. The amount of oil in an enclosed gear unit is often just as critical as the kind of oil.

Check oil level when machine is standing still; if checked while running, a false level may be indicated.

STEAM-CYLINDER LUBRICATION

Steam-cylinder oils must be of comparatively heavy body and possessed of high "wettability" in order to ensure that the lubricating film will cling to the valve seat and cylinder walls, withstand washing off by the steam, and resist the wearing or scraping effects of valves and pistons.

Viscosity or body is attainable by suitable refining.

"Wettability" is obtained by judicious treatment of the cylinder stock and the addition of certain fatty oils to obtain an emulsifying effect.

An oil having a viscosity range of between approximately 100 and 220 SSU at 210°F is generally required according to

1. The steam pressure and temperature involved
2. The type of steam valves
3. The means of application available

Cylinder oil may be classified according to viscosity as follows:

1. Light-bodied—100 to 120 SSU at 210°F
2. Medium-bodied—120 to 150 SSU at 210°F
3. Heavy-bodied—150 SSU (or above) at 210°F

Lubricant Distribution by Steam. The best way to get cylinder oil to all desired points is to use the steam itself as a carrier. The steam reaches all moving elements inside the valve chambers and cylinders with the possible exception of parts of certain types of Corliss valves.

If the oil is divided into minute globules and intimately mixed with the steam, only a very small quantity is required. The extent to which the oil is atomized will control both the efficiency of lubrication and the quantity necessary.

The more complete the atomization and the more thoroughly the oil is dispersed through the steam, the better will be the lubrication of the engine and the more economical the consumption of oil. Atomization is influenced by

1. The condition of the steam
2. The point of introduction of the oil
3. The velocity of the steam
4. The character of the oil

Factors Affecting Atomization. High steam pressure and temperature will atomize a given oil faster than low steam pressure since high temperatures thin down the oil to a greater extent. So a comparatively heavy-bodied oil may be atomized by a high-pressure steam as quickly as a light-bodied oil will be atomized by a low-pressure steam.

Conversely, low-pressure steam requires a lighter bodied oil in order to obtain efficient atomization. The point of introduction of the oil into the steam is also a factor; the farther this point is from the cylinder, the greater will be the opportunity for complete atomization.

Locating the Point of Introduction. If the oil is introduced directly into the valve chest or just above the throttle, a product must be used that will distribute itself very rapidly through the steam. Conversely, if the point of introduction can be located from 4 to 6 ft from the cylinder, the oil does not have to be able to atomize so readily as there is more time for the steam to develop the necessary atomizing effect.

If a heavy-bodied oil is used to meet the cylinder conditions, better lubrication may be secured more economically by locating the atomizer farther back from the cylinder.

There is a limit, however. If the oil is injected into the steam too far from the cylinder, too much condensation of oil on the sides of the steampipe will occur. This will increase the oil consumption, as the oil will run down into the steam chest without atomizing. Ultimately, it may cause detrimental gummy or carbonaceous accumulations.

How to Judge Effective Cylinder Lubrication. The condition of the cylinder, valve seat, and valve surfaces indicates the effect of lubrication. If valves function without undue noise and if there is a light film of oil on the piston rod, the engine should be well lubricated. Some oil in the condensate is also a good indication.

Time is necessary for the lubricating film to form and function properly; a hasty decision as to the suitability of such an oil should never be made. Any test should cover a period of several weeks. Then the engine should be shut down and the cylinder head and valve-chest cover removed to enable inspection of the interior. If there seems to be a film of lubricant sufficient to penetrate and leave the time-honored brownish stain on three or four thicknesses of cigarette paper, the surfaces may be regarded as being properly lubricated. Further evidence is a high polish and a color varying from bright iron white to steel blue.

Indications of Imperfect Lubrication. Rough, dry, dull, or rusty surfaces indicate that lubrication has either been insufficient or the wrong grade of oil has been used. In addition, if the stain on the cigarette papers appears streaked, blackish, or mottled, either the oil has been subject to carbonization or abnormal wear has taken place.

Lack of lubrication will also sometimes be indicated by sticky valves or groaning sounds from the cylinder when the engine is running. With Corliss-valve systems, slowness in action of the dashpots indicates insufficient lubrication.

Exhaust Steam Is a Guide. If the condensed steam shows considerable quantities of liquid oil, either too much oil is being fed, or it is not properly atomized.

If it shows minute drops of oil and is milky in color, it is probable that atomization is complete and the feed is correct.

If there is a film of oil on the piston rod and no oil has been fed directly thereto, it can be taken as an indication that atomization is satisfactory and that the surfaces are receiving sufficient oil.

Pools of oil lying in the bottom of the cylinder or in the counterbore indicate excessive lubrication. This can be determined, however, only at the time of engine overhaul.

A suitable film on the piston rod, minute drops of oil, and a milky appearance in the condensate, free action, and little or no noise in valve operation are indications that the oil is suitable, atomization complete, and the rate of feeding is about right.

Selecting the Oil. Pressure, temperature, and moisture must all be considered in selecting the oil, for each will have a decided influence upon effective lubrication. There is more or less of a relation between these conditions, also the steam velocity and the use which is to be made of the exhaust.

Temperature Depends on Pressure and Superheat. The operating temperatures in the steam cylinder and valve chest are most important. It can be noted from standard steam tables that temperature is directly related to the pressure:

1. As oil viscosity varies inversely with temperature, it will be decreased as steam temperatures are increased.
2. Where operating with high-temperature saturated or superheated steam, a heavier bodied oil should be used.
3. Conversely, where steam pressures are low or where a partial vacuum with accompanying low temperatures exists, a lower viscosity, more easily atomized oil is necessary.

The heat of the steam does not noticeably affect the rate at which a well-refined cylinder oil will be oxidized. For example, consider "the operating temperature" for saturated steam at, say, 600 lb, which is 486.6°F. This is below the open-cup flash point of any cylinder oil. Furthermore, it is possible that the flash point would be raised with increased pressure. As a result, there does not seem to be any reason to believe that a very high flash or fire test is necessary.

Saturated steam always contains moisture, and as there is relatively little oxygen present in the steam, it does not seem likely that ignition could be produced in the cylinder regardless of the flash point of the oil. There are some, however, who advocate using means for deoxidizing the steam and make-up feed water to retard corrosion of boiler tubes and steam lines.

Velocity Improves Atomization. Atomization of a cylinder oil is improved as the velocity and temperature are increased. The higher the temperature and velocity, the more readily will a heavy-bodied oil be atomized owing to the reduction in viscosity which occurs at the temperature of atomization. These conditions must be taken into account when deciding on the initial viscosity of the oil.

Moisture Requires Compounding. To secure proper lubrication under wet-steam conditions, it is necessary either to increase the rate of flow of a straight mineral oil or substitute an oil that contains a certain percentage of fatty compound such as lard oil, degras, or tallow.

Steam will always contain a certain percentage of moisture unless it is superheated to a sufficient extent to counteract any line and cylinder condensation which may be caused by the cooling effect of the piping or cylinder walls and the expenditure of heat by the expansion stroke.

Compounding develops an emulsion through reaction of the compound with the moisture in the steam. The lubricating film thus has a greater affinity for the cylinder walls and other wearing surfaces and becomes more resistant to the washing action of the water in the steam. The greater the percentage of moisture in the steam, the higher should be the fatty compound content of the lubricant, although 10 to 12 per cent is the usual maximum.

An excessive amount of fatty compound beyond that necessary to form the requisite emulsion will not improve the lubricating value of the oil. It may even be objectionable, as exposure to high temperatures can cause some fatty oils to decompose and form carbon and corrosive acids.

It is best to use just enough compound to conform to the moisture conditions of the steam and maintain a suitable film of oil on the cylinder walls. It is best, therefore, to reduce the quantity of compound and improve the quality.

Where Compounding Is a Detriment. The property that causes compounded oils to unite with water to form emulsions in the cylinders also prevents ready separation from water in condensed steam; likewise, the more completely the oil is atomized, the more difficulty it will have in separating from water. Obviously this will cause trouble where the condensate is used for make-up or led to open feed-water heaters.

Oil emulsions in a boiler combine with the boiler compounds to cause foaming or with the boiler impurities to produce a coating over the tubes and fire surfaces. This coating seems to form more readily over clean tubes than over dirty ones. A very thin layer of oily sludge over a tube or shell surface will so insulate it that there is not only a large loss in heat efficiency, but the rise in temperature of the metal may be so excessive as to cause burning out or explosion of the boiler.

Table 18-5. Quantity of Steam-cylinder Lubricant Required per 10-hr Day

Drops per min	Cylinder size, in.	No. of pints in 10 hr
4	8 × 8	0.4
8	12 × 12	0 75
16	18 × 18	1 5
24	24 × 24	2.25
32	30 × 30	3.0
40	36 × 36	3.75
48	42 × 42	4.5

Even if exhaust steam is used only for heating, the heaters may become coated with oil and lose in efficiency. This is also true if exhaust steam is used in tube and shell feed-water heaters. Likewise, in condensing engines an unsuitable oil will form layers on the condenser tubes with consequent loss of vacuum and thermal efficiency.

Where Straight Mineral Oils Should Be Used. Where the presence of a fatty oil in the exhaust steam is objectionable, it may then be advisable to use a straight mineral type of cylinder oil. However, as an increased amount is necessary to ensure proper lubrication, imperfect atomization may result. In consequence, oil accumulations in the cylinder will be prevalent and carbon deposits developed.

This will be especially true in multiple expansion engines equipped with receivers and reheaters, the high temperatures to which the oil is subject being very conducive to carbonization. In poppet-valve engines carbon formation of this nature may often cause imperfect operation of the valves. In any case, some carbon accumulations may form around the rings.

Table 18-6. External Lubrication for Reciprocating Engines

Lubrication system	Speed	Rod diam	Grade of oil	Viscosity of oil at 100°F, SSU
Hand-oiled or drip-feed oil cups.	High	1½ in.	Red engine	100–200
	Medium	1½–3½ in.		200–300
	Low	Over 3½ in.		300–500
Ring or chain oilors.	All	All	Pale	180–300
Splash lubrication.	All	All	Pale filtered	150–300
Circulating systems. . . .	Medium	Small and medium	Pale filtered	150–200
Circulating systems.	Low	Large	Pale filtered	200–300

Note: Reference to speeds in general for reciprocating engines is considered to be in the following range: low speed—below 50 rpm; medium speed—50 to 150 rpm; high speed—150 rpm and up.

AIR-COMPRESSOR LUBRICATION

In compressing air, heat is developed, caused by the rapid increase in the rate of impact between the molecules of the air involved. Actual friction contributes but little to the total amount of heat developed. Normally most of the machine friction will be dissipated by radiation or conduction.

The air compressor, however, is not a heat engine, for the heat developed during compression is largely removed from the compressed air by cooling.

Obviously, the air compressor must be lubricated. Furthermore, this must be done at fairly high temperatures, for the heat of compression is not removed until after discharge. Thus, the piston, cylinder walls, or vanes, according to the type of unit, become heated.

Air-compressor Cylinder Oils. These oils must be capable of withstanding heat with minimum tendency towards oxidation. All lubricating oils will be subject to more or less partial distillation under high temperatures. As this continues the lighter portions evaporate and are carried off with the discharged air, the heavier fractions meanwhile remaining in the cylinder or collecting in the heads or on the valves.

These oily deposits gradually thicken, carbonize, and collect any dust which may pass in with the air, finally developing to such an extent that they may interfere with the operation of the valves. Hence the advisability of using a suitable air filter.

It is usually the dust from the air along with gummy oil deposits (known as carbonaceous matter) from improperly refined oils or excessive use of oil which causes faulty valve action.

Cooling. Means of cooling and the availability of adequate ventilation or cooling water (depending on whether compressors are air- or water-cooled) are most important considerations with respect to lubrication, especially when dealing with portable compressors.

Most effective lubrication is attainable where the walls can be kept sufficiently cool to preclude vaporization of the oil film. Suitable fins around air-cooled cylinders and water-jacketing those which must be water-cooled take care of this.

Intercooling Protects Lubrication. Intercooling, where two or more stages are involved, assists in reducing the temperatures to which the oil is subjected. Along with aftercooling, it also aids in condensing oil vapors and any moisture that may have been taken up by the compressed air. This helps to increase the amount of air compressed per horsepower.

Therefore, wherever practicable, air should be suitably cooled to nearly the inlet temperature of the cooling water or the atmospheric temperature of the air in an air-cooled unit.

In this way oil is protected both in service, as a lubricant, and afterwards against blame as a possible cause of explosion. If the oil should be used to excess and not be entirely removed from the discharge air, the temperature of the latter, after suitable aftercooling, is normally too low to cause combustion or flashing of the oil.

Aftercooling Reduces Danger of Explosion. The aftercooler in turn is effective in correcting explosive conditions. It reduces the temperature of air discharged to the lines and removes excessive oil vapors which may have formed owing to improper lubrication.

Compressor-oil Characteristics. Air-compressor lubrication may present a problem due to the nature of the operating conditions to which the oil will be subjected. There may be a considerable temperature range; furthermore, as this may approach quite a high maximum, there may be considerable possibility of oil oxidation or combustion, with the accumulation of more or less carbon residue according to its base and refinement.

It is important to realize that enclosed mechanisms or interior surfaces are involved which do not enable easy inspection. They may be rotating or reciprocating, according to design, but all involve sliding friction.

Load conditions also may influence lubrication especially where high piston-ring pressure is involved. In addition, the operating conditions may be contradictory, for example, where a large loose-fitting piston might require a heavy oil but where the speed conditions would indicate that a lower viscosity oil might function best.

The function of lubrication is to prevent metal-to-metal contact. Theoretically, the variety of conditions which affects the performance of the oil might indicate that selection of the lubricant would be quite complicated. Actually, however, it simply involves the use of an oil of adequate stability and suitable body or viscosity.

Viscosity. Viscosity or body is related to piston ring or vane seal according to the type of machine; consequently, it affects the efficiency. If the oil is too light in body or too low in viscosity, it may not furnish the required seal at the peak temperatures of compression. This may result in increased oil consumption or air leakage.

It is related to the ultimate development of carbonaceous residues, for an increase in the rate of oil delivery is often the first thought. This only adds to the volume of oil that may be subject to deposit formation.

In turn, if the oil is too heavy or of too high a viscosity, it may be sluggish at intake temperatures, subject to imperfect atomization, apt to have too much internal friction, and likely to hold nonlubricating foreign matter more tenaciously than a lighter oil. It will also not be so readily subject to partial distillation, but the amount of carbon deposit developed will be proportionately higher.

The usual viscosity range for reciprocating-compressor service will be from 200 to 500 SSU at 100°F. For rotary (blade- or vane-type) units, somewhat heavier oils are preferred to maintain the seals, *i.e.*, from 50 to 125 SSU at 210°F, according to the design and operating conditions.

The rotary type of machine will normally require a heavier oil than the reciprocating unit (for the same compression range) owing to the scraping action of the blades against the cylinder wall as they are impelled outward by centrifugal force as the rotor turns.

The oil must not only maintain a suitable film on the cylinder walls to provide a running surface for the blades but it must also lubricate the slots that hold the blades.

Flash Point. The fact that oil vaporization may occur in air-compressor cylinders requires consideration of the flash point or that test which is indicative of the relative initial volatility of the oil.

There should be no flame or spark in an air compressor, so the flash point is not a measure of explosion possibilities.

Oils that show the highest flash points will usually develop the most carbon residue. Furthermore, this may be hard, adhesive, and resistant to removal either with the discharged air or by mechanical means.

To connect the flash point with the possibility of compressor or air-line explosion is unwarranted unless there is some adverse condition of construction or operation. For example, the use of dirty air along with an unsuitable (gum-forming) lubricating oil may have built up restrictions at fittings or bends in the air lines.

Whenever such obstructions have become extreme and become incandescent by increase in air friction, if the oil content in the air is high, failure of materials may occur owing to reduced tensile strength through overheating, irrespective of the vaporization tendencies of the oil. Otherwise, the oil would have to be subjected to a temperature far in excess of the laboratory flash point or even its spontaneous-ignition temperature before actual combustion could occur.

The safest procedure is

1. To use sparingly a suitable compressor oil of the lowest possible carbon-forming tendency
2. To see that the air is cool and clean
3. To check the discharge valves for leakage periodically
4. To be sure the oil has a high spontaneous-ignition temperature

Carbon. Carbon is caused by exposing an oil to direct heat and air. This involves a chemical change. If this heat is sufficiently intense, it causes abnormal vaporization of the more volatile constituents to result in accumulation of nonlubricating (oxidized) matter which may be of tarry or gummy nature. Continued exposure of the latter to heat causes coking or baking in piston-ring grooves, on valves, or elsewhere in the compressor.

The cleaner an oil will vaporize, the lower will be the amount of objectionable residues. Normally carbon residue will vary directly with the viscosity as well as with the flash point. A distillate compressor oil such as is customarily used for normal conditions (within the viscosity ranges mentioned above) should show a carbon-residue content well below 0.50 per cent by the Conradson test.

Any such residual matter would be of a gummy nature and would tend to hold any dirt or dust taken in with the air. Therefore, the latter should also be as clean as possible; a suitable air filter will assure this.

Application of Compressor Oils. How compressor oils are applied is of considerable importance in the ultimate development of effective lubrication. It is as vital as selecting oils of the proper physical characteristics, for if the rate of oil delivery cannot be properly controlled, the possibility of future difficulties due to carbon formation will be increased.

Point of Delivery. Air-compressor oil will function best when completely distributed over the cylinder walls.

On larger units the oil is usually pumped into the cylinders themselves by a mechanical force-feed oiler. On smaller units, splash may be used. Where controlled lubrication is possible, on account of the small amount of oil involved and the fact that a filtered oil is more readily atomized, the lubricator is generally located fairly near the intake or cylinders as atomization is effected in a relatively short distance.

In this manner the possibility of overfeeding oil to any particular part is reduced; oil flow stops and starts with the compressor, and the requisite lubricating and sealing film is more nearly assured.

Force-feed lubricators are not affected by variations in air pressure, and they will feed the oil at the desired rate in accordance with their adjustment, the speed of the compressor, and the viscosity of the oil.

Oil Control. It is inadvisable to establish any hard and fast rule in regard to the theoretically proper amount of oil that should be supplied to an air-compressor cylinder. There are too many variables involved such as the size of the compressor, its speed, and the condition of the piston and cylinder walls.

The main point is to avoid overlubrication since more trouble will be caused by the use of too much rather than too little oil. Any excess of oil will probably remain in an air-compressor cylinder considerably longer than in the cylinders of either a steam or internal-combustion engine since there is less atomization (owing to lower temperatures) and little or no washing action or dilution of the oil film.

As a consequence, very much less oil will be required per unit of cylinder surface over the same time interval. In theory, about two drops a minute should suffice. As this may not provide sufficient oil at the uppermost parts of the pistons and cylinders in horizontal machines to ensure satisfactory lubrication, the oil feed is generally increased above the usual theoretical requirements. This must not be done to excess, however, since it may lead to the collection of oil pockets in low parts of the machine.

During inspection, if the discharge valves have an oily appearance, enough oil is being fed to the cylinders; on the other hand, if the parts appear too oily or little pools of oil are found in compressor pockets or in any of the air lines, oil is being fed in excess of that required, or it has been improperly distributed.

Another rule states that from one to two drops of oil per 500 to 1,500 sq ft of cylinder area covered by the piston per minute (in reciprocating machines) should assure adequate protection in average service.

The size of the drops of any oil will vary with the viscosity, the temperature, the service conditions, the diameter and shape of the lubricator orifice, and the type of design of the lubricator, *i.e.*, whether it is mechanical force feed or sight feed.

The number of drops per pint will vary. As a result the number of drops secured per minute from an oil having a viscosity of 200 SSU at 100°F would differ from the number obtained from an oil of 300 SSU viscosity.

It, therefore, is not considered good practice to make an absolute recommendation as to the number of drops per minute that should be used in air cylinders of various sizes on account of the great variation in operating conditions that will be met.

Two compressors of the same design and size and built by the same manufacturer may, in fact, be operating in a room under identical conditions, yet it will be practically impossible to secure the same fit of piston rings and valves and the same polished cylinder surfaces. Experience has shown that two such compressors may require a surprisingly different amount of oil for air-cylinder lubrication.

If an oil is unsuitable for compressor service, an excessive amount may be required to keep the pistons operating freely, but excessive oil will cause carbonization in the air passages, particularly on the discharge valves. Sticking of these valves, with the passage of hot compressed air back into the compressor cylinder, is generally indication of overlubrication.

REFRIGERATION-MACHINERY LUBRICATION

Refrigerator-compressor lubrication must be studied with regard to its relation to capacity and efficiency. There must be adequate lubrication but not a continued excess, for immediately this latter occurs, the possibility of oil slugging will develop. This holds true regardless of the type of lubricating system employed.

The service involved and the operating conditions are important factors. Lubrication of such machinery is unique in that the action and effects of the oil upon parts not requiring lubrication, as well as upon the actual wearing surfaces, must be considered.

Disregard of such factors as the method of lubrication, the temperature in the expansion or cooling coils, the mechanical condition of the unit, and the location, type, and efficiency of the oil separator may frequently lead to marked increase in maintenance costs and reduction in capacity.

Oil congealed in any part of a cooling system will tend to reduce refrigerating efficiency owing to its becoming so sluggish under the low temperatures involved as to form an interior lining in the expansion coils and materially affect the heat transfer.

An oil for refrigerating-machinery lubrication must, therefore, remain fluid at the lowest temperatures to which it may be subjected during operation. These temperatures will be encountered in the expansion or cooling side of the system.

Should congealment occur, oil deposits may accumulate on the inner surfaces of the refrigerating piping to form more or less of an insulating medium which will prevent proper abstraction of heat from the compartment or material which is to be cooled. If this is allowed to continue, the capacity of the system will be reduced, and ultimately it will be necessary to clean out these congealed oil deposits.

Consideration must also be given to water. The oil at all times must be practically free from water; otherwise this will freeze if carried over to the cooling coils, in which case it would probably remain in the system and result in a certain decrease in evaporative efficiency. Water may also lead to ice deposits on certain of the valves.

When Viscosity Must Be Considered. Viscosity is a factor where enclosed-crankcase high-speed machines are used in connection with evaporating systems which may allow liquid refrigerant to return to the compressor. Oils should be used that will stand agitation in the presence of the refrigerant and a certain amount of water if present.

The one oil often lubricates the entire machine. As a result it must be capable of serving both the cylinders and the bearings. It should not emulsify to any great extent, for this might result in stoppage of the lubricating system or impairment of refrigeration should oil work past the piston rings and over to the cooling side.

The condition of the valves, piston rings, and stuffing boxes must also be considered in deciding upon the viscosity of the oil.

Practically as important as its lubricating properties will be its sealing and compression-forming ability. If the cylinder walls and moving parts are in first-class condition, a straight mineral oil having a viscosity of from 150 to 200 SSU at 100°F should afford adequate lubrication.

In large machines or in those where the cylinder walls are scored or rings are worn, an oil of higher viscosity may be essential to maintain the requisite seal and degree of compression. Under such conditions an oil having a viscosity of approximately 300 SSU at 100°F should be satisfactory.

Methods of Lubrication. The type and capacity of the compressor control the method of lubrication. The centrifugal machine requires the least complex system since there are only the bearings of the rotor to be lubricated. Ring oilers or pressure circulation are used for oil distribution. Simultaneously, the oil performs an additional function in maintaining an automatic oil seal against loss of vacuum.

Reciprocating or rotary compressors present more parts to be lubricated. Circulation of oil is accomplished by splash or force feed. The system involved for the lubrication of cylinders, stuffing boxes, and enclosed bearings will have a decided influence upon the grade of oil that should be used.

Splash Oiling. Splash lubrication is particularly applicable to fractional-tonnage units and also to vertical machines.

With this type of lubrication there is but little possibility of starved lubrication provided the compressor parts are designed for thorough circulation of the oil after this is begun by the crank splash elements. All that is necessary is to maintain a suitable oil level to enable the splashers to dip to a sufficient extent at each throw of the crankshaft.

Continued operation will result in the crankcase being filled with a lubricating vapor above the main body of oil which will ensure adequate lubrication of main, wrist pin, and crank-pin bearings.

Careful attention is necessary, especially when recharging the case with oil, to see that the level is not raised too high. If it is, churning by the crank can cause such violent agitation in the main body of oil as to retard effective precipitation of any solid impurities that may have gained entry. There would also be possibility of loss of lubricant past the piston rings with subsequent entry of an excess of oil into the condensing and evaporating parts of the system.

Piston-ring Fit. When piston rings are not sufficiently tight, if the crankcase contains too much oil or agitation is too violent, the excess which naturally will reach the cylinder walls will tend to work past the rings as so frequently occurs in an automobile engine.

This is not only wasteful but also a detriment, for oil in the refrigerating lines will impose an added load on the oil separator. Furthermore, if by chance the oil is not of sufficiently low pour test, there will be possibility of its congealment within the system with reduction in the amount of refrigeration developed.

Even with rings in good condition, oil pumping may occur owing to faulty design of the evaporator or careless operation whereby liquid refrigerant enters the cylinders from the evaporator.

Use of excess oil in a splash-oiled system will also involve the possibility of difficulty when draining and cleaning, especially where sludging has occurred.

Continued churning of improperly refined oils will cause sludge formation. In part this is due to oxidation. It will be most probable where water is present or the oil is laden with foreign matter such as dirt, metallic particles, or carbon.

Periodic cleaning must therefore be done with study of the condition of the used oil, for this will very often indicate both the approximate suitability of the latter and the extent to which effective lubrication is attained.

Pressure Systems. Pressure lubrication is widely used on larger types of vertical or horizontal machines. It assures accurate control of the amount of oil delivered to cylinder walls and compressor bearings. It is also possible to filter or purify the oil where there is provision for circulation.

Pressure lubrication is designed to function either alone or in conjunction with the splash system. The objective is to obtain positive circulation of oil throughout the compressor and to eliminate foaming as far as possible. The extent to which foaming may be objectionable will depend upon the oil level and to the location of the suction valves. With compressors that take suction through the crankcase, if the foam level rises to a sufficient degree, foam may be carried over to the high side with the refrigerant seriously to retard heat transfer. Ultimately, if allowed to continue, cleaning of the system may be necessary at considerable expense. To eliminate this, some builders have arranged their design so that all refrigerant gases and vapor are excluded from the crankcase.

Mechanical Force-feed Oilers. External lubricators of the mechanical force-feed type are extensively used where cylinders only are to be pressure oiled. Excellent economy will be attained by regulating such lubricators so that just enough oil is delivered to maintain the requisite lubricating films with the least amount of excess.

It is frequently good practice to lubricate internal and external parts individually. The mechanical oiler should be used with perhaps three outlets for cylinder and stuffing-box service and an independent circulating system for the bearings.

Mechanical force-feed oilers are especially adapted to cylinder and rod lubrication because of an oil lantern or oil recess within the piston-rod stuffing box.

By properly constructing a stuffing box with a lead to come from the lubricator it is possible to operate the piston rod continually through a ring of oil. In this way effective rod lubrication and sealing are maintained.

In some designs the cylinders can be oiled by feeding an excess of oil to the stuffing-box lantern and providing an overflow pipe to carry this to the refrigerant suction line.

This is similar to the principles of steam-cylinder lubrication, the refrigerating gas being impregnated with vaporized lubricant prior to its passage through the compressor.

Hand-pump oilers can also be used for this purpose, but mechanical force-feed units are more positive and require less attention.

Vertical compressors of the enclosed-crankcase marine type are best suited for automatic controlled systems. They are usually of comparatively small capacity and require but little attention beyond periodic inspection.

Pump Design. A variety of pumping devices is used for positive handling of lubricating oil throughout the compressor mechanism. The objective is to maintain controlled lubrication especially where dealing with refrigerants that are miscible with mineral lubricating oils. This necessitates consideration of strainers and oil separators in conjunction with pump design.

All are more or less related to compressor efficiency and capacity, *i.e.*, a continued excess of oil will lead to slugging and reduction in efficiency, yet there must be a slight excess on starting to ensure positive lubrication of piston and cylinder walls, especially in the reciprocating machine. This can be accomplished by proper pump adjustment.

The Gear Pump. The gear pump, based upon automotive practice, has been proved to be as equally dependable in refrigerator-compressor service. The gear pump as designed for positive delivery of oil is a comparatively simple device consisting of a pair of gears mounted in a suitable housing. The normal location of such a pump is in the base of the crankcase of the compressor. Some designers prefer to place it at the lowest part of the case; others are of the opinion that it should be set just above a depression or catch basin in the case to provide means for trapping foreign matter and preventing it from being circulated through the lubricating system. However, there will be a negligible amount of foreign matter in a well-designed system using properly refined oil.

Cylinder Wear. In the vertical machine the side thrust is taken directly by the cylinder wall which tends to make the cylinder wear to oval shape. This thrust varies at different points of the stroke so that the wear on the cylinder wall is not even over the entire stroke.

Horizontal compressors have connecting rods considerably longer than vertical machines, and the component of thrust at right angles to the travel of the piston is much less. This thrust is taken up by the crosshead guides which are readily accessible for adjustment. The only wear in the cylinder, therefore, is caused by that part of the piston weight which is unbalanced by the pressure of the guide and long stuffing box.

The Oil Separator. Wherever an excess of oil finds its way to the evaporating or cooling side, certain detriments will be involved. This is normally prevented by use of an oil separator.

This unit serves to remove any particles of oil from the refrigerant while it is in gaseous form after it has left the compressor. The larger the oil particles, the more effective will be the separator.

The capacity of any separator should be ample so that the velocity of the gas passing through will not be too high. But should an excessive amount of oil be fed to the compressor, a heavy load will be imposed upon the oil separator.

Location and Installation. The separator should be placed between the discharge of the compressor and the point of entry of the gas into the condenser. In certain machines a drain valve can be installed below the condenser to enable removal of any oil that may have passed the separator.

Where the oil separator fails to function properly, the reason is that it is set too near the compressor, the rush of hot gas preventing proper condensation and collection of the oil.

Oil will always vaporize to some extent owing to the heat which exists. This oil vapor will naturally tend to pass into the system with the refrigerant, to condense, and remain in the colder parts unless it is effectively removed before it enters the condenser.

This can be avoided by locating the separator as close to the condenser and as far away from the compressor as possible. It should always be of sufficient size to allow ample reduction in the velocity of the gas in order to permit effective separation.

A smaller separator located some distance from the compressor may often prove more effective than a large separator located nearby.

Where the main oil separator must be located adjacent to the compressor, an oil of as low an atomizing tendency as possible should be used. This property will usually accompany high viscosity. The choice of a heavier oil would, therefore, be advisable. In general, a viscosity of from 200 to 300 SSU at 100°F is satisfactory.

The efficiency of an oil separator can be checked by comparing the amount of oil removed from it with the amount fed to the compressor. Any extensive difference would indicate that the oil is not being entirely removed or trapped.

Allowance should be made for oil leakage around the stuffing box. To just what extent this may occur will depend on the installation, the care given to lubrication, and the viscosity of the oil.

Sealing the Stuffing Box. Suitable stuffing-box seals and properly lubricated piston rods are very important. Ammonia has a certain corrosive action upon brass, copper, and bronze. Stuffing boxes for such service are therefore built of cast iron or steel, using metallic, asbestos, or rubber packing.

The "Oil Lantern." In some types of machines a hollow space or "oil lantern" is located between two separate sets of packing.

This space surrounds the rod and is filled with oil. It not only serves as a seal to prevent loss or leakage of ammonia but also serves as an effective means of piston-rod lubrication. Lubricant is usually fed to the "oil lantern" by means of a hand or automatic oil pump.

Where the piston rod is efficiently lubricated, its surface will have a smooth gloss and will show a light film of oil. There will be no indication of overheating, and a relatively perfect seal will be maintained with a minimum of leakage.

In some types of double-acting machines, the "oil lantern" serves also as an auxiliary means of introducing the lubricant to the compressor cylinder by allowing oil to work past the piston-rod packing.

There is an added advantage to this method in that certain grades of packing which might be reacted upon by ammonia gas will be protected by the lubricant. This "lantern" can also be connected to the suction so that the outer set of packing is under low pressure.

Significant Physical Characteristics of Refrigerator Oil. *Pour-point Test.* Refrigerator oils are manufactured to pour points[2] ranging from −15 to −45°F, varying with the viscosity and type of oil. With the trend toward temperatures approaching −100°F in deep-freeze work, metals assembly, and medicine, very low wax-separation point has also become a factor. The pour point is associated with the wax content and viscosity of the oil. Thus, as either of these increases, the pour point will tend to rise. With some oils the recorded pour point may be due to both wax content and the viscosity or to the viscosity alone; in the case of the latter, the term "viscosity pour" is used. Oils derived from naphthenic-base crudes will ordinarily show a "viscosity pour," as these, by their nature, contain very little if any wax.

"Wax" Floc. "Wax" floc is variously interpreted as the temperature at which a 10 per cent oil −90 per cent refrigerant mixture, when chilled, changes from clear to hazy, or the temperature at which the mixture develops a definite floc.

The manufacture of wax-free oils has become increasingly important with the adaptation of refrigeration machinery to temperatures considerably below the conventional 0 to −25°F usage range. When a compressor, its lubricating oil, the expansion valve, and other parts must function at temperatures around −80 to −100°F or lower, it is important to consider the presence of any material in the oil that would congeal to interrupt the refrigeration cycle.

The petroleum chemist has spent much time studying hydrocarbons that would tend to congeal and solidify in the oil when exposed to very low temperatures. There seems to be no direct relation between the usual physical tests of these hydrocarbons and the pour test. Two refrigeration-grade oils may have practically the same physical characteristics including pour point, yet one may show a "wax" floc from 20 to 30° lower than the other. Obviously it is desirable to remove objectionable waxlike hydrocarbons in the course of refining the oils. This is the objective in modern refinery practice. When more positive information is available as to their true identity, the economics of the refining procedure may well be affected.

R. J. Thompson and W. W. Rhodes, of Kinetic Chemicals, Inc., in discussing Freon 22, have suggested some pertinent thoughts. Should oil separation, wax separation, and accumulation of wax at the low temperatures required be experienced, they state:

1. Use an oil as low in viscosity as is practical for satisfactory compressor operation.
2. Use an oil which has been dewaxed or degummed at a sufficently low temperature so that any remaining wax will not be precipitated out of the oil at the lowest operating temperature when the oil is in solution with liquid Freon 22.
3. Install an oil separator between the compressor and the condenser and as close to the cylinder head as possible, so as to prevent the oil containing wax passing through the condenser, receiver, liquid line, expansion valve, and into the low-temperature evaporator.

Water. Water in a refrigerator oil can exert several bad effects:

1. It can throw off the accuracy of the floc test by creating a false cloud.
2. It can contribute to chemical reaction with some refrigerants.
3. It can become a potential cause of stoppage owing to formation of ice particles at the expansion or regulating valve.

Water can be removed from the oil before the latter is charged to the refrigerating unit by blotter pressing which assures complete removal of suspended moisture from a new oil. For extreme low-temperature service, where it may be desired to remove even dissolved moisture at the time of charging the system with oil, this can be done by vacuum dehydration at elevated temperatures assisted by dry air or nitrogen. Where a refinery-dehydrated oil has been purchased, the storage must be so planned that the oil will not be exposed to air containing moisture. Petroleum oils readily absorb moisture from the air. A refrigerator oil, therefore, must be most carefully handled.

The Chemical Effect of Water. In an SO_2 system, water will react with the refrigerant to form a corrosive acid which can attack certain metallic parts of the system.

Water is serious in a methyl chloride system as it may produce corrosion products owing to slow reaction of the methyl chloride with water to form hydrochloric acid.

Table 18-7. Refrigeration-lubrication Chart

Refrigerant	Chemical formula	Primary use	Type of compressor	Behavior in presence of mineral lubricating oils	Approx. viscosity, SSU 100 °F	130 °F	210 °F
Ammonia.......	NH_3	Ice manufacture	Reciprocating	Some types of impure ammonia may cause sludge	155	82	42
Carbon dioxide	CO_2	Cold storage, ice manufacture	Reciprocating	No change	155 315 500	82 138 204	42 47 53
Sulfur dioxide. . .	SO_2	Electric refrigeration	Reciprocating	May go into solution to cause sludge or gums	155	82	42
Methyl chloride .	CH_3Cl	Electric refrigeration	Reciprocating	Mixes to cause reduced viscosity and lower pour test	315	138	47
Carrene No. 1 . .	CH_2Cl_2	Air conditioning	Centrifugal	Same as above	315 155	138 82	47 42
Freon 12......	CCl_2F_2	Electric refrigeration, cold storage, air conditioning	Centrifugal, reciprocating, rotary	Same as above	315 500	138 204	47 53
Freon 114.......	$C_2Cl_2F_4$	Industrial and electric refrigeration	Rotary, centrifugal or reciprocating	Same as above	315	138	47
Freon 11 or Carrene No. 2........	CCl_3F	Air conditioning	Centrifugal	Same as above	315	138	47
Freon 21..........	$CHCl_2F$	Air conditioning	Rotary	Same as above	315	138	47
Freon 113.	$C_2Cl_3F_3$	Air conditioning	Centrifugal	Same as above	155	82	42
Freon 22... ...	$CHClF_2$	Ultralow-temperature service, air conditioning	Reciprocating	Same as above	315	138	47

Water also can promote "sludge" formation and copper "plating." It is regarded by authorities as the chief offender in regard to copper "plating."

In systems using any of the Freon refrigerants, water beyond the point of saturation may promote reaction to corrosion products. Over long periods of time these refrigerants may slowly react with water and develop acid products to a sufficient extent to present a corrosion problem.

Significance of the Dielectric Strength.[3] A high dielectric strength is today used as an indication that a refrigerator oil will not contain moisture to such an extent as to cause trouble. While the dielectric strength does not indicate moisture content in terms of percentage by weight, there is some correlation between the two, *i.e.*, an oil of some 30,000 volts dielectric strength may contain as much as 0.01 per cent moisture by weight; a 25,000-volt oil may contain as much as 0 03 per cent moisture by weight.[4] Separation of any of this dissolved water by a slight drop in temperature will immediately lower the dielectric strength to 10,000 volts or lower. These quantities in ultra-low-temperature refrigeration can be considered harmful.

Relation to Moisture Solubility. Solubility of water in the oil is regarded as significant. Water in solution becomes water in suspension as the temperature of the oil is reduced owing to temperature-solubility effect. Research has indicated that moisture present in an oil up to its saturation point has little if any effect on the dielectric strength. Presence of moisture above the saturation point, however, immediately causes considerable loss in dielectric strength. The temperature is a factor. If an oil is saturated with moisture at a higher temperature and the dielectric strength is run at a lower temperature, a low value will be obtained at the latter temperature in contrast with a much higher value which may be observed at the higher temperature. This indicates the need for dehydration below any moisture content indicated by the dielectric-strength test if the oils are to be used in ultra-low-temperature service.

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SILICONES AS SPECIAL LUBRICANTS*

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Editor's Note. There is increased activity in the development of synthetic lubricants and the improvement of mineral-oil lubricants through the use of various additives and inhibitors. Since the Second World War the synthetic field has been developing rapidly. The armed forces and industry require low pour point (−40 to −80°F), low viscosity, noncorrosive fluids for use in instruments such as gun sights, recorders, compasses, fuse mechanisms, fire controls, and depth bombs. Ordnance and aviation equipment requires chemically stable and low-pour-point lubricants having nonvolatile characteristics. Lubricants possessing these qualities are the long-chain esters commonly referred to as "diesters." These synthetics have been employed successfully as lubricants in the liquid state, blended with mineral oils, or dispersed in various metallic soaps to form low-temperature greases. The latter have been successfully employed in aircraft and ordnance fire-control equipment. Lithium-soap greases made from diesters have shown excellent oxidation stability, low torque, and good lubricating properties.

Through various processes such as polymerization, chlorination, hydrogenation, and alkylation, a large number of important synthetic lubricants have been developed. Their use in industry is growing, although in general, the high cost of these materials is influential in preventing rapid industrial adoption as long as mineral oils are available at attractive prices. The polyalkylene glycols are being produced commercially and have been used successfully in many applications including high-temperature service, lubricants for rubber, heat-transfer systems, as hydraulic fluids, and in cutting and drawing operations. They are available as water-soluble or water-insoluble products in a wide range of viscosities. Like many of the other synthetics, the polyalkylene glycols have low pour points and high viscosity indexes.

Because of the rapidly changing synthetic picture, no effort is being made to cover the multitude of products that are available for specific services. The following subsection on silicones describes the unusual characteristics of one particular family of synthetics.

INTRODUCTION

Silicone oils and greases are perhaps the newest and most interesting contribution of high-polymer chemistry to the field of lubrication. The fluid silicones and the greaselike compounds made from them extend the temperature range of effective lubrication for certain applications beyond that now covered by petroleum oils and greases. This is a result of the heat stability of the silicone polymers combined with other desirable properties such as oxidation resistance, low volatility, inertness, high flash point, and water insolubility.

A variety of silicone products is now commercially available for specialized lubrication uses. All of them have as their basic structure a chain of silicon atoms, joined to each other through an oxygen atom with one or more organic groups attached directly to each silicon atom.

The inherent stability of silicone products is due to their chemical structure. Their skeleton of silicon–oxygen–silicon atoms is highly resistant to oxidation and to thermal decomposition. A wide variety of organic groups can be attached to the silicon atoms to alter such physical properties as viscosity index, lubricity, and pour point. The size of the silicone molecules can also be varied to affect such properties as intrinsic viscosity, flash point, and volatility.

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* Superior numbers refer to specific references listed at the end of this subsection.

SILICONE FLUIDS

Silicone fluids are made in a wide variety of types and viscosities. In all their various forms these liquid silicones are crystal clear, stable to heat and oxidation, incompatible with water and most organic materials, and inert toward metals. They vary considerably in lubricity depending upon their molecular size and chemical structure. The fluid types now available and of most general interest to lubrication engineers are listed in Table 18-8. A detailed description of each type will be given to aid in selecting the proper silicone lubricant.

Table 18-8. Physical Properties of Representative Silicone Fluids Useful as Lubricants

Principal use	Fluid type	Viscosity grade		Viscosity-temperature coefficient	Pour point (ASTM)		Volatility after 48 hr		Flash point, min deg, F	Lb per gal at 25°C	Specific gravity, 25°C
		Centistokes at 25°C	SSU at 100°F	$1 - \dfrac{V \text{ at } 210°\text{F}}{V \text{ at } 100°\text{F}}$	Deg C	Deg F	At deg C	Per cent			
High-temperature lubrication.....	A	500	1,060	0.89	−12	10	250	< 15	600	9.25	1.11
Lubrication at both high and low temperatures...	B	100–150	300–400	0.75	−48	−54	250	< 15	600	8.90	1.08
Hydraulic fluid (special metals)	C	100	355	0.60	−55	−67	200	< 2	600	8.08	0.969
Antifoam additive	C	1,000	3,500	0.62	−50	−58	200	< 2	640	8.12	0.973

Type A Silicone Fluid. The Type A silicone fluid was developed for general lubrication at temperatures above the range of most organic lubricants. It remains liquid down to 10°F (−12°C) and useful up to 500°F (260°C). This silicone fluid is comparable in lubricity to petroleum oils of the same viscosity at normal temperatures under light to moderate loads. At elevated temperatures, it is generally superior because of its high flash point, relatively low volatility, and freedom from gumming.

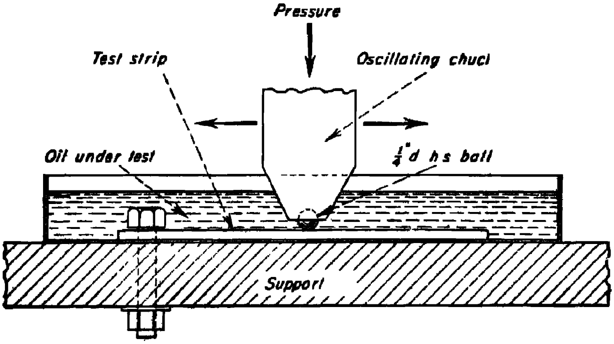

Fig. 18-5. Sliding-ball lubricant wear tester.

To evaluate the lubricity of this fluid at room temperature under various loads and between various metal combinations, a hardened-steel ball in a chuck was made to slide over metal strips covered with the test oil. Wear of the metal strip measured in terms of weight lost is plotted against time in Fig. 18-6 and against load in Fig. 18-7.

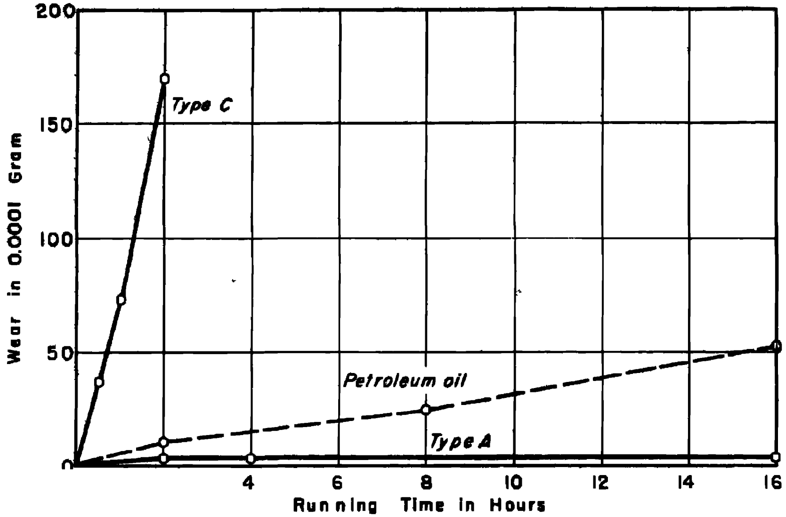

FIG. 18-6. Wear-tester results showing wear at various intervals of time when using a steel ball and a brass strip.

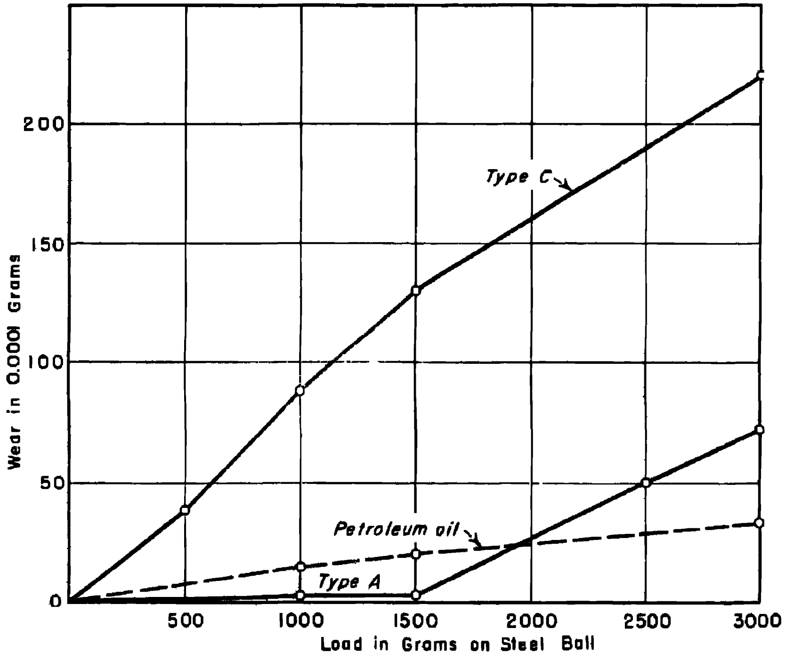

FIG. 18-7. Wear-tester results showing wear under various loads when using a steel ball and a brass strip.

Comparable values for an SAE 20 oil are also shown. The device used in this series of tests is shown in Figure 18-5.

Silicone oils of this type are used for lubricating mechanical devices such as clocks, instruments, conveyor bearings, and oven machinery exposed to high temperatures. Where high humidities are also involved, this silicone oil is available with a rust inhibitor added. An indication of serviceability as a lubricant for conveyor chains is given by a simple test device in which bicycle chains are cycled slowly through an oven at 350°F for 80 min, then through room temperature for 40 min, then through a humidity chamber for 80 min, and then back through room temperature to the oven. Conveyor chains lubricated once with this type of silicone oil remain operable for more than 500 days of such cycling, while an organic oil designed as a lubricant for oven conveyors oxidized and allowed the roller bearings to rust and freeze in 35 days.

Type A silicone fluid is normally supplied in a viscosity grade of 500 centistokes at room temperature, but higher viscosities can be made if the use requires them. It has the steepest viscosity–temperature slope of the silicone fluid listed in Table 18-8, but it has the same order of heat stability and oxidation resistance.

Colloidal graphite suspended in this type of silicone fluid is used for very-high-temperature lubrication because the silicone fluid completely wets bearing surfaces and carries with it the finely divided graphite to form a smooth and tightly bonded finish which has very good lubricity and permanence.

Type B Silicone Fluid. Type B silicone fluid is designed for use as a lubricating oil at temperatures ranging from −54°F (−48°C) to over 300°F (149°C). Its stability to heat is demonstrated by the fact that it does not gel or decompose to form gums after 1,000 hr of exposure to 482°F (250°C). Type B silicone oil has good lubricity under medium to light loads between hardened-steel surfaces and bronze, babbitt metal, or hardened steel, but it is not suitable for moderate to heavy loads or for bearing surfaces of cast iron or soft steel. This silicone fluid is also inert to metals but will prevent atmospheric corrosion only of nonferrous metals and alloys.

Type B fluid is available with a rust inhibitor to prevent rusting of iron or steel surfaces. This type of silicone oil with rust inhibitor added is used as a relatively permanent lubricant at both high and low temperatures for instruments, clocks, and conveyor chains.

Type C Silicone Fluid. Type C silicone fluids are available in viscosity grades ranging from 100 to upward of 30,000 centistokes. These silicone fluids have properties unlike those of any other liquids, and they have therefore found a variety of applications including use as damping and hydraulic fluids, liquid dielectrics, mold release agents, waterproofing materials, and special-purpose lubricants. Only two viscosity grades of the Type C fluids that are of interest to lubrication engineers are shown in Table 18-8.

Type C silicone fluids are water white and inherently stable liquids which have flash points of 600°F or higher and pour points in the range of −58°F (−50°C). They show exceptionally little change in viscosity with temperature as shown by the data given in Table 18-8 and in Fig. 18-8.

Type C silicone fluid in the 100-centistoke viscosity grade is a useful hydraulic fluid. Comparative testing of silicone fluids of this type and a petroleum oil, meeting the specifications of AN-VV-O-366b, was done by the Naval Research Laboratory.[1] These experiments showed that there was no significant change (less than 2 per cent) in the viscosity of the silicone fluid after circulating at 1,500 psi continuously for 500 hr or 105,000 cycles. The petroleum oil used as a control retained less than 50 per cent of its original viscosity after 18,000 cycles at the same pressure. This silicone fluid has been operated successfully in an aircraft hydraulic system at pressures as high as 3,000 psi and at temperatures up to 200°C.

In these tests, however, it was found that the silicone fluid had very limited load-carrying capacity where both loaded sliding surfaces were ferrous. Where one or both of the bearing surfaces are nonferrous or can be replaced by suitable nonferrous metals, the silicone fluid gives excellent performance. For example, a steel thrust knuckle in a Vickers piston pump showed wear, but performance was entirely satisfactory when a bronze knuckle was used. In a Pesco gear pump, there was no wear on the hardened-steel gears but considerable wear on the cast-iron end bushings. Replacement with bronze bushings gave satisfactory performance.

The performance characteristics of this silicone fluid in unilaterally loaded journal bearings was also studied intensively by the Naval Research Laboratory.[2] It was found that this fluid was a good lubricant between certain metal combinations such as bronze on steel but poor between steel on steel. It was also found that heating the bearings in this silicone fluid at 150°C causes the formation of a silicone "lacquer" film on the steel surfaces which minimizes wear. The same kind of film was formed when bearings were run in carefully under light loads at elevated temperatures. After the silicone "lacquer" film was formed, loads as high as 5,500 psi could be applied to the journal bearings.

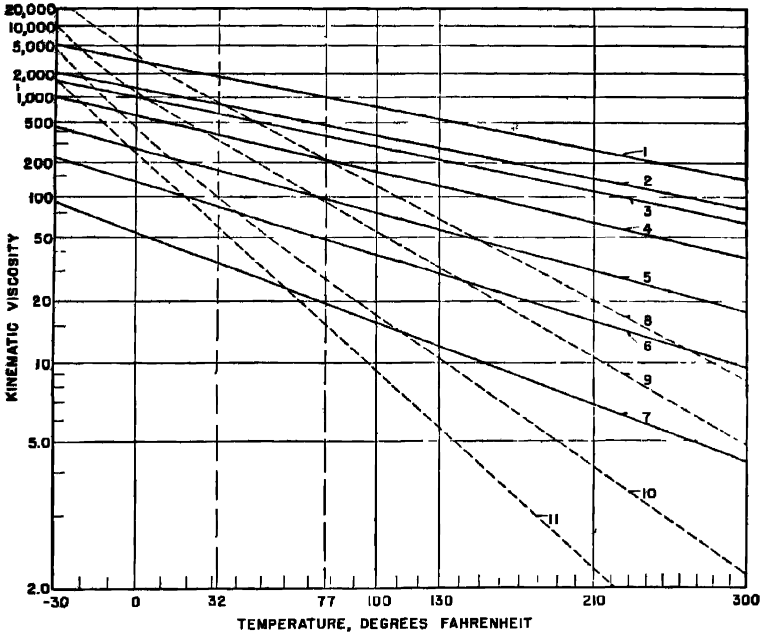

Fig. 18-8. Viscosity-temperature slopes showing viscosities at various temperatures of seven viscosity grades of Type C silicone fluid and four petroleum oils.

Type C Viscosity Grades, Centistokes

1. 1,000
2. 500
3. 350
4. 200
5. 100
6. 50
7. 20

Petroleum Oils

8. Hydraulic oil A
9. Hydraulic oil B
10. Hydraulic oil C
11. Transformer oil

This same silicone fluid in the 1,000-centistoke grade is used as an antifoam additive for petroleum oils. It is effective in concentrations as low as 1 part to 100,000 parts of oil.[3]

High-viscosity Type C silicone fluids are being used in a torsional-vibration damping device on automotive and diesel-engine crankshafts.[4] Torsional vibrations of the shaft are damped out by the shear resistance of the silicone-oil film which separates a freely suspended flywheel encased in a hermetically sealed housing attached to the shaft.

SILICONE GREASES

Silicone greases are compounded from certain silicone oils and thickening agents that have a high order of heat stability. At the present time there are three types of silicone greases made to conform to the value given in Table 18-9.

Table 18-9. Test Data for Representative Silicone Greases

	Type I	Type II	Type III*
Worked penetration (ASTM D217-44T)	260–300	310–330	310–330
Flash point—determined on oil—(ASTM D92-46)	600°F	600°F	525°F
Dropping point (ASTM D566-42) not less than	415°F	415°F	415°F
Maximum per cent bleed at 300°F†	4.0	2.0	3.0
Maximum per cent evaporation at 300°F†	1.0	1.5	3.0

* Types II and III silicone grease can be supplied in different consistencies.
† Method AN-G-3a 24-hr test.

Volatility of Silicone Greases. Even though performance is the ultimate test of a lubricating grease, volatility can be measured and service life tentatively predicted from the measurements. In Table 18-10 the volatility of three silicone greases is compared to the volatility of high-temperature hydrocarbon greases.

After 40 hr exposure at 175°C, a crust had formed over the surface of the petroleum greases. The silicone greases were still soft and serviceable.

Type I Silicone Grease. Type I silicone grease, compounded of very finely divided carbon black and a silicone fluid of good lubricity, does not melt at temperatures above 500°F. It gradually thickens, however, at temperatures above 400°F, and more frequent lubrication is required at operating temperatures in excess of 300°F. Type I silicone grease is serviceable in antifriction bearings operating at speeds up to 4,000 rpm and at temperatures between −40 and 400°F.

Table 18-10. Percentage of Weight Lost after 40 hr at 347°F (175°C)

High-temperature petroleum greases			Silicone greases		
			Type I	Type II	Type III
58.3	39.3	34.8	4.2	3.3	4.6

In one test device, No. 212 ball bearings with bronze retainer rings were mounted on a shaft and spring loaded with 150 lb radial load. Both the shaft and bearings were enclosed in a housing that could be heated to any desired temperature. At a speed of 1,750 rpm, the average bearing life after one application of Type I silicone grease was 6,000 hr at 257°F (125°C) and 1,300 hours at 302°F (150°C).

Despite its more limited lubricity between steel bearing surfaces, Type I silicone grease has been used successfully in the bearings of the rotating spreader of an Economic stoker. The bearings in this spreader have steel retainers but are still serviceable after 2 years of nearly continuous operation at 300 to 350°F and at a speed of 900 rpm.

This silicone grease has proved to be especially useful for lubricating pumps handling hot fluids and for lubricating oven machinery and conveyor systems operating at high temperatures and in corrosive atmospheres. It is not satisfactory for lubricating high-speed bearings operating at speeds of more than 4,000 rpm.

Type II Silicone Grease. Type II silicone grease, compounded of a silicone fluid and a heat-stable metallic soap, is designed for the lubrication of antifriction bearings operating at speeds up to 20,000 rpm and at temperatures ranging from −40 to 350°F. Over this temperature range, it is practically nonvolatile and highly resistant to hardening. It will separate, however, at temperatures above 400°F. Test results summarized in Table 18-11 indicate that this silicone grease is an almost completely stable lubricant suitable for the permanent lubrication of ball bearings.

Table 18-11. Life Test on Type II Silicone Grease

Bearing temperature		1,750-rpm, tester life, hr	10,000-rpm, tester life, hr	1,750-rpm, electric-motor life, hr
Deg C	Deg F			
115	239	. .	.	5,600
125	257		6,500	3,800
150	302	2,600	2,600	
200	392	450		

Single Bearing, 1,750-*rpm Tester.* Tested in a No. 212 bearing with a bronze retainer ring mounted on a shaft spring loaded with 150 lb radial load in a heated enclosure, Type II silicone grease had a life of 2,600 hr at 302°F (150°C) and 450 hr at 392°F (200°C).

10,000-*rpm Tester.* The 10,000-rpm tester, shown in Fig. 18-9, consists of two standard No. 204 bearings in electrically heated housings and a belt-driven shaft. The shaft is driven at a speed of 10,000 rpm. Bearing failures are indicated by changes in torque, as measured by temperature rise in the bearings, and by changes in the hot and cold coasting time.

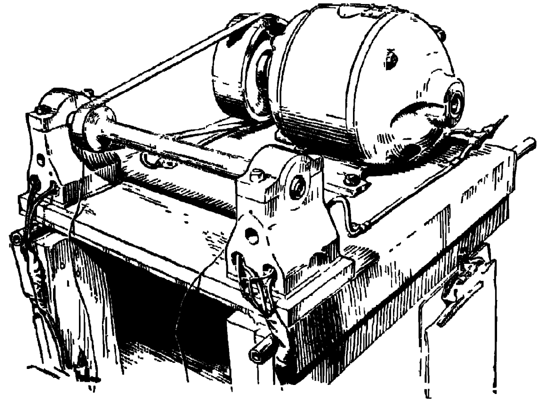

FIG. 18-9. Grease tester, 10,000 rpm using No. 204 bearings (housings are electrically heated).

Type II silicone grease failed after 2,600 hr at 302°F (150°C) and after 6,500 hr at 257°F (125°C) in this device. During these tests, however, the bearing temperature were inadvertently allowed to go above 200°C on two separate occasions. Once the temperature went up to 437°F (225°C) for 1 hr, and another time the temperature went up to 464°F (240°C) for 15 min. It is likely that the grease life was considerably shortened and probably the bearing metals were harmed by these runaway temperatures.

1,750-*rpm Silicone-insulated Motor.* To check these laboratory test results, this silicone grease was also tested in the bearings of silicone-insulated electric motors. The bearing temperatures are controlled by loading 7½-hp 1,750-rpm, 440-volt motors through belt-driven generators. The motors were insulated with inorganic materials bonded and impregnated with silicone resins and varnishes. They could therefore be operated at temperatures up to 590°F (310°C) by resistance to bring bearing temperatures up to 302°F (150°C).

A bearing lubricated with Type II silicone grease and operated at 239°F (115°C) failed after 5,600 hr in this test. A second bearing failed after 3,800 hr at 257°F (125°C).

From the data given in Table 18-11 it appears that Type II silicone grease gives long life to bearings operating at 239°F (115) to 257°F (125°C). At normal operating temperatures for electric motors, Type II silicone grease may be expected to give relatively permanent lubrication in sealed bearings. Such bearings are now being made for some of the leading manufacturers of electric motors.

Type III Silicone Grease. Type III grease is made of another silicone fluid thickened with a metallic soap. It is especially designed for low-temperature lubrication and is serviceable over a temperature range of −95 to 300°F and at speeds up to 20,000 rpm. Even though it is designed primarily for low-temperature applications, Type III has superior lubricating life at temperatures up to 300°F.

This grease is suitable for use as a permanent lubricant in sealed bearings subjected to extreme temperature variations and to outdoor weathering. For example, Type III is being used to lubricate the annular ball bearings in a wind-direction and velocity indicator which is exposed to the full range of ground temperatures and to the rain and weather. These bearings are relatively inaccessible so that relubrication is difficult. Type III was specified to meet these adverse operating conditions. It is also being used to lubricate certain aircraft bearings.

Other Silicone Greases. None of the following silicone greases is designed for lubricating bearings. They form heat-stable films which have good resistance to a variety of chemicals and sufficient lubricity to facilitate the turning of stopcocks, valves, and flowmeter bearings.

Stopcock Grease. The first silicone greases were designed for general laboratory use. One of these silicone greases effectively lubricates glass and ceramic stopcocks handling laboratory reagents including alkaline materials. Besides being nonvolatile, it does not harden or melt over a temperature range of −40 to over 200°C. This grease also facilitates the assembly and disassembly of rubber hosing and glass tubing and protects the rubber from oxidation and deterioration due to ozone.

High-vacuum Grease. High-vacuum grease is a special-purpose silicone lubricant which has properties similar to the silicone stopcock grease except that it is more cohesive and better suited to high-vacuum uses. It seals without channeling and effectively lubricates stopcocks and other ground-glass joints in vacuum systems operating at pressures of less than 10^{-6} mm Hg.

SILICONE LUBRICANT FOR PRESSURE-LUBRICATED VALVES AND FLOWMETERS

Another special-purpose silicone lubricant is used to lubricate and seal pressure-lubricated valves, automatic-control valves, and flowmeter bearings. It is resistant to a wide variety of chemicals and to relatively high temperatures. It is useful from −40 to 400°F and is resistant to steam.

This silicone valve lubricant is insoluble in water and unaffected by aqueous chemicals. It is noncorrosive to metals and useful against steam and many corrosive chemicals. It forms an effective seal in systems operating at high pressure or under vacuum.

SUMMARY

Silicone lubricants were developed to give effective lubrication under adverse conditions, including abnormally high- or low-temperature operation. Their function is to supplement rather than to displace petroleum oils and grease.

Silicone lubricants are especially useful in applications where both high and low temperatures are involved or where more nearly permanent lubrication is required. In general, the silicone oils and greases would seem to justify consideration in most applications to which petroleum oils and greases are not well suited.

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SECTION 19

MECHANICAL PACKINGS

BY F. C. THORN AND J. R. BOYER

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  C. Rotary displacement pumps and turbo-pumps
  D. Rotary gas compressors
  E. Agitators
  F. Fluid drives, fluid brakes, and rotary shock absorbers
  G. Swivel joints
  H. Hydraulic turbines and propeller shafts
  I. Single-acting reciprocating air compressors
  J. Bearings and gear cases
IX. Valve-stem and expansion-joint packings
X. Miscellaneous sliding-contact packings
XI. Gasket and sliding-contact-packing classification and recommendations
XII. Diaphragms
XIII. Packing testing and evaluation
  A. Field tests
  B. Bench tests
  C. Laboratory tests
    1. Tensile and elongation tests
    2. Set tests
    3. Hardness tests
    4. Hot-air aging tests
    5. Solvent or oil-immersion tests

MECHANICAL PACKINGS

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DEFINITIONS

The terminology of mechanical packings has never been accurately defined. The following definitions are believed to conform to accepted usage.

A *packing* is a nonrigid element designed for inclusion in the wall of a fluid container and designed to prevent or minimize leakage between the rigid parts of such a container.*

Packing (collective) is material in sheet, strip, or bulk form from which packings may be prepared.*

A *gasket* is a packing designed for inclusion between rigid parts of a fluid container in essentially stationary relationship.*

Gasketing (collective) is material in sheet, strip, or bulk form from which gaskets may be prepared.*

A *valve disk* is a specialized gasket composing the working face of a rigid valve and designed to improve its tightness when closed.*

(NOTE: In certain applications, as in the check valves of reciprocating pumps, the valve disk may compose the entire valve. More generally it is contained in a rigid *disk holder*.)

A *valve seat* is identical with a valve disk, except that it composes the stationary face of the valve. When a valve seat is employed, the disk itself may be wholly of rigid material.

A *sliding-contact packing* is designed for inclusion between rigid parts of a fluid container which are capable of relative motion. The packing is in stationary contact with one part and in sliding contact with the other part.

A *packing ring* is a sliding-contact packing designed for insertion into the annular space between two concentric cylinders in relative motion.

A *packing set* is a plurality of packing rings designed to fill such annular space which is generally adjustable for depth in order to bring pressure to bear on the packing set. The movable end wall is known as a "gland" and the annular space as a "stuffing box."

An inner cylindrical member to which a packing ring or set is affixed is a *piston*, and the packing ring or set used with it is a *piston ring* or *piston packing;* the annular space in which the packing is held is known as a *piston groove.*

An inner cylindrical member which has no packing affixed to it but which moves through a packing is known variously as a *rod*, *plunger*, *ram*, *valve-stem*, or *shaft*, according to size, function, and type of motion. Packings used with these are designated accordingly as *rod packings*, *plunger packings*, etc.

Cup packings, *hat* or *flange packings*, *U packings*, *V packings*, *O packings*, and *cones* are variants on the usual rectangular cross section of packing rings and, in the main, are self-descriptive terms. Although usually employed in sliding contact, any of these may be used as a gasket.

The term "seal" in the loose sense is applicable to anything that stops a leak and is therefore synonymous with "packing." In the restricted sense, the term is employed to designate one of two types of rotary shaft packing. A *radial seal* (sometimes, because of its frequent use in conjunction with bearings, known simply as an *oil seal*) is a hat or flange packing which is forced to contract against the shaft by the action of an internal spring. An *axial* or *shoulder seal* is a flexibly suspended packing ring making sliding contact on one of its flat faces with a similar ring which may be either rigidly or flexibly supported; one of the pair is affixed to the shaft and the other to the stationary part of the fluid container. Shafts packed with one or the other type of seal are frequently called "packless" but erroneously, since these are specialized packings.

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* These definitions were accepted by ASTM Committee D-11, Subcommittee VI on Packings, and were published in the *ASTM Bull.*, May, 1940.

A *diaphragm* (in the packing sense) is a packing intended for inclusion between parts of a fluid container in limited relative motion and designed to absorb such motion by its own deformation.*†

The common forms of packing diaphragm are the *flat diaphragm* and the *cylindrical diaphragm*, or *bellows*. Either can function as a valve stem or piston packing.

Diaphragm sheet is sheet from which flat diaphragms may be cut.*

PACKING MATERIALS AND THEIR FABRICATION

The number of materials and fabricating methods employed by the packing industry is legion; space will permit only a brief reference.

Metals. Under certain conditions metals become sufficiently "nonrigid" to qualify as packings. Solid iron, copper, lead, etc., in the form of sheet, strip, wire, etc., are made into gaskets. Thin metal sheet is employed with asbestos paper, etc., in the manufacture of corrugated, jacketed, and spiral-wound gaskets and also for metal diaphragms. Braided wire or twisted foil are employed as rod packings. Wire is also used to strengthen asbestos yarns and, in the form of wire cloth, to reinforce rubber gasketing. The harder metals, such as cast iron, bronze, and babbitt, are used to construct axial-seal rings and segmental rod and piston packings; they are also used for such packing adjuncts as end rings and lantern rings. Steel is used for cases and springs. Comminuted metal is used in "plastic" packing.

The methods of manufacture are equally varied and include casting, rolling, forging, extrusion, wire drawing, cutting, stamping, spinning, twisting, wrapping, braiding, welding, brazing, machining, and pulverizing, none of which needs to be described here.

Carbon. Carbon in solid form (amorphous or graphitic) is machined into axial seals and segmental rod packings.

Natural Rubber. The first step in the conversion of natural rubber to packings is mastication followed by the incorporation of compounding ingredients. The most important compounding ingredient is sulfur, since this material combines with the rubber under the influence of heat and time to form the nonplastic "vulcanized" or "cured" rubber. With sulfur are added small quantities of organic accelerators to speed up vulcanization, zinc oxide to "activate" the accelerators, plasticizers to render the compound workable, and fillers to stiffen it to the desired extent. Prolonged vulcanization, in the presence of sufficient sulfur, also increases the stiffness of the compound, the end product being "hard rubber," or "ebonite."

Since vulcanization renders the compound nonworkable, it is postponed until late in the manufacturing operation. In the manufacture of packings, the unvulcanized rubber compound is treated in one of the following ways:

1. It is calendered into single- or multiple-ply sheets, which may be (*a*) vulcanized on drums with a fabric liner, which, after removal, leaves a fabric impression on the rubber sheet (this is the method employed in preparing common "red sheet"); (*b*) vulcanized in long hydraulic presses, giving a smooth surface; (*c*) rolled on a mandrel and vulcanized in steam in the course of which it coalesces to form a sleeve from which gaskets may be cut; (*d*) cut into blanks for molding.

Molding involves the simultaneous application of pressure and heat to rubber compound contained in a confined space, in the course of which the compound assumes the shape of the space and is simultaneously vulcanized. The space, of the desired shape and size, is carved into a block of steel, which is usually separable into two or more parts at the conclusion of the molding operation to facilitate discharge of the molded article. The parting lines between these parts provide channels through which excess compound exudes in the form of a thin fin; this fin is afterwards removed, but enough of it generally remains to identify the product as a molded article. Four systems of molding are generally recognized:

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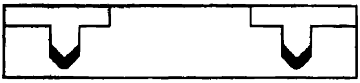
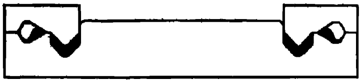
* These definitions were accepted by ASTM Committee D-11, Subcommittee VI on Packings, and were published in the *ASTM Bull.*, May, 1940.

† In the broad sense, hose, auto tires, and even balloons are flexible parts of fluid containers and hence "diaphragms." Perhaps a packing diaphragm is best defined as a "diaphragm that functions like a packing."

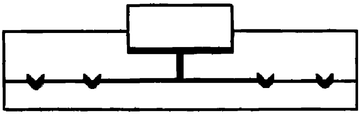
a. "Closed" molds (Fig. 19-1*a*), in which no particular provision has been made for the escape of that portion of the volume of the blank in excess of the mold volume when the mold parts are forced together by a hot hydraulic press. Since the use of a closed mold with a pure-rubber compound or other pure plastic requires that the blank be accurately weighed, the method is used more generally with fabric packings (see 2 below) in which internal voids are present to compensate for inaccuracies in blank preparation.



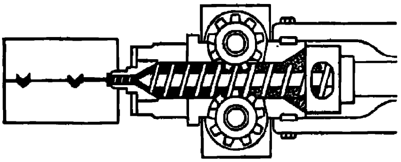
(a) Closed mold



(b) Open or flash mold



(c) Transfer mold



(d) Injection mold

FIG. 19-1. Types of rubber compound molds.

b. "Open" or "flash" molds (Fig. 19-1*b*), in which a flash groove has been provided to receive the excess material. These are generally employed with rubber compounds. Although a flash mold is also closed by a hot hydraulic press, the real pressure develops later when the compound heats up and tries to force the mold apart by virtue of its high rate of thermal expansion. Blanks for flash molding need conform only roughly to the shape and size of the mold.

c. "Transfer" molds, in which a cylindrical slug of rubber compound is forced into the surrounding mold cavities through tiny gate openings (Fig. 19-1*c*).

d. "Injection" molds (Fig. 19-1*d*), in which the mold cavities are recharged on each cycle by an independent screw extrusion machine. Injection molding is adapted only to small rubber or plastic parts in mass production, for which reason it has not found much application in the packing industry.

Molding is regularly employed for the preparation of packings of odd shapes not readily produced by other means; it may, however, be employed for the production of simple rings and disks where good finish or accuracy is wanted and where the number of pieces to be produced justifies the initial cost of the mold.

2. The unvulcanized rubber compound is applied to fabric by a calender, the operation being known as "frictioning" when the compound is driven into the interstices or as "skimming" when it is merely laid on the surface. Cloth-inserted gasketing or diaphragm sheet is built up by successive frictioning and skimming operations and vulcanized by method 1*a* or 1*b* above. Frictioned cloth may also be used in the preparation of slab and rolled packings or incorporated into molded packings.

3. The rubber compound may also be extruded by a worm extrusion machine as strip of any cross section which may be (*a*) cured in steam for the production of strip gasketing, (*b*) in the case of hollow cross sections, mounted on mandrels and cured as sleeves, or (*c*) cut into blanks for molding.

4. The rubber compound may also be made into solution in gasoline, which may be (*a*) applied to fabric by painting or dipping, the resultant coated cloth being used in exactly the same manner as frictioned cloth under 2; (*b*) applied to braid, metal foil, paper, etc., to fill the interstices; (*c*) used for joining the component parts of

molding blanks; (*d*) used for the preparation of compounds of higher filler content than is possible by dry mixing alone. In this category come compressed-asbestos sheet and rubber-bonded plastic packings whose manufacture is peculiar to the packing industry and deserves special description.

In the manufacture of compressed asbestos sheet, a small amount of rubber solution is placed in a dough mixer, and a large volume of asbestos fiber is kneaded into this, the operation requiring several hours. The resultant dough is fed to a special two-roll calender, known as a "sheeter" (Fig. 19-2), in which the larger roll is steam-heated while the smaller roll is water-cooled. The dough adheres to the hot roll in the form of a thin film and dries in the course of a single revolution, the solvent fumes being exhausted by appropriate ducts. At each revolution, the rolls are separated a little farther by worm gearing, thereby permitting a continuous build-up of dried sheet on the hot roll. When the correct thickness has been attained, the machine is stopped, and the finished sheet is removed as a large square, usually 10 ft on a side. This is press-cured or is calendered and then oven-cured.

The manufacture of rubber-bonded "plastic" rod packing is similar except that, in addition to fiber, it is customary to incorporate a variety of other materials such as flake graphite and particles of metal. The product is discharged from the mixer as a wet crumb and is either dried and sold as such or is fed to a worm extrusion machine from which it issues as a more or less coherent coil, which is sometimes further strengthened by the application of a braid of cotton or asbestos yarn. Other binders can be substituted for the rubber solution.

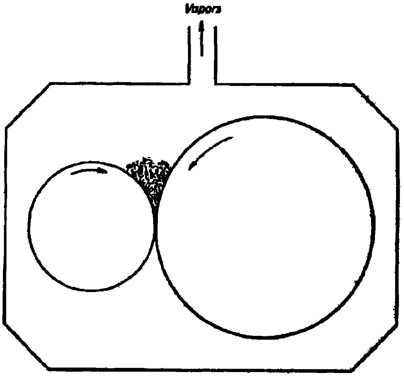


FIG. 19-2. Two-roll calender (known as "sheeter").

Gutta Percha and Balata. Gutta percha and balata are vegetable gums similar to natural rubber but, being incapable of vulcanization, have found little application in packings.

Synthetic Rubber—Styrene Type. Synthetic rubber—styrene type, known in Europe as buna S and in this country as GR-S, was the rubber available in the greatest quantities during the Second World War. It is a copolymer of two light hydrocarbons, butadiene and styrene, and resembles natural rubber closely. It employs the same compounding ingredients and fabrication methods and vulcanizes in about the same manner.

In general, GR-S compounds are physically weaker than natural-rubber equivalents but more resistant to the degrading effect of heat. For the latter reason, GR-S will probably continue to find important packing applications even if crude rubber is available in quantity.

Synthetic Rubber—Acrylonitrile Type. Synthetic rubber—acrylonitrile type, known in Europe as buna N and in this country under various proprietary names such as Perbunan, Hycar, Butaprene, and Chemigum, is similar to GR-S, substituting acrylonitrile for styrene in the molecule. It also is compounded, fabricated, and vulcanized like natural rubber but, being somewhat intractable, does not lend itself to all operations. The vulcanizates have fair physical and heat-resistant properties and are characterized by extraordinary resistance to the swelling action of ordinary petroleum oils. Toward benzol, carbon tetrachloride, etc., their superiority over natural rubber is not so pronounced and toward acetone is inferior.

Synthetic Rubber—Chloroprene Type. Synthetic rubber—chloroprene type, known as neoprene, is a polymer of chlor-butadiene. It likewise can be compounded, fabricated, and vulcanized like natural rubber but, unlike it, requires no sulfur to effect vulcanization. It is easier to handle than the acrylo-rubbers, and the products are stronger, although not quite so good in respect to oil or heat resistance.

Synthetic Rubber—Thio Types. Synthetic rubbers—thio types, marketed under the generic name "Thiokol," are believed to be organic polysulfides. They also can be compounded, fabricated, and vulcanized, although with more or less difficulty. Only metal oxides are needed to effect vulcanization. They are physically inferior to the other synthetics, more thermoplastic, less heat resistant, and characterized to varying degrees by strong onionlike odors. In resistance to the swelling action of benzol, chlorinated solvents, etc., they outrank the other synthetics.

Synthetic Rubber—Isobutylene Type. Synthetic rubber—isobutylene type, sold as butyl rubber, is a copolymer of isobutylene with isoprene, both hydrocarbons. Butyl is compounded, fabricated, and vulcanized like natural rubber; the vulcanizates are physically inferior and lacking in any appreciable solvent resistance but are characterized by appreciable resistance to many troublesome chemicals, such as nitric acid. Principal developed packing use is for acid gaskets.

Reclaimed Rubber. Reclaimed rubber, or more properly, reclaimed rubber compound, is vulcanized rubber or GR-S compound, mostly from used tires, which has been rendered plastic by drastic treatment with heat and plasticizers. Neither the combined sulfur nor the fillers are removed by this treatment, but the compound is rendered capable of processing and of revulcanizing with added sulfur. Reclaim is inferior to compounds made from new rubber or GR-S but is employed in certain types of packing, replacing new rubber wholly or in part.

Hydrocarbon Plastics. The hydrocarbon plastics include hydrocarbon polymers incapable of vulcanization. In this group are polystyrene (including high-styrene copolymers with butadiene), polyethylene (Polythene), and polyisobutylene (Vistanex). The last is processed like rubber, the first two by powder-molding or injection-molding. Owing to their thermoplasticity, the members of this group have only limited packing application in spite of their extraordinary resistance to corrosive chemicals such as nitric acid. Polyisobutylene confers high viscosity on mineral waxes, and such mixtures (Tervans) are used as binders in braids for high-temperature service, especially braided blue asbestos used for acid-pump packing.

Closely related to polyethylene is polymerized tetrafluorethylene, known as "Teflon." Although thermoplastic, the softening point of this material is high enough to permit its use as packing on all but the highest temperature, and its complete chemical inertness and resistance to all known solvents recommend it for chemical gasketing. Its principal drawbacks have been cost and manufacturing difficulties, since the only fabricating methods developed to date are sintering from powder, machining from the solid sheet or block, or incorporating, in granulated form, in a plastic.

Cellulose Plastics. The cellulose plastics, cellulose acetate, cellulose nitrate, ethyl cellulose (ethyl rubber), are thermoplastic, which has limited their packing application.

Vinyl Plastics. Vinyl plastics include polyvinyl chloride (Koroseal, Tygon), polyvinyl chloroacetate (Vinylite), polyvinyl butyral, polyvinylidene chloride (Saran), and polyvinyl alcohol (Resistoflex). Most of these can be plasticized to a rubber consistency and fabricated like rubber except for vulcanization. Although limited by thermoplasticity, they have found certain packing applications for cold service in the form of sheet, cut and extruded gaskets, and molded rod packings. The closely related acrylic plastics, familiar as Plexiglas, Lucite, etc., have, as far as is known, not been employed in packings.

Polyesters. The polyesters, which include the well-known "alkyd" varnishes, etc., include also a few rubberlike types (Agripol, Norepol, Paraplex, etc.) which have been employed in packings alone or in combination with synthetic rubbers.

Thermosetting Plastics. The thermosetting plastics include the phenolic, urea, melamine, etc., types. These become completely infusible and insoluble with heat and are acquiring increasing importance in packings, subject to the limitation that they are exceedingly hard. Principal forms are plastic masses containing large ratios of asbestos fiber which, after molding, are used for high-temperature valve disks, etc., and laminated slab built up from resin-coated paper or cloth. From the latter are cut high-temperature valve disks, piston rings for oil pumps, and axial-seal rings. Mixtures of phenolic resins with synthetic rubbers are known also.

Polysilicones. The polysilicones are the most recent and, in some respects, the most interesting of the plastics. They are available in forms ranging from thin liquids through rubberlike materials to hard resins. All are characterized by resistance to atmospheric oxidation and slow change of consistency with temperature. Up to date the high cost and physical weakness of the silicone rubber has somewhat limited its packing application, but these disadvantages will probably be overcome. During the Second World War silicone-rubber gaskets were used in searchlights and turbosuperchargers.

Polyamides. The polyamides include nylon which may have interesting packing possibilities. It has been available for too short a period to permit a proper evaluation.

Proteins are represented in the packing industry by gelatin which, in combination with glycerine, forms the binder in most paper gasketing.

Petroleum Derivatives. Petroleum derivatives such as mineral oils, waxes, and asphalts are used as plasticizers in rubber compounds and as binders in braided, plastic, and leather packings.

Coal Derivatives. Coal derivatives such as coal-tar oils, pitch, and cumarone resins are employed as rubber plasticizers.

Wood Derivatives. Wood derivatives such as rosin, pine tar, pine oil, and shellac are used as plasticizers in rubber compounds. Shellac is a binder in some metal-foil packings.

Simple Esters. Simple esters such as dibutyl phthalate, etc., are used as plasticizers for synthetic rubber and other synthetic polymers.

Fatty Oils and Waxes. Fatty oils and waxes and their derivatives are extensively used in the packing industry. Palm oil, cottonseed oil, stearic acid, and combinations of fatty oils with sulfur (factices) are used as rubber plasticizers. Tallow, raw, hydrogenated, and blown castor, soybean and fish oils, Japan wax, and carnauba wax are used in braided, plastic, and leather packings. Oxidized linseed oil applied to cloth (oilcloth) has been used for pump diaphragms. Glycerine is employed with gelatin in paper gasketing, with soap in certain braided packings, and alone as a surface treatment for rubber–asbestos rod packings.

Graphite. Graphite in powdered form is a common surface treatment for rod packings where it functions as a "surface leveler." Because of reputed corrosive effects, it has been partially replaced by talc and mica, which, however, are not so satisfactory as graphite in other respects.

Flax. Flax, in the form of braid, is the oldest and still perhaps the most widely used packing for water-pump rods. The property which flax has of swelling in water is important in this connection. Flax consists of, of course, the bast fibers of the flax plant from which the softer tissues have been removed by retting in water. It is received in this country in the form of tangled hanks which are combed in a special machine to separate the long fibers from the short fiber, or "tow." The long fibers are fed, end to end, into a drawing frame, and the resultant sliver is transferred to a spinning frame from which it issues as "long-line" flax roving. The tow, in turn, is put through a carding machine and spinning frame to form a second grade of flax roving. Both grades are used in the production of braided packings.

Although the braiding operation is employed in other industries, its use in the packing industry is sufficiently important to warrant a digression. Braiders are of three general types: (1) eight carrier, (2) multiple carrier, and (3) interlocking. The resultant braids are shown in Figs. 19-28*a*, *b*, and *c*. In the first two types, the spools carrying the yarns are in two groups, one group pursuing a clockwise orbit and the other a counterclockwise orbit, the orbits intersecting in the manner of a Maypole dance. In the eight-carrier braider, it is customary to produce the finished cross section in a single pass through the machine; the spools are wound with multiple-twisted yarns of sufficient diameter to accomplish this purpose. The product is square in cross section, but the squareness is usually accentuated by introducing straight yarns at four points in the circumference; these yarns are known as "corners" or "warps."

Other straight yarns are sometimes introduced into the center to control the finished diameter. These are known as "centers." The eight-carrier braider is usually employed in braiding flax yarn. Wherefore, this type of braid is sometimes known as a "flax braid."

Multiple-carrier braiders are available in 12, 16, 20, 24, or higher number of carriers. The product is normally polygonal in cross section, *viz.*, 6-sided for a 12-carrier braider, 8-sided for a 16-carrier braider, etc., but can be made to approximate a square cross section by the introduction of corner yarns at appropriate points. Centers are commonly used in multiple-carrier braids and may be large in relation to the cross section; also they are frequently of a dissimilar material, such as rubber or metal. Except where a center of large diameter is used, it is customary to build up the diameter of multiple braids by successive passes through the machine, the product being sometimes described as "braid-over-braid."

Interlocking braiders are basically similar to multiple-carrier braiders but have extra carriers which crisscross back and forth in the cross section, locking the structure into a solid mass. The advantage of interlocking braid is that the structure does not fall apart when the outer layer of yarns is worn through.

A braid is described by the number of braids, the number of carriers in each, yarns per carrier, size of yarn, presence or absence of corners and centers, and the "pitch," which is the axial length of braid for one complete revolution of any one carrier. The latter is controlled by gearing on the take-off capstan and is usually adjusted to equal the braid circumference, thereby giving all yarns except the corners and centers a uniform 45-deg angle to the braid axis; this gives a satisfactory balance between flexibility and stretch.

To be useful as a packing, a braid must generally be filled with some sort of binder. This binder is sometimes incorrectly described as a "lubricant," but its real function is to stop leakage through the body of the braid and to minimize leakage along the faces by acting as a surface leveler. In the case of square-braided flax, the binder is usually tallow or similar animal fat, or a mixture of same with paraffin, and is applied by soaking the finished braid in the molten binder, the excess being removed by draining or by passing the braid through a wiper or close-fitting die. Graphite is sometimes applied to the surface to assist the surface leveling.

Cotton Yarn. Cotton yarn, in the form of braid or woven cloth, is a common ingredient of packings for low-temperature service. The principal fabrics used in fabric-rubber packings are sheetings, drills, Osnaburgs, hose, belting, and tire and tent ducks. Raw cotton also is necessary in commercial asbestos yarn to carry the fiber through the carding process. Braided cotton is competitive with braided flax and braided asbestos.

Other Vegetable Fibers. Jute yarn is used to replace flax wholly or in part in braided packings; service life is believed to be somewhat lowered, but there is some saving in cost. As far as is known, no packing use is made of either domestic hemp, Manila hemp, or sisal, although flax or jute packings are sometimes erroneously referred to as "hemp." Ramie fiber makes a very durable packing which has been used as a stern-gland packing on vessels.

Rayon. Both the filament and staple types of viscose rayon have been used recently in braids with excellent results. Staple rayon also replaces cotton in asbestos-yarn mixes.

Other Synthetic Fibers. Both nylon and vinyl acetate (Vinyon) yarns are being employed experimentally in braided and fabric packings.

Asbestos. Two forms of asbestos are employed in packings. Chrysotile is a hydrated magnesium silicate found extensively in Canada, Russia, Rhodesia, and elsewhere. It appears to have been deposited from solution in cracks in the rock in the presence of considerable pressure and temperature, under which conditions it formed elongated crystals at right angles to the crack. The width of the crack therefore determines the fiber length. Originally, only the widest veins were saved, and the rock was carefully chipped away from these with hammers, the product being sold as "No. 1 crude" or "No. 2 crude," according to fiber length. This expensive process has been largely superseded by milling, wherein the entire product of the mine is crushed by successive passages through roll mills, each passage being followed by treatment on a vibrating screen equipped with a suction hood at the lower end, the liberated fiber being collected in cyclones. Milled fiber is rated, as shown in Table 19-1, according to the percentages remaining on standard shaking screens which were developed at the Canadian mines.

Table 19-1. Standard Classification Canadian Milled Asbestos Fiber

Group No.	Min residues, oz per lb, on			
	First screen	Second screen	Third screen	Pan
3 (spinning fibers)				
3F	7	7	1.5	0.5
3K	4	7	4	1
3R	2	8	4	2
3T	1	9	4	2
3Z	0	8	6	2
4 (shingle fibers)				
4H	0	5	8	3
4K	0	4	9	3
4M	0	4	8	4
4R	0	3	9	4
4T	0	2	10	4
4Z	0	1.5	9.5	5
5 (paper fibers)				
5D	0	0.5	10.5	5
5K	0	0	12	4
5M	0	0	11	5
5R	0	0	10	6

Grades such as 3F and 3K are only slightly inferior to the hand-picked crudes; they, however, are becoming increasingly scarce as the highest grade ore becomes exhausted and hence are reserved for the finest yarns. Grades 3R, 3T, and 3Z are employed for ordinary yarns. The No. 4 grades are designated as "shingle" grades. Various No. 4 grades are used in the manufacture of compressed-asbestos sheet and other fiber-containing rubber formulas, alone or in conjunction with No. 3 grades. The No. 5 grades are designated as "paper" stock and are employed in the manufacture of asbestos paper which is used as a packing itself and as a component of the various corrugated, jacketed, and spiral-wound metal gaskets.

The manufacture of asbestos yarn begins with a hammer mill designated variously as a "willow" or "opener." This breaks up the clots and eliminates sand either by air separation or screening. If crude fiber is used, it is first crushed on a Chile mill before being fed to the opener. The cleaned and opened fiber is then blended with a certain percentage of cotton in a device known as a "mixing picker." The purpose of the cotton is to carry the fiber through the subsequent carding operation, the longer fibers requiring the least cotton. A 100 per cent asbestos yarn can be made from selected crudes but is very expensive and is reserved for use in oxygen packings where the presence of vegetable fiber would be hazardous. Grade AAAA yarn contains less than 10 per cent cotton and is specified in special packings. Grade AAA (15 per cent cotton maximum) and Grade AA, or "underwriters" grade (20 per cent maximum), are used in packings; Grade A or "commercial" (25 per cent maximum) was used during the Second World War but is inferior.

Carding is common to most textile operations and consists in passage of the fiber over a drum lined with "card clothing," which resembles Turkish toweling but is made of wire. The action of the drum and adjacent rolls is to spread the fiber out into a film, or "lap," of almost gossamer fineness. This is broken up into bands, each of which passes between rubbing leathers to issue as a continuous but weak "roving." The rovings are fed to a spinning frame which imparts a twist. The resultant "single" yarn is plied up on a similar but larger machine, and at the same time one or more fine wires (copper or brass) may be twisted into it to give added strength. Yarn is identified by some such designation as "1021" which implies that the single yarn is "10 cut," *i.e.*, ten 100-yard lengths to the pound. There are two plies and one wire.

The preparation of cloth from yarn is a simple weaving operation, common to all branches of textile manufacture. Braiding follows the methods described under flax. Asbestos braids are quite generally saturated with a soft binder such as petrolatum, and inasmuch as this results in a binder content too great for certain services, alternative methods are employed wherein the yarn is pretreated before braiding with controlled amounts of binder, or treated and untreated yarns are mixed in the cross section. Graphite is quite generally applied as a surface leveler and may, in addition, be incorporated throughout the cross section, giving what is known as a "graphited-throughout" construction which is believed to have advantages for certain services.

Blue or Crocidolite Asbestos. Blue or crocidolite asbestos is found principally in the Union of South Africa. Chemically it is an amphibole, with a high iron content to which it owes its blue color and much less water of hydration than chrysotile. It appears on the market as fiberized crude, as twisted rovings for calking joints, and as spun yarns and cloth, generally made without cotton. The fiber is harsher than chrysotile and the yarn somewhat uneven and bulky. However, it is much more resistant than crysotile to the action of strong minerals acids and alkalies and, hence, preferred for chemical applications.

Glass Fiber. Glass fiber is a modern product which has been employed in packings in the form of felt, yarn, cloth, and braid. Since it is chemically resistant, it has competed somewhat with blue asbestos. Although very strong in tension, glass fiber is somewhat inelastic and tends to crumble under tightening loads. Also, in the form of braid, it is somewhat lacking in power to retain binder. As the art of fiber making improves, glass fiber will become increasingly important in packings.

Wool in the Form of Felt. Wool in the form of felt has been widely used as dust seals and also in the manufacture of oil seals, using a synthetic-rubber binder, or as a gasket with a surface coating of synthetic rubber.

Paper. Paper from waste rope or kraft pulp, and purposely made porous, is a widely used gasket material. It is saturated with glycerin–gelatin or with a natural- or synthetic-rubber solution. By the incorporation of granulated cork in the beater, the porosity and compressibility are further increased. Asbestos paper has been mentioned.

Cork. Cork, in the form of the cork stopper, constitutes one of the most widely used "gaskets." In the form of corkboard, made from cork granules with a binder of glue, synthetic rubber, or plastic, it is used in bottle caps and for many other gasketing applications.

Leather. Leather is one of the oldest packing materials and is still widely used, especially for heavy-duty hydraulic rams and plungers. Small, cheap cup packings, such as those used in Flit guns, are often made from leather shoe trimmings.

For hydraulic leathers all tannages are used, but the favorite, perhaps, is a combination of vegetable and chrome tannage. Blanks are cooked in wax and afterwards hot pressed to cup or V form along the lines of rubber molding, except that there is no vulcanization, but the mold is cooled before discharging the product to set the wax and fix the shape of the article. Binders other than wax are also employed. Thin sheepskin leather is employed as a diaphragm, especially for gas meters.

Wood. Wood in solid form has been employed as a gasket material, this in addition to its use as a source of wood pulp for paper and of ground wood as a filler for rubber or plastics.

GASKETS

A gasket will stop leakage if it is exerting against the confining members a unit compressive force greater than that of the fluid pressure. Mere contact is not enough, and adhesion cannot normally be depended upon. It follows that any gasketed joint should embody means for putting the gasket under this state of compression and keeping it there for the expected life of the gasket between retightenings.

The most widely used type of joint is the flange joint, shown in various modifications in Fig. 19-3. In this joint, the bolts or other tightening means have the following functions to perform:

1. Support stresses in the container walls caused by the fluid load acting across an area equal to the inside area of the gasket.
2. Support any additional stresses in the container walls caused by gravity, misalignment, or thermal expansion and contraction. These, however, should be eliminated as far as possible by suitable design.
3. Develop a unit gasket load over all portions of the gasket area equal to the

fluid pressure times a safety factor* sufficient to take care of natural relaxation of the gasket during the interval between tightenings.†

4. Develop additional load over those portions of the gasket area first contacted to depress these sufficiently to permit development of the safe unit gasket load over the rest of the area. The necessity for this arises from irregularities in the gasket or in the flange faces. Irregularities in the flange faces may be present originally or induced by flange deflection under bolting load.‡

Under Materials were mentioned a large number of gasket types suitable for the bolted flange joint. Some of the more important ones are summarized in Table 19-2. Although the number is large, they are by no means equally important. It seems probable that plain-rubber or GR-S gaskets pack 90 per cent of all flange joints operating against gases, water, or aqueous solutions below 220°F. Vegetable-fiber paper and oil-resistant synthetics occupy a similar role against oils and solvents below 220°F, and compressed-asbestos sheet a similar role for all services between 220 and 500°F.

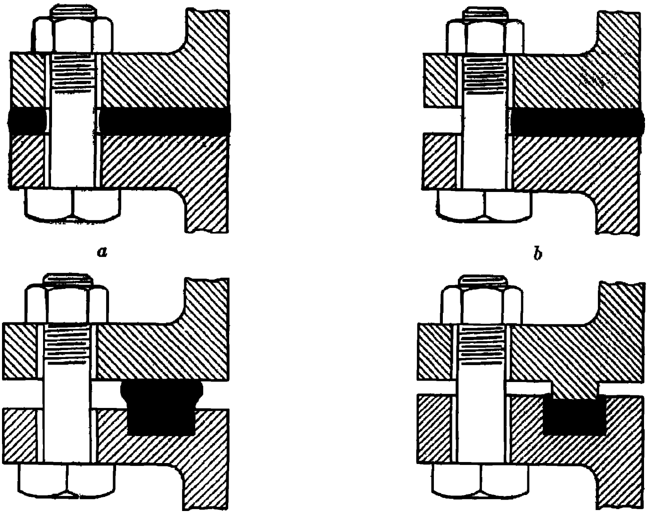


Fig. 19-3. Types of flange joints.

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* This safety factor is known as the "contact-pressure ratio" and was designated by the British Institution of Mechanical Engineers in their First Report of the Pipe Flange Research Committee, 1936, as e_3. A somewhat similar factor m appears in the ASME formulas for Bolted Flanges (ASME Code for Unfired Pressure Vessels, 1943, Section UA-20) except that m is 0.5 lower than e_3 owing to the assumption in the American formulas that fluid load is effective to the middle of the gasket area instead of to its inside edge.

† The amount of this relaxation is usually underestimated. For instance, a high-grade synthetic-rubber gasket in $\frac{1}{16}$-in. thickness will lose 60 per cent of an initial 2,000-psi gasket load between smooth flanges after 20 hr at 212°F. It follows that e_3 must be at least 2.5 (m at least 2.0) if the gasket is to survive even this mild service exposure. Thick gaskets poorly confined, and higher temperatures further increase the rate of relaxation and the consequent need for a high safety factor unless frequent retightening is resorted to. Substitution of other types, such as compressed-asbestos sheet, reduces but by no means eliminates relaxation. On the other hand, thinner gaskets (where flange accuracy permits), roughened flange surfaces, and long stretchy bolts capable of exerting a "follow-up" effect reduce relaxation and reduce the need for high safety factors or frequent retightening.

‡ The ASME formulas recognize the need for a minimum gasket load y which is defined as the "gasket or joint contact-surface unit seating load." Unfortunately, this is treated as a property of the gasket, whereas, at least for soft gasketing, it is almost wholly a measure of the inaccuracy of the joint and will vary with each joint.

The need for flange accuracy will become apparent when it is realized that the total compressibility of a $\frac{1}{8}$-in. compressed-asbestos gasket is about 0.020 in., *i.e.*, if the cumulative deviation of the two [illegible]

[illegible]

a l[illegible]r extent to even the more compressible gasket types.

Among the fabric-inserted gaskets, the cotton-inserted type is preferred to plain rubber by some users because the material holds its shape better while being cut—the insertion has no other function. Metal-wire insertion, however, has some merit in restraining lateral flow on gaskets subjected to heavy bolting loads as in hydraulic lines. Flat asbestos-fabric gaskets have been largely superseded by compressed-asbestos sheet, but folded asbestos-fabric gaskets are still widely used where a gasket more compressible than compressed-asbestos sheet and involving less waste in manufacture is needed. Such applications include boiler handholes and manholes, large reaction vessels, column stills, etc.

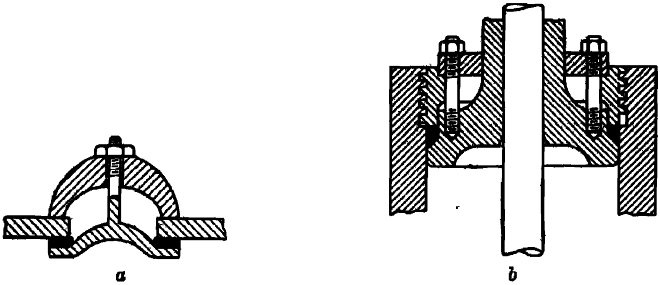

Fig. 19-4. Self-tightening flange joints.

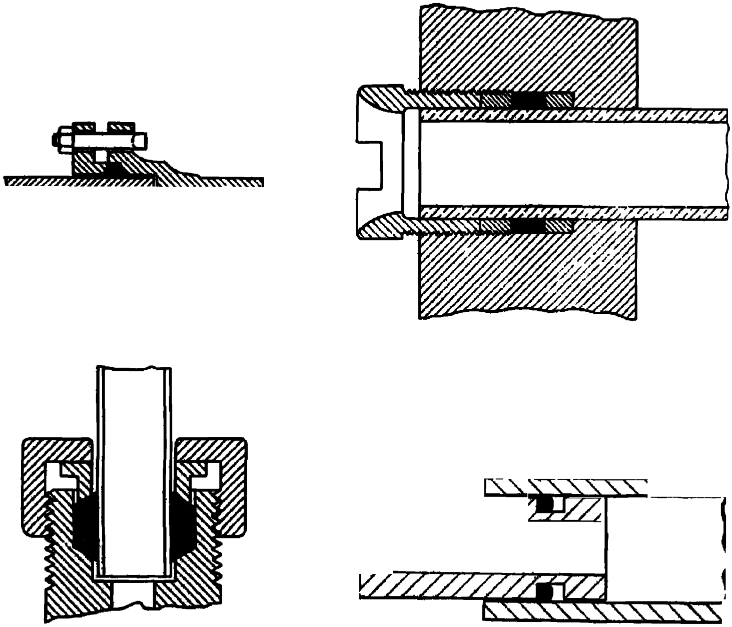

Fig. 19-5. Concentric joints.

Leather gaskets are tough and are often used where surfaces rotate at the time of closing, under which conditions a rubber gasket would be injured. Braided-asbestos gaskets, are used for the same services as asbestos-fabric gaskets, especially in locations such as furnace doors where destruction of the rubber binder by oxidation would cause a fabric gasket to open at the seams. Silicone rubber in conjunction with asbestos braid makes an interesting product for such applications. Braided flax has been used for hatch gaskets on merchant vessels but is probably inferior to rubber.

Metal gaskets, because of their cost, have largely been limited to severe temperature and pressure applications. Polyvinyl chloride and Teflon are employed principally where their chemical inertness is of value.

Flange joints can be made so that they are self-tightening under fluid pressure, as in the case of the boiler handhole (Fig. 19-4*a*) and valve-bonnet joints (Fig. 19-4*b*). Since any danger of failure through relaxation is avoided with these designs, it is rather surprising that they are not more widely used.

In another class of gasketed joint, the surfaces to be packed are concentric and parallel. These joints, except for absence of relative motion, correspond exactly to sliding-contact joints and have the great advantage that the axial spacing of the parts to be joined need not be determined in advance. Examples are the cast-iron pipe joint (Fig. 19-5*a*), the condenser tube-sheet ferrule (Fig. 19-5*b*), the gauge-glass gasket (Fig. 19-5*c*), and the "O-ring" gasket (Fig. 19-5*d*). Except for the last type, which is self-tightening at all loads provided the ring has been given an initial squeeze, concentric gaskets require the application of bolt load equal to the fluid load times an appropriate safety factor to take care of gasket relaxation. Bolts, however, are not required to support stresses in the container walls, for which independent provision must be made. Concentric gaskets are usually either plain rubber or rolled fabric.

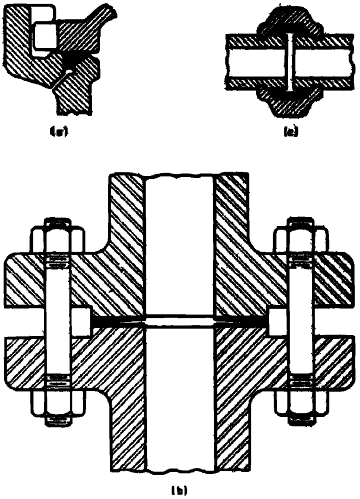

FIG. 19-6. Types of gasket joints.

In still another type of gasketed joint, the gaskets are made cup-shaped to admit the fluid pressure to the interior and thereby render the gasket self-tightening. Such joints can be of either the flange or concentric types. The vulcanizer joint (Fig. 19-6*a*), hollow metal gasket joint (Fig. 19-6*b*), and pipe joint (Fig. 19-6*c*) belong to the former category. It has been noted above that the O-ring design, although not cup-shaped, is also self-tightening. All self-tightening gaskets, to be effective, must be carefully designed and molded to extreme accuracy.

Since a gasket, except a cup gasket, is exposed only on the edge to the fluid being packed, its problems are primarily mechanical rather than chemical. It is possible, however, for a gasket to fail by chemical attack. In prescribing gaskets for various fluids, therefore, it is necessary to take both mechanical and chemical conditions into account. A list of recommendations for flange gasketing for various services is given in Tables 19-3 and 19-4. Similar recommendations for the various "specialty" joints (Figs. 19-5 and 19-6) can, in some instances, be secured from the packing manufacturer.

VALVE DISKS AND VALVE SEATS

Pump Valves. The standard check valve of a reciprocating pump (Fig. 19-7*a*) consists of a bronze grid which screws into the valve deck, a center post, and a rubber disk which rides up and down on the post under the influence of the escaping water and a helical spring. The design is not very good, since the rubber valve tends to take the imprint of the grid, and when the valve later rotates slightly, it cannot seat and must be refaced. Better results are secured by recessed ribs or an overhead suspension—if the soft valve cannot support itself over the larger opening, it can be contained in a metal disk holder (Fig. 19-7*b*).

Other common faults of the grid arrangement are (1) that the valve tends occasionally to be dragged sideways by the escaping water and wears the center hole elliptical. In the absence of a metal disk holder, as in Fig. 19-7*b*, this can be partially corrected by purchasing valve disks with a molded-in bushing of metal or other material; (2) that the edges of the ribs are frequently ground with a razorlike edge which cuts the disk—these should be filed off before use.

Rubber pump valves range in hardness from pure gum (used for condenser vacuum pumps) to ebonite (used for boiler-feed and other hot-water pumps). Other materials used are leather, laminated phenolic resin, and even metal, in the form of superimposed thin stampings (Fig. 19-7*c*). Similar valves used in reciprocating air compressors are almost invariably of polished sheet steel, although seme small compressors such as those used in garages have rubber check valves.

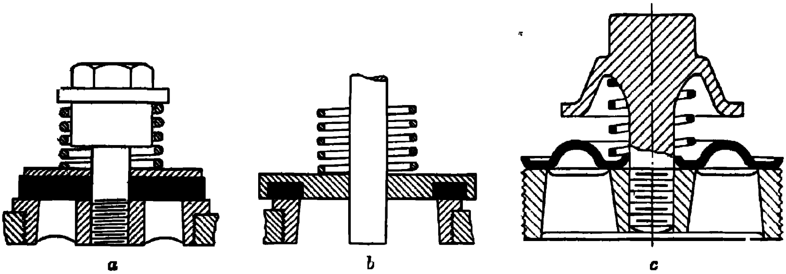

FIG. 19-7. Pump valves.

Valve Seats. Valve seats take the place of valves in some of the pot-valve designs used on heavy-duty pumps such as those which pump drilling mud (Fig. 19-8). Ball valves are also used in pumps; in some cases the ball is rubber; in other cases it is metal engaging a rubber valve seat.

Line Check Valves. Line check valves are similar in general to pump valves but are usually contained in a disk holder. Ball checks are also used.

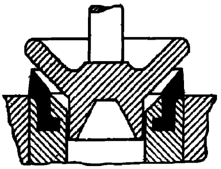

FIG. 19-8. Valve seats.

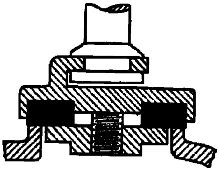

FIG. 19-9. Disk holder.

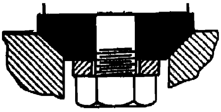

FIG. 19-10. Hydrant valve.

Disks for Globe and Angle Valves. Disks for globe and angle valves are contained in disk holders (Fig. 19-9) which have either round or oval center pins and are swiveled to prevent rotation of the disk when closed. Disks are usually hard rubber with asbestos fiber present as reinforcement to the extent of from 10 to 70 per cent. They are employed on steam service up to 150 psi, but at the higher pressure range tend to undergo too much indentation by the valve seat, with consequent cracking and scouring out by escaping steam. Laminated or solid phenolic resins are longer lived but require a more accurate finish in the valve seat. Metal-clad disks are also used.

Faucet Bib Washers. Faucet bib washers are similar to globe-valve disks but softer, *i.e.*, in the semiebonite range. A good faucet, like a good globe valve, will have its disk holder swiveled so that the disk does not rotate as it makes contact. Hydrant valves (Fig. 19-10) are modified globe valves; they employ beveled disks of leather, semihard rubber, or duck-inserted rubber.

Flap Valves (Fig. 19-11). Flap valves cut from leather or rubber are also used as check valves. Some soft pump valves of conventional design are employed as flap valves by holding them down in the center and forcing the edges to lift. A disadvantage of flap valves is that they fail by fatigue cracking.

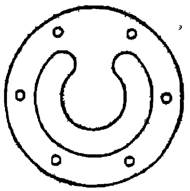

Fig. 19-11. Flap valve.

SLIDING-CONTACT PACKINGS—GENERAL

Sliding-contact packings differ in several important respects from gaskets.

1. They do not make intimate contact with the sliding member, since such contact would lead to destruction of one or both surfaces. A badly fitting packing makes dry contact over a portion of its area, leading initially to high friction and wear, but this condition tends to correct itself as the high spots wear away.

2. Over all that portion of the interface which is not in dry contact there is a clearance which is filled with the fluid being packed. This fluid is constantly being lost on the low-pressure side and replenished on the high-pressure side. There is no such thing as a "leakless" sliding-contact packing, although the leakage under certain conditions may be so low that it is not detected.*

3. In an ideal packing, no portion of which is in dry contact, leakage conforms to the laws of fluid flow, which means that it is proportional to pressure differential and to the width of the escape path (*i.e.*, circumference in the case of a shaft or rod packing), inversely proportional to fluid viscosity and to the length of escape path (*i.e.*, packing depth), and proportional to the third power of the clearance. In a badly fitting packing the clearance is reduced to a series of channels, but leakage is relatively high owing to the thickness of these channels.

4. Relative motion has little influence on leakage provided it does not cause a change in clearance. However, if a rotary shaft at high speed is subject to gyration (generally known as "runout"), it will batter an opening through the packing, which is larger than itself, and the fluid leakage may under such conditions become uncontrollable. Similar effects result from a reciprocating shaft subject to lateral thrust (generally known as "weave") or, in the case of axial seals, a tipping of the plane of one ring with reference to that of the other ring, provided these movements occur at a rate too rapid for the packing to respond.

5. In an ideal packing friction conforms to the laws of fluid shear, *viz.*, proportionate to both packing width (*i.e.*, circumference) and depth, to viscosity, to speed of relative motion (in feet per minute), and inversely proportional to clearance. The materials of which the packing and sliding members are composed are without influence. It will be noted that three of the factors that make for high friction, *viz.*, packing depth, viscosity, and low clearance, are also included in those which make for low leakage. As long, therefore, as circumference and speed are fixed, there is an inverse relationship between leakage and friction.

In the badly fitting packing, the laws of solid friction apply over a portion of the area, *viz.*, proportionate to the area that is in dry contact, to unit load over this area, and to the coefficient of friction between the material of the packing and the sliding member. Speed and fluid viscosity are without influence on this portion of the total friction. In general, the friction of a badly fitting packing is much higher than that of an ideal packing owing to the high unit loads employed in an effort to depress the high spots and reduce the leakage channels.

6. In the case of an ideal packing, the pressure of packing against the sliding member need be enough only to overcome the pressure of the fluid in the clearance, which averages midway between the high-side and low-side pressures. Thus, in an ideal packing that is leaking to the atmosphere, a unit load equal to half the fluid

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* It has been recently brought out that certain combinations of fluid viscosities, clearances, and surface speeds give rise to vortices in the contained fluid which are "leakless" against low-pressure differentials. This does not detract from the general statement.

pressure is sufficient to keep the packing from blowing away. This, it will be noted, is just half the minimum required to prevent leakage, *i.e.*, to make the packing function as a gasket. In practice, it is not possible to avoid the application of some excess pressure, which leads to development of dry contact over a portion of the area, and the packing is no longer "ideal." The closer it is possible to maintain the packing load at the theoretical 50 per cent value, the more nearly do ideal conditions obtain.

In the case of a badly fitting packing, high unit loads are necessary initially to reduce leakage channels. The amount of such necessary excess pressure is governed by the initial roughness and by the ability of the packing material to smooth up under applied pressure. As the packing wears in to a condition approaching ideal, such excess loads can be dispensed with.

7. The heat developed within a packing is proportional to the product of friction and surface speed. The temperature of the packing rises until the rate of heat dissipation equals the rate of heat generation. Temperature can be held down by improved means of heat dissipation, such as water-cooled shafts, water-jacketed stuffing boxes, etc., but the most effective way to reduce temperature is to adjust for a higher leakage rate and a correspondingly lower friction.

High temperature in a packing may lead to excessive thermal expansion with further increase in load and ultimate seizure. If the packing is so constructed that this thermal expansion does not occur or does no harm, temperature will eventually reach a ceiling owing to lowered viscosity of the escaping fluid and consequently lower friction.

From the foregoing, it is clear that the two major requirements of a successful sliding-contact packing are (1) that its working face should conform as accurately as possible to that of the sliding member or should acquire this condition within a reasonable time after starting and (2) that means should exist for developing the unit load normal to the sliding surface which is necessary to control clearance. Other requirements are (3) that it should be composed of material that will not damage the sliding member during the process of wearing in to a fit, (4) that it should be capable of withstanding without injury the heat resulting from packing friction, and (5) that it should not be attacked by or contaminate the fluid being packed. A summary of the most widely used sliding-contact packings is shown in Tables 19-5 and 19-8.

The stuffing box is the oldest known means of controlling packing load and has heretofore, until very recently, controlled the development of sliding-contact packings. In the stuffing box, the load is applied axially by advancing the gland; the packing must, therefore, be of a type that will transmit the load at right angles, *i.e.*, must be capable of spreading radially. Originally this limited stuffing-box packings to soft materials such as braided yarns and rubber compounds, and these were supplied generally in square cross section and in the form of coils, helices (incorrectly known as "spiral"), and cut rings. However, the development of molded conical (Fig. 19-26*d*) and V-ring (Fig. 19-26*c*) cross sections has made it possible to employ a number of more rigid materials in stuffing boxes. In the conical design, lateral spreading is effected by the principle of the inclined plane, in the V ring by that of the toggle joint.

Contrary to popular impression, the gland does not exert a following load on the packing unless it is driven by springs. More usually, the packing load is maintained by the escaping fluid acting in two ways: (1) along the rod or shaft, trying to drive the packing radially outward, or (2) along the front wall of the stuffing box, trying to drive the packing axially backward, *i.e.*, against the gland. Since the latter is the higher pressure and is transmitted radially by the packing, it overpowers the former, especially as the gland is approached. Removal of a worn stuffing-box packing will almost always reveal that the ring next to the gland has been forced against the rod or shaft most strongly, has been worn the most, and has done most of the work.

A weakness of all stuffing boxes is that the packing, being wholly confined, is subject to unstable thermal expansion and seizure if overtightened. This can be minimized by effective cooling, by the employment of materials of low coefficient-of-volume expansion, by providing voids within the set into which expansion can take place harmlessly, or by providing a fusible binder which escapes harmlessly from the box if overexpansion occurs. The opposite type of weakness is also encountered, *viz.*, coarse clearances at back and front of the box, through which packing tends to extrude unless prevented by end rings of strong construction but of no packing value, known as "antiextrusion rings."

Since the gland has no function other than to develop an initial load normal to the rod or shaft, it can frequently be dispensed with. In the O-ring packing (Fig. 19-26*b* or 19-34*c*), the initial load is effected by crowding the ring into a groove, thereby deforming it into an ellipse; thereafter the packing is tight against all fluid pressures. In the flange, or "hat," packing (Fig. 19-26*e*) and in the "cup" packing (Fig. 19-34*e*), initial normal load is developed by molding the lip to a slight flare; fluid pressure invading the interior of the packing supplies the continuing load. Fluid load can be supplemented by a spring as in the radial seal (Fig. 19-32*a*,*b*). U packings (Fig. 19-26*f*) are similar but pack against both the rod and outside circumference of the groove. Packings that are capable of contracting radially can be installed in oversize grooves; fluid pressure acting on the outer perimeter develops the necessary packing load, as in the case of the metal packing (Fig. 19-31). In the case of piston packings, fluid pressure admitted to the inside circumference forces the packing outward, as in the rectangular piston ring (Fig. 19-34*a*). In these designs the packings are not closely confined. The effect of thermal seizure is therefore minimized.

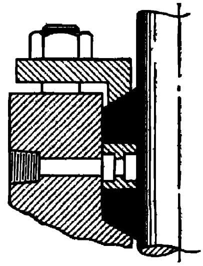


FIG. 19-12. Lantern ring.

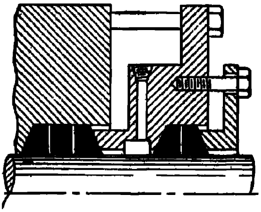


FIG. 19-13. Tandem gland.

Not all sliding-contact packings used to pack rotating shafts operate on cylindrical surfaces. The axial seals (Table 19-5) operate on flat faces. They are subject to the same laws of packing operation as stuffing-box packings but can generally be made to give better performance by careful attention to design and finish.

Although leakage is a necessary characteristic of all sliding-contact packings, there is a way by which any leakage of the fluid being handled can be avoided for the full life of the packing. This consists in injecting into the packing a liquid, known as a "sealing liquid," at a pressure higher than that of the fluid. Under these conditions the flow is reversed, the sealing liquid flowing into the apparatus in addition to escaping to the atmosphere. The means employed to inject sealing liquid (sometimes erroneously referred to as "lubricant") into a stuffing box are generally some modification of the "lantern ring" (Fig. 19-12) or "tandem gland" (Fig. 19-13). In a similar manner, sealing fluid can be introduced into the space between two radial or axial seals or groove packings.

Sometimes one of the above devices is equipped with two connections, one to introduce sealing liquid and the other to bleed away part of it, with incidental removal of some of the generated heat within the packing set. Such an arrangement is called a "circulation fitting." A lantern ring located at the bottom of a packing set is known as a flushing connection and is sometimes employed when there are no objections to the introduction of large volumes of sealing fluid into the apparatus.

Sealing-liquid injection is indicated in examples such as the following:

1. All packings against gases, which, because of low viscosity, leak excessively. This includes all vacuum applications, the steam engine, steam turbine, gas compressor, and all high-pressure autoclaves in which the agitator enters the gas space at the top. It is of interest to note than an original Watt patent for the steam engine called

for seal-oil injection into the piston rings, a rational arrangement which unfortunately has been forgotten by most of his successors.

2. All packings for liquids of very low viscosity or high volatility, such as gasoline, water, etc., under conditions of high speed, high pressure, or both. Instances are the injection of lube oil into the stuffing boxes of oil refinery pumps handling propane, etc., the injection of grease into ram packings operating against high-pressure water, the injection of oil between double axial seals on water pumps.

3. All packings for liquids containing large amounts of dissolved or suspended matter, which tends to crystallize or filter out in their passage through the packing. Instances are the water sealing of centrifugal-pump boxes handling caustic soda, etc., and the water sealing of boxes of hydraulic turbines operating on muddy water.

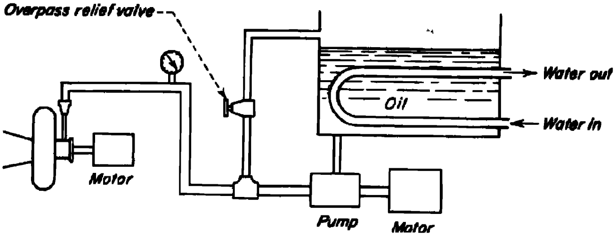


FIG. 19-14. Independent seal-oil system.

4. All packings for fluids of any kind that are poisonous, corrosive, or too valuable to lose. Instances are pure-gasoline injection into the boxes of pumps handling anhydrous hydrogen fluoride in the "alkylation" process, oil injection into ammonia-compressor rod packings, etc.

The sealing medium can be injected continuously or periodically, as by a pulsating lubricator or grease gun, but the latter generally is ineffective as the fluid starts to invade the stuffing box as soon as the original injection pressure has died down. The ideal arrangement introduces the sealing liquid continuously at a pressure only slightly higher than that of the fluid, whereby minimum contamination of the fluid by the sealing liquid is obtained. Means to effect this are the independent seal-oil system (Fig. 19-14), connected directly or through a differential regulator (Fig. 19-15), spring-loaded grease cup (Fig. 19-16), oil bottle (Fig. 19-17), and differential cylinder (Fig. 19-18).

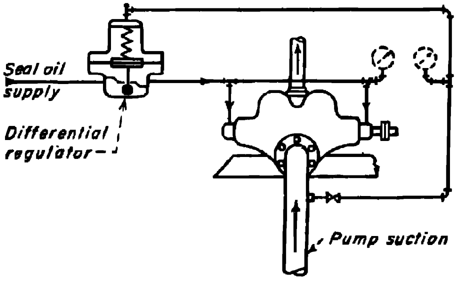


FIG. 19-15. Differential regulator.

Another disagreeable but unavoidable characteristic of sliding-contact packings is that, at the conclusion of operations, fluid is always left trapped in the clearances and in an abnormally favorable condition to cause corrosion of the moving member assuming that this is metal and that the fluid is one, such as water, in which the metal has a measurable solution pressure. In fact, many fluids that are quite incapable of corroding metals in open vessels will corrode them quite freely when trapped in the clearance between packing and the metal. This phenomenon, known as "crevice

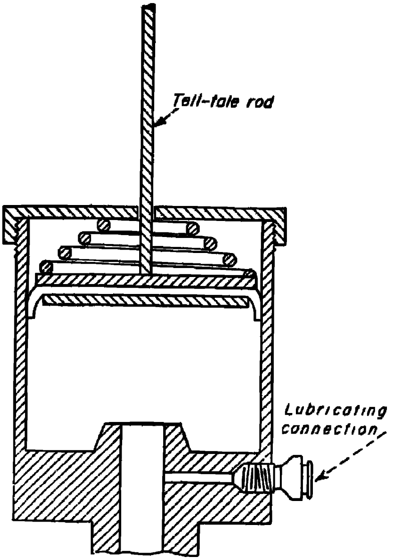


Fig. 19-16. Spring-loaded grease cup.

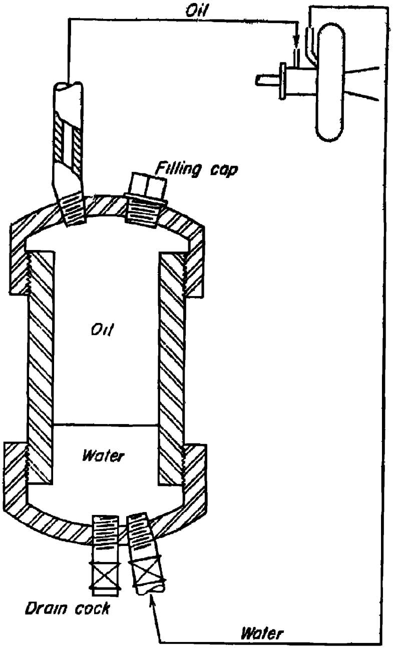


Fig. 19-17. Oil bottle.

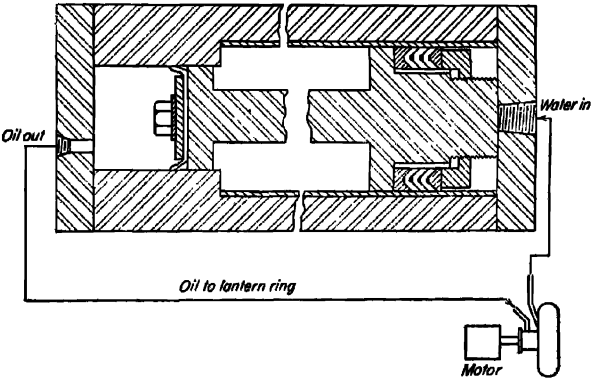


Fig. 19-18. Differential cylinder.

corrosion," is not fully understood but is believed to be associated with the formation of concentration cells within the fluid because of variable content of dissolved oxygen. The packing is usually blamed for such corrosion, but quite erroneously, unless it happens to contain some ingredient that dissolves in and promotes the acidity of the fluid (such as glycerine or free sulfur), or unless it happens to contain a substance which is a conductor and which has a solution pressure differing from that of the metal, under which conditions galvanic corrosion can result. Packing materials that are suspect in the latter connection are copper and brass reinforcements and graphite, although it must be admitted that the whole problem is complex and little understood. Oily packings or packings that have been supplied with oil during operation are much less likely to give rise to corrosion than other types. A good rule, however, in connection with all sliding-contact packings, and especially with those which are used to pack water or aqueous solutions, is to install them just prior to use and to pull them out if the equipment is to be idle for any extended period.

ROD, PLUNGER, OR RAM PACKINGS

Steam Engine. To a considerable extent, the extension of the reciprocating steam engine to progressively higher pressures and temperatures has kept pace with the development of the necessary rod packings. The first engines were packed with braided flax whose life was necessarily short. The introduction of the duck-and-rubber ring (Fig. 19-29*a*) in the late nineteenth century was a great step forward. It is still used for low-pressure steam-driven pumps and steam hammers. With the advent of steam pressures higher than 100 psi, it became necessary to develop better packings; the answer was found in asbestos. A conventional design in rolled rubberized-asbestos fabric is shown in Fig. 19-29*b*. These are made with either a hollow core or some sort of rubber core and usually present a bare asbestos surface to the rod, which is coated with glycerine and graphite. The function of the latter is to level up the surface, cut down abrasion, and, to some extent, shield the rubber layer from the action of hot cylinder oil emulsified inside the engine and carried out on the rod. For services where there is plenty of oil present, as in the case of vertical marine engines, a channel packing with a metallic lead core (Fig. 19-29*c*) is favored.

The principal weakness of these solid packings is that they must be put initially under a gland pressure greater than that of the steam since they become self-tightening only with difficulty. With the added heat caused by rod friction, they become overtight, and if the gland pressure is not reduced promptly, damage results. On a fast-moving rod, therefore, they tend to be somewhat unstable. Also, they, in spite of the use of rubber cores, etc., are not too well adapted to absorb shaft weave. These difficulties have been to some extent overcome by the molded V packings of asbestos-fabric–rubber which, being completely self-tightening, requires little or no initial gland load. Being, in effect, floating packings, they are capable of absorbing expansion derived from frictional heat and also shaft weave. They are, therefore, rapidly becoming the standard type of packing for reciprocating steam-engine piston rods and valve rods for temperatures up to 400°F.

For the higher temperatures, such as those encountered on uniflow engines, segmental metal packings of the type shown in Figs. 19-31*b*,*c*, are used exclusively, with provision for mechanical lubrication. Steam ends of pumps and steam hammers, are, in general, packed like steam engines.

The internal-combustion engine is usually single acting and therefore employs no rod packing. Exceptions are the large gas engines used for powering blowers in blast-furnace plants and compressors on natural-gas lines. Rod packings on these are of the segmental metal type with a lube-oil seal.

The reciprocating compressor is packed with the same types of packing as the steam engine. Service requirements for compressors used to recompress steam are obviously identical with those of a steam engine. The direct-driven air compressor is a particularly difficult rod-packing application because the air is hot and has a strong oxidizing effect on rubber. For this reason, it is customary to omit rubber cores when packing air compressors with the type of packing shown in Fig. 19-29*b*. Another difficulty is oil lubrication of the rod, which is generally less effective in an air compressor than in a steam engine. The larger air compressors, such as those used as blowing engines for blast furnaces, are always packed with segmental metal packing, as are the large gas compressors employed on natural-gas pipe lines; these are supplied with oil by a mechanical lubricator.

Direct-driven ammonia compressors represent a less severe service and are generally packed with duck-rubber rings with a lube-oil seal to prevent escape of gas. Most refrigerant compressors today, including those used for domestic refrigeration, are of the single-acting type and therefore employ no rod packings. Their packing problems are discussed under Piston Packings and Rotary Shaft Packings.

Compressors for hydrogen, as employed in the production of synthetic ammonia, represent a difficult application both because of the high pressures and the mobility of the gas. Compressors for oxygen and chlorine involve a complication due to the chemical reaction of these gases with packing materials and lubricants.

Air Motors. Pneumatic cylinders are employed widely as a means for transmitting power. The largest single use is the railroad air brake, but the cylinders employed on this are of the single-acting type and do not employ rod packing. Rod packings are employed on double-acting pneumatic cylinders and, since the air is cold, present no particular difficulties provided an ample supply of lubricating oil or condensed water is present for sealing purposes. Double-acting pneumatic cylinders are used for raising and lowering the locomotive reverse gear and for the pneumatic rock drill, air hoists, and air-operated valves and machinery. The most widely used packing is perhaps the duck-rubber V ring or one of its modifications. Braided asbestos is used for the smaller rod-packing applications.

The reciprocating pump represents the original, and still the largest, use for reciprocating rod and plunger packings. Types employed include square or V rings of plain rubber or other nonreinforced polymer "plastic" packings, leather, braids, metal-foil packings, rubber-fabric constructions, and combinations of these. For water, square-braided flax is the oldest and still one of the most widely used; it owes some of its effectiveness to the tendency of flax to swell in water. A similar product saturated with hard wax is used for high-pressure water in hydraulic systems. Rubber backs (Fig. 19-30*a*) and cores are employed to absorb shaft weave. Rubber or duck-rubber rings are used in conjunction with braided flax to exclude suspended matter and to prevent extrusion through clearances. Similar effects are secured by "channel" (Fig. 19-30*b*) and "braided diagonal" (Fig. 19-30*c*) combinations. Duck-rubber V rings have to some extent superseded these older types for water service. Metal-foil rings and fabric V rings bonded by oil-resistant rubbers are used for oil.

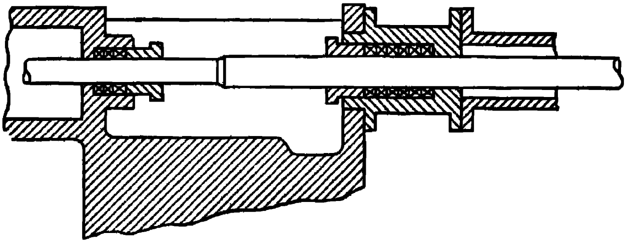


FIG. 19-19. Water gland.

Fluid sealing to exclude the product from the stuffing box is not so successful with reciprocating as with rotary pumps owing to the motion of the rod. When employed, as on heavy-duty water pumps, the "fluid" is generally a heavy grease which will make a permanent film on the rod and not be washed off on each instroke. Damage to packing caused by dissolved salts drying on the rod can be avoided by causing the rod to pass through a "water gland" (Fig. 19-19). A similar arrangement employing a suitable solvent can be used in connection with dissolved organic materials.

Water Motors. The most widely used reciprocating water motor is the hydraulic ram. Developed originally for baling cotton, it has acquired increasing importance in related fields such as oilseed extraction, plastic molding, and various metallurgical operations such as shearing, piercing, forging, and extrusion. Most hydraulic rams are single acting, vertical, and designed to return by gravity. This necessitates a packing that is almost completely self-tightening, so that friction on the return stroke may be minimized. U packings originally used for this service have been largely displaced by V rings of fabric–rubber or leather treated with hard wax.

The dead-weight type of hydraulic accumulator, used with hydraulic systems, functions alternately as a motor and a pump. The hydraulic intensifier, used to boost hydraulic pressures, is essentially a water motor, direct-connected to a pump of smaller plunger diameter. Packings for these are generally V rings.

Oil Motors. Oil hydraulic cylinders, also known as "oildraulic," deserve almost a section by themselves because of the tremendous expansion in their usage in the years prior to and during the Second World War. The advantages of oildraulic systems over hydraulic lie in freedom from scoring and corrosion of the piston rods and cylinder walls and in the possibility of using simple rotary pumps, such as gear pumps, for generating the necessary pressure. One of the earliest oildraulic power applications was the steering engine for ships. Later came the automobile and aircraft shock absorbers and the somewhat similar devices employed for absorbing gun recoil. These function alternately as pumps and as motors. More recent are oil jacks and lubricating lifts for autos, oildraulic presses, and oildraulic actuating cylinders for aircraft, dump trucks, tractors, tanks, and machine tools.

Packings employed on the large rams used for plastic-molding presses, etc., are usually duck–neoprene V rings. For the smaller double-acting piston motors, two types of rod packing are widely employed. One is a V ring of pure buna-N compound without fabric reinforcement; the other is a buna-N O ring. Single-acting piston motors of the type used for automobile brakes do not, of course, employ any rod packing, and their packing requirements are dealt with under Piston Packings. A rather complete list of reciprocating-pump, compressor and fluid-motor packing recommendations for various services is given in Tables 19-5 and 19-6.

Sleeve valves are used as boiler blowoff valves and as selector valves for use with pneumatic, hydraulic, and oildraulic motors. Packings are required to be of rigid design to permit crossing open ports. Rectangular hard-rubber rings are used for the former, V rings and U packings for the latter.

Reciprocating expansion joints and valve stems are dealt with later.

PISTON PACKINGS

A piston packing differs from a rod or plunger packing in that it packs on its outer circumference, thereby increasing the width of the leakage path and the difficulty of securing low leakage. Also it is generally not readily accessible for tightening. As a partial compensation, the leakage requirements of a piston packing are less exacting, since, at least in the case of double-acting pistons, leakage is not lost to the system but merely lessens the efficiency of the equipment.

Steam engines, internal-combustion engines, and gas compressors invariably use metal piston rings in grooves. In the smaller cylinders, these are merely cast-iron rings with a bevel cut (Fig. 19-34*a*). They are made larger than the cylinder diameter to give an initial pressure but in action derive their effective pressure from steam or gas getting into the groove beneath the ring and forcing it out against the lesser pressure of escaping steam or gas along the working face. To pack effectively, they require a liquid seal which is derived from condensed steam or from lube oil, the latter being either atomized into the cylinder or fed through appropriate ports in the piston. Larger units of these types have segmental piston sets (Fig. 19-34*f*).

Air-motor pistons are sometimes packed with metal rings, as in the pneumatic rock drill, but more generally are packed with one or two cups, according to whether the cylinder is single or double acting (Fig. 19-34*e*). The cups are of leather or of oil-resistant rubber, with or without reinforcement of cotton fabric. A cup used on the single-acting operating cylinders of railroad air brakes is shown in Fig. 19-20; similar cups are used on the double-acting cylinders employed to operate the locomotive reverse gear. All cups are self-tightening in that the full fluid pressure within the cup opposes the reduced pressure of the escaping fluid along the working face. To make this effective, it is necessary that contact with the cylinder wall be principally at the lip and that the heel be held slightly away. Also, air cups are effective only when supplied with an oil film, which must be renewed at intervals.

Reciprocating-pump pistons for water or aqueous solutions are usually packed with a hard variety of duck-rubber packing known as "hydraulic." Hardness is secured, in part, by substituting fine fabrics (sheetings or drills) for duck. It is supplied in the form of rings cut from bias strip with a step joint (Fig. 19-34*b*) or prepared from such strip in the field. Rings are used singly in grooves (Fig. 19-34*a*) or in sets with the joints staggered. Rings of hard rubber (ebonite) are used for the same service. Many water pumps, including the familiar farmhouse well pump, are packed with leather or duck-rubber cups. Mud pumps used in rotary drilling represent a severe piston-packing service—they are packed with massive rubber cups reinforced with fabric or metal or with V rings (Fig. 19-34*d*). Oil- and solvent-pump pistons are also packed with cups or with rings cut from laminated phenolic resin.

Liquid piston motors are generally of the oil-operated type. The pistons are packed with V rings or O rings. Rubber cups are employed, however, on the automobile hydraulic brake, the fluid in this instance being an alcohol mixture which has no appreciable swelling action on rubber.

Piston valves are packed in the same manner as the pistons of engines, air motors, or liquid motors, with the added complication that the packing is required to pass ports and must therefore be of fairly rigid construction.

Commonly used piston packings are classified in Table 19-8. A list of recommendations for utilities and general services is given in Table 19-9 and for chemical services in Tables 19-7 and 19-10.

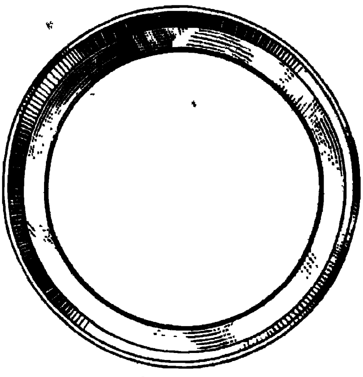


FIG. 19-20. Railroad air-brake cup.

ROTARY SHAFT PACKINGS

Rotary shaft service is generally more severe than reciprocating in that surface speeds are high, and facilities for dissipating frictional heat are poor. Solid duck-rubber packings, V packings, etc., such as are commonly used on reciprocating service, are not suitable on rotary service for any but the slowest speeds, since they will heat up and seize. Service is limited to types which are either inherently low in friction or which contain voids or a fusible binder which will act as a safety valve against thermal expansion.

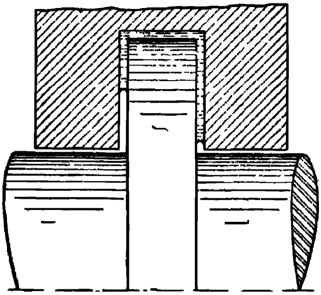


FIG. 19-21. Water seal.

Steam-turbine packings are of three types, none of which is employed to any extent for other rotary applications. The carbon-ring packing is similar to the segmental metal type except that it is made of graphite; it is installed in an oversize groove. It depends for its effectiveness on the seal formed by condensate. The water seal (Fig. 19-21) is scarcely a packing at all; it employs coarse clearances and depends on the principle of the U tube; the normal gravitation head of the water in the seal is intensified by the high rotational speed. Means must be provided to replenish the water in the seal if condensation is not sufficient. Similar liquid seals employing mercury are used on vertical centrifugal pumps. The labyrinth packing (Fig. 19-22) is employed on the larger turbines. It likewise employs coarse clearances, but the escape of steam is retarded by the length of escape path, which includes a miniature vortex within each annular pocket. Condensate is not necessary for the effectiveness of a labyrinth packing.

The centrifugal pump represents probably the largest single use for rotary packings. Speeds are high, but pressures opposite the packing are usually low, approximating that of the pump inlet. Nevertheless, to get any length of life from conventional packing necessitates a leakage tolerance on the order of 1 cc per min per circumferential inch of shaft when handling water. Pumps operating under higher pressures, such as those used as booster pumps on closed systems or multistage pumps, necessitate even higher leakage tolerances unless the pressure can be partly bled away by a lantern ring at the bottom of the box connected to a point of lower pressure. Some pumps are equipped with internal "pump-out" arrangements which make it possible to dispense entirely with packing except when the pump is idle. With more viscous liquids such as oil, somewhat lower leakage rates can be maintained, and the necessity for special arrangement is correspondingly lessened.

Since a centrifugal packing makes no pretense of being fluid-tight, sealing is indicated for all vacuum applications and for all positive pressure applications involving suspensions, concentrated solutions, or fluids that are extremely volatile, noxious, or valuable. For concentrated aqueous solutions with conventional packings, the usual sealing fluid is cold water from an independent source. Oil-refinery pumps handling solvents of low viscosity or high volatility are usually sealed with lube oil.

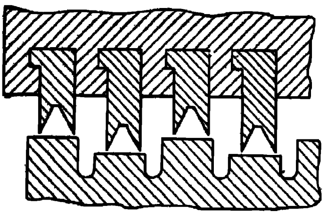


Fig. 19-22. Labyrinth packing.

Packing types employed with centrifugal pumps include "plastic," metal foil, braided, and axial seals. The first three are used in conventional stuffing boxes and require no special description. Axial seals are increasing in popularity and, in many cases, result in superior performance.

An axial seal consists essentially of a rotating ring or shoulder attached to the shaft making contact on one of its flat faces with a stationary ring attached to, or integral with, the pump housing. The patent literature on axial seals embraces several thousand designs, and new ones are being patented continually. These, however, may be reduced to a relatively few categories. A primary division is between single seals and those involving two seals in tandem ("double" seals). A second basis of classification for single seals is between those (designated "inside") in which the fluid pressure acts to tighten the seal (Figs. 19-33*a*, *b*, *g*, *h*, *i*) and those (designated "outside") (Figs. 19-33*c*, *d*) in which the pressure tries to blow the seal apart. Double seals always include one "inside" and one "outside" seal and may be differentiated as "expanding" (Fig. 19-33*e*) and "contracting" (Fig. 19-33*f*). Since all seals involve one flexibly suspended member, a third division is between those (which may be called "regular") in which the flexibly suspended member is stationary (Figs. 19-33*a*, *c*, *g*, *h*) and those (which may be called "inverted") in which the flexibly suspended member rotates with the shaft (Figs. 19-33*b*, *d*, *i*). A fourth basis of classification is the type of flexible suspension employed, *viz.*, sliding contact, flat diaphragm, and cylindrical diaphragm (or "bellows"). These are illustrated, for inside regular designs only, in Figs. 19-33*a*, *g*, and *h*, respectively.

The advantages and disadvantages of these various constructions are for the most part based on relative cheapness of design, accessibility, etc., and all types are capable of giving good service when operating against pure liquids such as water or oil. When operating against gases or against pure liquids containing dissolved or suspended matter, it is necessary that they be sealed just as any other packing. This may be done by injecting lube oil or grease directly into the sealing face or, in the case of a double seal, into the space between the two seals at a pressure higher than that of the fluid being packed. Selection of materials for seal rings, suspension member, and spring is governed by service conditions and should be referred to the seal manufacturer. Axial seals do not eliminate leakage, although they reduce it to a very low value—on the order of $^1/_{100}$ cc per min. As compared to other packing types, they are expensive, difficult to repair or replace, and if incorrectly selected for the application, may fail as quickly as, or quicker than, conventional types. Their use on centrifugal pumps is increasing.

Rotary displacement pumps and turbopumps are packed the same as centrifugal pumps. Surface speeds are generally lower, but pressures opposite the packing are generally higher, since the shaft in these pumps represents a point midway in the leakage path across the face of the rotor and therefore is at a pressure midway between inlet and discharge. In screw pumps, full discharge pressure comes opposite one packing. These pressures can be avoided, however, by tapping the bottom of the stuffing box directly to the inlet side at a minor sacrifice of pump efficiency.

Rotary gas compressors range from simple fans to pressure blowers of the Roots-Connersville type. Sealing liquid is indicated unless already present in the compressor body, as in the case of the Nash compressor. The type of packing is governed by the sealing fluid employed.

Agitators are packed the same as centrifugal pumps except that, on the slower moving shafts, it is sometimes possible to use duck-rubber V rings. Packings operating against gases, which applies to practically all chemical reactors in which the agitator enters at the top, should be sealed with some fluid that is compatible with the material in the reactor. Sometimes one of the reactants is employed for sealing purposes.

A list of recommendations for centrifugal pumps, displacement pumps, rotary gas compressors, or agitator shafts against fluids is given in Tables 19-6 and 19-7.

Fluid drives, fluid brakes, and rotary shock absorbers are similar to displacement or turbopumps and are packed in the same manner.

Swivel joints are required to handle all kinds of fluids at slow speeds; packings are generally cups or V rings. Two important swivel-joint applications are (1) steam connections for railway cars and (2) drilling mud connection to rotary rock drill. Ball joints used as swivels are described under Miscellaneous Sliding-contact Packings.

Hydraulic Turbines and Propeller Shafts. Packings for shafts of hydraulic turbines and for propeller shafts of vessels are analogous to centrifugal-pump packings and are packed in the same way. Braided flax seems to be preferred for these with, occasionally, rubber end rings to filter out suspended sand. Better results on dirty water are secured by sealing with clean water. This is often done with water wheels but less often with propeller shafts. Either application is operated on a continuous water leak.

An important application for rotary shaft packing is the crankcase of the *reciprocating* internal-combustion engine. There is little pressure but a great deal of surface speed, shaft gyration, and heat. This is an important application for the radial seal (Fig. 19-32) which consists essentially of a synthetic-rubber flange packing and a spring, the whole being usually encased in a stamped-metal shell which merely needs to be pressed into or bolted to the cavity surrounding the shaft. Springs are of the lug or garter type. Sometimes the shell reduces to a reinforcing member within the element, and the spring is omitted. Unfortunately, many internal-combustion engines, including the rear end of the crankcase on the ordinary automobile engine, are so designed that any seal employed must be split for installation. As a result the radial seal has not heretofore been able to displace conventional packings for this application, the latter consisting usually of half circles of cork, felt, or braid installed in grooves. Split radial seals, however, have become recently available. Radial seals are regularly employed on shafts leading to generators, distributors, and other accessories. There are a great many such seals on the modern aircraft engine. The piston ring seal (Fig. 19-31*a*) is used for somewhat similar purposes.

Single-acting reciprocating air compressors employ crankcase seals similar to those used on engines. Single-acting refrigerating compressors, however, usually operate with a pressure in the crankcase, which rises to the vapor pressure of the refrigerant at shutdown times. Radial seals are not well designed to pack against pressure, and resort is had to conventional packings or to axial seals. Conventional packings used for refrigerant service are largely duck-rubber slab or metal foil, packing being sealed by lube oil from the compressor lubricating system. Where the refrigerant has a swelling action on the rubber, as for instance, Freon or methyl chloride, packings with solvent-resistant binders must be employed. Success with conventional packings on refrigerant compressors is largely a matter of care in wearing in. Axial seals are expensive and difficult to replace but when properly designed are satisfactory and require no attention. Axial seals of the metal-bellows suspension type were widely employed on domestic refrigerating machines until the advent of the hermetically sealed unit, which requires no packing.

Bearings and Gear Cases. The most conspicuous expansion of the radial seal has been in the general field of bearings and gear cases. Until comparatively recently, such apparatus was either allowed to leak oil freely or was lubricated with semisolid grease. The automobile manufacturer, because of the need of protecting his expensive ball and roller bearings, inaugurated the first widespread use of radial seals. Many such seals are now incorporated integrally with the ball bearings; since the bearing in such cases is supplied with grease, the function of the radial seal appears to be principally that of excluding dirt and water. Many such seals, however, are employed against liquid lubricants and serve to cut losses to a minimum. Instances are electric-motor bearings, pillow-block bearings, all kinds of reduction gearing, belt pulleys, chain sprockets, conveyor rollers, rollers for dam gates, etc. Very large radial seals are employed on steel-mill roll necks and on the gun platforms of warships. Work is under way that will probably lead to the use of radial seals for railway-car journals.

VALVE-STEM AND EXPANSION-JOINT PACKINGS

Valve-stem-packing service is distinguished from preceding types of sliding-contact service by slight and infrequent movement, continuous high pressure, and little or no attention. Regarding the latter, many valves are in inaccessible locations where packing adjustments can be made only with great difficulty.

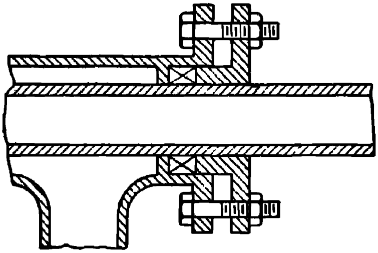


FIG. 19-23. Expansion joint - double-tube heat exchanger.

Direction of motion in valve-stem packing can be reciprocating, rotary, or a combination of both, *viz.*, helical. Which direction is not of great importance because of the low speed involved. Packings may be of the reciprocating types, *viz.*, fabric–rubber-slab or rolled constructions; V rings, cones, O rings, etc., or they may be of the rotary types, *viz.*, braids, plastics, or metal foil. The real problem is one of fluid sealing, since valve users will seldom go to as much trouble to seal valve-stem stuffing boxes as they will to seal pump stuffing boxes. The same necessity for sealing, however, exists when dealing with gases, vacuum, or liquids that are of low viscosity, volatile, corrosive, poisonous, or full of dissolved or suspended matter. The following sealing means, both more or less makeshift, are employed:

1. Valves carrying condensible gases, such as steam, are provided with extension bonnets with cooling fins to promote condensation, the liquid providing the necessary sealing. Inverting the valve is also helpful.
2. Stuffing boxes, if accessible, are provided with lantern rings and alemite or other grease connection for intermittent grease injection. This arrangement excludes the fluid from the packing for only a brief period, in addition to which it is frequently neglected. In lubricated plug cocks, the fluid injected to lubricate the plug serves also to seal the stem packing.

Most of the troubles associated with valve-stem packings would be avoided if manufacturers would provide sealing connections and if users would couple these to a central source of lube oil at a pressure greater than that of any of the fluid lines or else provide each valve with a properly designed spring-loaded grease cup. Some such device is especially needed with control valves to minimize hysteresis resulting from valve-stem-packing friction. Such arrangements are for fluids other than pure water or oil, these causing no particular trouble if a slight leak is permitted.

Certain valves impose special requirements on the stem packing. In some conical plug cocks, the stem packing is required to hold the plug in place; this necessitates a somewhat denser construction than usual. In a plumbing faucet, the stem packing is driven flush into the stuffing box by the act of tightening the bonnet and is not thereafter adjustable. This necessitates a packing that is closely controlled for volume, does not lose volume rapidly in service, and has voids to absorb thermal volume changes. The duck-rubber-slab packing meets these requirements.

Reciprocating expansion joints, although larger than valve stems, are similar in respect to service requirements, since movement is slow and infrequent and also since they are required to handle a wide variety of fluids. In addition to their use in pipe lines, they are employed in heat-exchanger equipment of the double-tube (Fig. 19-23) and shell-and-tube (Fig. 19-24) types.

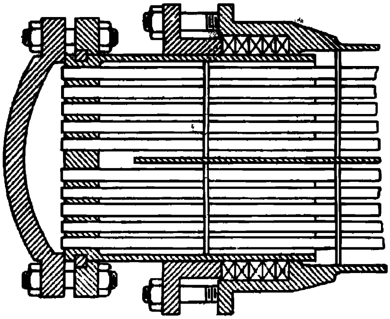


FIG. 19-24. Expansion joint—shell-and-tube heat exchanger.

Recommendations for valve-stem and expansion-joint packings for various services are given in Tables 19-6 and 19-7.

MISCELLANEOUS SLIDING-CONTACT PACKINGS

The preceding material has dealt with packing the interface between concentric cylindrical members. Sliding-contact packings are also used to pack flat, conical, and spherical interfaces.

Axial seals represent a type of flat-surface packing, although used in lieu of a conventional cylindrical-surface packing. More typical flat packings are those used with ported valves. As far as is known, the familiar steam-engine D slide valve depends entirely on metal-to-metal fit, but somewhat similar three- and four-way valves used for air, water, or oil are sometimes packed along the flat interfaces. Relatively hard packings, such as laminated phenolic resins, are used to avoid damage to the packing in passing ports.

The only well-known application of a packing for a conical surface is the so-called "lined" plug cock. The lining, which is really a packing, is required to pass ports as the plug rotates. It therefore must be made of firm material. A molded asbestos-ebonite composition is favored.

Spherical packings for ball joints are widely used. Ball joints are located in pipes to permit rocking or a combination of rotation and rocking. One of the former type is the Mallet engine joint shown in Fig 19-25. In addition to packing the ball against steam leakage, the packing in this case is also required to support the ball thrust and therefore must be made of firm material. A folded asbestos-fabric-rubber construction is currently employed.

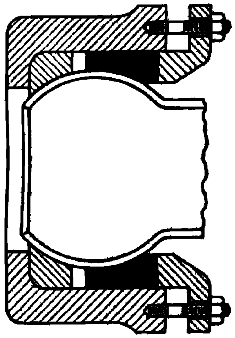


FIG. 19-25. Mallet ball joint.

GASKET AND SLIDING-CONTACT-PACKING CLASSIFICATION AND RECOMMENDATIONS

Author's Note. The gasket and rod-packing recommendations given in this section represent merely the most widely used industrial practice. They do not pretend to be exhaustive and undoubtedly there are instances under the various service classifications wherein types of packing not indicated may be giving service equal to or better than the types shown.

HOW TO SELECT PROPER PACKING

Classification of mechanical packings and recommendations for specific services have been summarized in Tables 19-2 through 19-10. To select the proper packing or gasket for a specific service, refer to the appropriate table to find the type or types recommended. Then turn to the preceding classification table to interpret the recommendation.

EXAMPLE: To find recommended gasket for standard pipe flanges in ammonia gas service, see Table 19-3, in which gasket recommendations for utilities and general services are presented. For ammonia gas at temperatures up to 220°F, style 211 gaskets are recommended. Now referring back to Table 19-2, in which flange gaskets are classified, it will be noted that style 211 gaskets are of the compressed asbestos type (2, first column), made from white asbestos fibers (1, second column), and bonded with natural rubber or GR-S (1, third column).

Table 19-2. Classification of Flange Gaskets

Nonmetallic gaskets

General type	Materials		Forms available
	Reinforcement	Binder	
1. Plain rubber or other polymer	0. None	1. Natural rubber or GR-S 2. Neoprene 3. Buna N 4. Butyl 5. Thiokol 6. Teflon 7. Polyvinyl chloride (Koroseal, Tygon, etc.) 8. Silicone rubber	Sheet and gaskets cut from sheet Strip and gaskets spliced from strip Sleeves and gaskets cut from sleeves Molded shapes
2. Compressed asbestos	1. White asbestos 2. Blue asbestos	1. Natural rubber or GR-S 2. Neoprene 3. Buna N 4. Butyl	Sheets and gaskets cut from sheet
3. Leather	1. Combination tannage leather 2. Chrome-tanned leather	5. Thiokol 8. Wax 9. Phenolic resin	Hides and gaskets cut from hides
4. Braid	1. Flax or jute 2. White asbestos with copper or brass wire 3. Blue asbestos	0. None 1. Natural rubber or GR-S 8. Silicone rubber	Eight-carrier braid Multiple-carrier braid (solid) Multiple-carrier braid (hollow) Interlocking
5. Fabric	1. Cotton fabric ("CI sheet") 2 White-asbestos fabric 3. White-asbestos fabric with copper or brass wire 4. Blue asbestos	0. None 1. Natural rubber or GR-S 2. Neoprene 8. Silcone rubber	Sheets and gaskets cut from sheet Strip and gaskets spliced from strip Gaskets rolled or folded from endless sleeves

Table 19-2. Classification of Flange Gaskets (*Cont.*)

Nonmetallic gaskets

General type	Materials		Forms available
	Reinforcement	Binder	
6. Paper..............	1. Vegetable fiber 2. Vulcanized fiber (vegetable fiber treated with zinc chloride) 3. White asbestos 4. Blue asbestos	0. None except sizing 1. Natural rubber or GR-S 2. Neoprene 4. Butyl 5. Glue-glycerine	Sheets and gaskets cut from sheet

Metallic and semimetallic gaskets

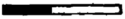
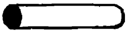
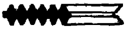
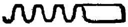
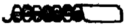
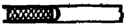
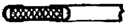
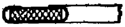
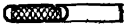
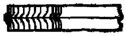
General type	Materials of construction		Physical shape or construction	Sketch showing cross section
	Gasket	Filler		
7. Solid metal..	1. Aluminum 2. Copper 3. Iron or soft steel 4. Monel 5. Stainless steel 6. Lead	0. None 1. White-asbestos yarns 2. Blue-asbestos yarns	1. Rectangular or flat 2. Oval or octagonal (ASA ring) 3. Lens shape 4. Profile 5. Corrugated	 Rectangular  Oval  Profile  Corrugated (without filler)  Corrugated (with Asbestos filler)
8. Jacketed....	1. Aluminum 2. Copper 3. Iron or soft steel 4. Monel 5. Stainless steel 6. Lead 7. Zinc 8. Nickel 9. Teflon	1. White-asbestos millboard 2. Compressed-white asbestos 3. Woven-asbestos fabric 4. Rubber or synthetic rubber	1. French type 2. Edge jacketed 3. Semienclosed jacketed 4. Totally enclosed jacketed	 French type  Edge jacketed  Semi-enclosed jacketed  Totally-enclosed jacketed
9 Spiral wound	1. Cadmium-plated or galvanized low-carbon steel 2. Stainless steel 3. Monel 4. Bronze	1. White-asbestos paper 2. Blue-asbestos paper 3. Teflon	1. Spiral-wound crimped metal ribbon with interlayers of filler	 Spiral wound Asbestos filled type

Table 19-3. Gasket Recommendations for Utilities and General Services

Service	Type of gasketed joint						
	Standard pipe flanges			Large-diam flanged joints of tanks, boiler manholes, condensers, etc.		Irregular-shape gasketed joints, cylinder heads, heat exchangers, etc.	
Temperature	Up to 220°F	220 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	Up to 220°F	220 to 500°F
1. Water (boiler feed, condensate, drinking, sea, calcium chloride brine, etc.)	101	211	....	101	211	101	211
	211	....	....	211	531	211	8214
	...	...	.	531	....	8214	
2. Oil (fuel, hydraulic, heat transfer, lubricating)	102	....	...	102	....	102	
	103	....	...	103	....	103	
	211	211	8214	211	211	211	211
	8214	8214	7302	8214	8214	8214	8214
	618	....	...	532	532	618	
3. Gasoline....................	213	....	....	103	....	213	
	618	....	...	213	....	618	
	103	...	..	532	....	103	
4. Steam......................	101	211	9211	101	211	101	211
	211	9211	...	211	531	211	8214
	....	....	...	531	9211	8214	
5. Air......................	101	211	....	101	211	101	211
	211	....	...	211	....	211	
6. Vacuum..................	101	....	...	101	...	101	
7. Ammonia gas..........	211		...	....	....	211	
8. Freon.................	211	..	....	211	....	211	
	212		.	212	....	212	
	213	....	...	213	....	213	
9. Dowtherm..................	....	...	7302	....	8314		
	..	...	7304				
	....	...	7204				

Table 19-4. Gasket Recommendations for Chemical Services

Service	Standard pipe flanges			Large-diam flanged joints (manholes, distillation columns, etc.)		Irregular-flanged joints (cylinder heads, heat exchangers, etc.)		Glass, glass-faced, or Haveg joints
Temperature	Up to 220°F	220 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	Up to 220°F	220 to 500°F	Up to 220°F
1. Water								
(Neutral, alkaline, or weakly acid solutions)								
Salt solutions and brines								
Ammonium nitrate								
Sodium carbonate								
Sodium chloride								
Sodium hypochlorite								
Sodium sulfite								
Trisodium phosphate								
Organic acids								
Dilute acetic	101	211	....	101	211	101	211	101
Citric	211	......	....	211	531	211	531	
Lactic	......	......	....	531	......	531		
Carbonic								
Alkalies								
Caustic soda								
Caustic potash								
Aqua ammonia								
Slurries								
Miscellaneous								
Sugar solutions								
Molasses								
Dilute hydrogen peroxide								
2. Food products								
Aqueous								
Fruit juices Beverages Pharmaceuticals	101	. .	. . .	101	.....	101	....	101
Oils and emulsions								
Milk Mayonnaise	102	.....	. ..	102	......	102	....	102
3. Mineral acids								
(With or without dissolved salts or suspended matter)								
Hydrochloric	101	......	...	101	......	....	....	101
Hydrofluoric (aqueous)	101	....	....	101	.... .	.. .	....	101
Hydrofluoric (anhydrous)	106	... ..	....	8921 8931				
Phosphoric	101	... .	....	101	......	....	....	101
Sulfuric to 60 per cent	101 211	....	. . . .	101 211	....	. .. . ..	....	101 8931
Sulfuric above 60 per cent	211	211	.	211	211	. ..	....	8931
Nitric to 60 per cent (including mixtures with sulfuric)	106 107 221	106 221	....	8921 8931 221	8921 8931 221	....	....	8931 644 107
Nitric above 60 per cent (including mixtures with sulfuric)	106 221	106 221	.... . .	8921 8931 221	8921 8931 221	. ..	....	8931 644
4. Oils and solvents								
(With slight or no swelling effect on natural rubber or GR-S)								
Alcohols								
Methyl alcohol								
Ethyl alcohol								

Table 19-4. Gasket Recommendations for Chemical Services (*Cont.*)

Service	Standard pipe flanges			Large-diam flanged joints (manholes, distillation columns, etc.)		Irregular-flanged joints (cylinder heads, heat exchangers, etc.)		Glass, glass-faced, or Haveg joints
Temperature	Up to 220°F	220 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	Up to 220°F	220 to 500°F	Up to 220°F
4. Oils and solvents (*Cont.*) Glycerine Ethylene glycol Aliphatic aldehydes and ketones Formaldehyde Acetaldehyde Acetone Methyl ethyl ketone Aliphatic amines Mono- or di- or triethylamine Mono- or di- or trimethylamine Aliphatic nitro compounds Nitromethane Nitroethane Silicone oils	101 211	211	....	101 211 531	211 531	101 211 8214	211 8214	101
(With slight or no swelling effect on Neoprene or Buna N) Aliphatic ethers Diethyl ether Dimethyl ether Cellosolve Animal & vegetable oil Lard oil Cottonseed oil Aliphatic hydrocarbons Crude oil Petroleum naphtha Gasoline Kerosene Gas oil Fuel oil Lubricating oil Freon F-11, F-12, F-122	102 103 211 212 213 618	211 212 213	....	102 103 532 211 212 213 618	211 212 213 532	102 108 211 212 213 618 8214	211 212 213 8214	102 103 618
(With strong swelling effect on all rubbers) Phenols Phenol-anhydrous (carbolic acid) Cresol Aromatic aldehydes and ketones Benzaldehyde Acetophenone Cyclohexanone Aromatic ethers Dibenzyl ether Esters Methyl acetate Butyl acetate Aromatic or terpene hydrocarbons Benzene Toluene Xylene Coal tar Naphthalene Manufactured gas drip oil Turpentine	211 212 213 105 106 618	211 212 213 106	....	211 212 213 105 618 8931	211 212 213 8931	211 212 213 106 105 618 8214	211 212 213 106 8214	105 618 8931

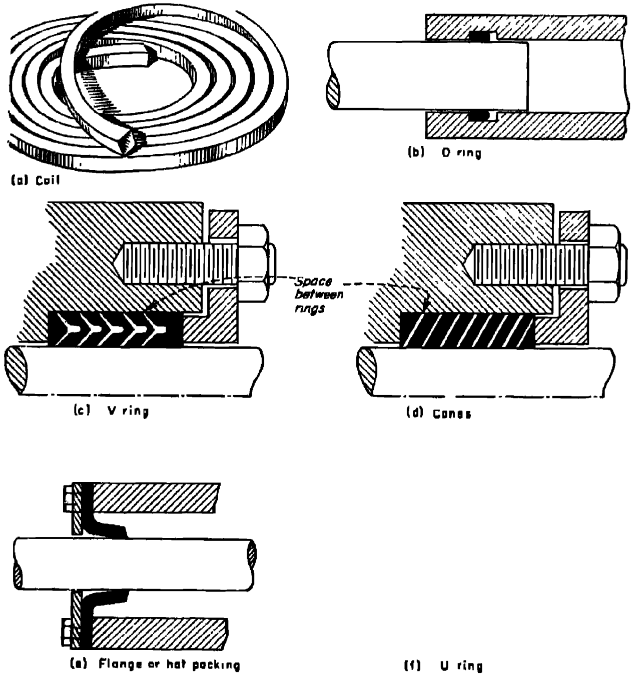
Table 19-4. Gasket Recommendations for Chemical Services (*Cont.*)

Service	Standard pipe flanges			Large-diam flanged joints (manholes, distillation columns, etc.)		Irregular-flanged joints (cylinder heads, heat exchangers, etc.)		Glass, glass-faced, or Haveg joints
Temperature	Up to 220°F	220 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	Up to 220°F	220 to 500°F	Up to 220°F
4. **Oils and solvents** (*Cont.*)								
Pinene								
Aromatic amines								
Aniline								
Mono- or di- or triethylaniline								
Mono- or di-butylaniline								
Di-phenylamine								
Benzidine								
Aromatic nitro compounds								
Nitrobenzine								
Di-nitrobenzine								
Mono- or di-nitro toluene								
Chlorinated hydrocarbons								
Carbon tetrachloride								
Chloroform								
Monochlorbenzene								
Trichlorethylene								
Freon F-21								
Carbon bisulfide								
5. **Special liquids**								
Liquid chlorine Liquid bromine Sulphur chloride	106 8931	.....	...	8931	......	106	...	8931
Acetic acid (conc.) Acetic anhydride	211	211	...	211				
90 per cent hydrogen peroxide Tin and titanium chloride	106							
6. **Dry gases** (With little effect on gasket materials)								
Acetylene Carbon monoxide Carbon dioxide Ethane Ethylene Hydrogen Helium Methane Natural gas Nitrogen Propane Producer gas Sulfur dioxide Hydrogen chloride	101 211	211 7204	9211 7204 8811	101 211 531	211 531	101 211	211	
7 **Special gases**								
Oxygen	106	106						
Chlorine and bromine	211	211	8811					
Hydrogen fluoride (HF)	106	106						

Table 19-5. Classification of Rod, Plunger, or Shaft Packings

A. Conventional rod, plunger, or shaft packings

Type	Reinforcement	Binder	Physical shape or method of fabricating
1. Plain	0. None	1. Natural rubber of GR-S 2. Neoprene 3. Buna N 4. Teflon	1. Rectangular cross section (coil) 2. Round cross section ("O" ring) 3. Nested V rings 4. Nested cones 5. Flange packings 6. U packings
2. Fibrous ("plastic" packing)	1. White asbestos 2. White asbestos with lead particles 3. Blue asbestos 4. Teflon flake	0. None 1. Natural rubber of GR-S 2. Neoprene 3. Buna N 8. Castor oil 6. White petrolatum or equivalent	1. Bulk, or extruded
3. Felt	1. Wool	0. None 1. Natural or synthetic-rubber interlayers	1. Cut strips or rings
4. Leather	1. Combination tannage leather 2. Chrome-tanned leather	1. Synthetic rubber 5. Wax 9. Phenolic resin	1. Nested V rings 2. Flange packings 3. U packings



(a) Coil (b) O ring (c) V ring (d) Cones (e) Flange or hat packing (f) U ring

Fig. 19-26. Plain rod packings (see Table 19-5, Types 1 to 4).

Table 19-5. Classification of Rod, Plunger, or Shaft Packings (*Cont.*)

A. Conventional rod, plunger, or shaft packings

Type	Reinforcement	Binder	Physical shape or method of fabricating
5. Twist	1. Cotton yarn 2. White-asbestos yarn 3. Blue-asbestos yarn 4. Aluminum foil with or without core 5. Lead foil, with or without core 6. Copper foil, with or without core	0. None 1. Natural or synthetic rubber 5. Mineral or fatty oil or wax, or mixtures 6. White petrolatum or equivalent 8. Castor oil 9. Shellac	1. Twisted strands of yarn or ribbons of metal foil
6. Braid	1. Cotton yarn 2. Flax ramie or jute 3. Rayon 4. White-asbestos yarn 5. White-asbestos yarn with metal 6. Blue-asbestos yarn 7. Teflon ribbon	0. None 1. Natural or synthetic rubber 4. Chlorinated or fluorinated lubricants 5. Mineral or fatty oil or wax, or mixtures 6. White petrolatum or equivalent 7. High-flash mineral oil 8. Glycerine, soap, or mixtures	1. Eight-carrier braid 2. Multiple-carrier braid 3. Interlocking braid

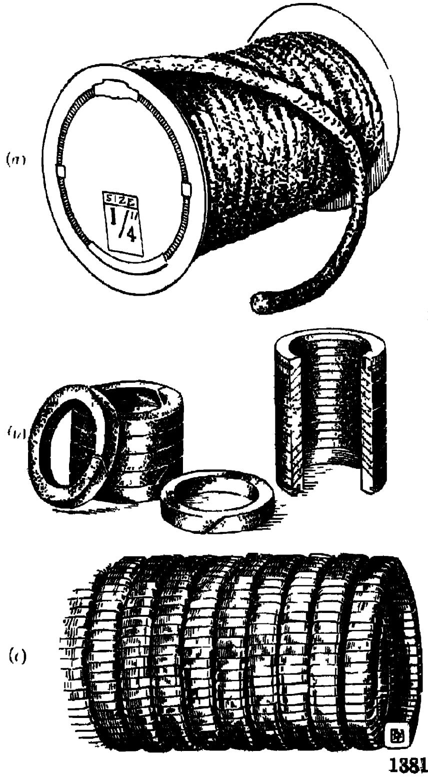


Fig. 19-27. Twisted packings (see Table 19-5, Type 5). (*a*) Twisted asbestos valve-stem packing; (*b*) twisted metal-foil—formed rings; (*c*) helically wrapped metal foil—in spiral form.

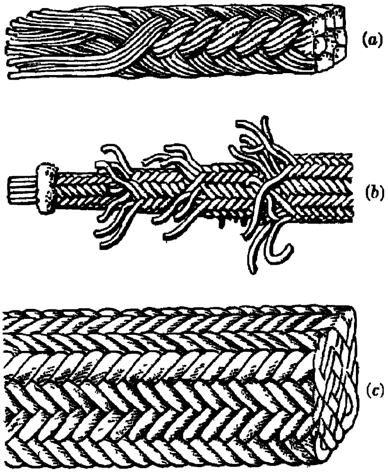


Fig. 19-28. Types of braid (see Table 19-5, Type 6). (*a*) Eight-carrier; (*b*) multiple carrier; (*c*) interlocking.

Table 19-5. Classification of Rod, Plunger, or Shaft Packings (*Cont.*)

A. Conventional rod, plunger, or shaft packings

Type	Reinforcement	Binder	Physical shape or method of fabricating
7. Fabric	1. Cotton duck 2. White-asbestos fabric 3. Blue-asbestos fabric	1. Natural rubber or GR-S 2. Neoprene 3. Buna N 9. Phenolic resin	1. Rectangular cross section cut from slab 2. Same with diagonal plies 3. Rectangular or round cross section–rolled–with or without a rubber core 4. Channel with lead core 5. Nested V rings 6. Nested cones 7. Flange packings 8. U packings 9. Segmental cut rings

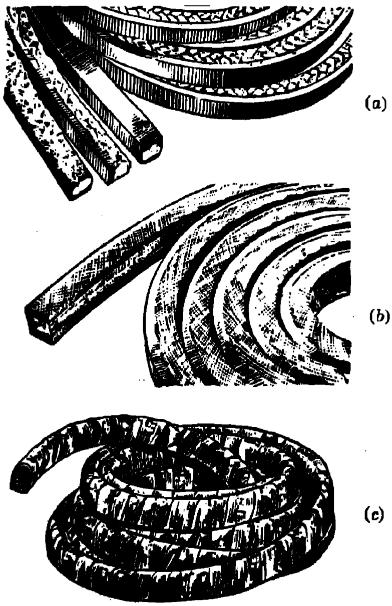


FIG. 19-30. Combination packings (see Table 19-5, Type 8). (*a*) Braided flax with rubber back; (*b*) braided flax in channel of cotton fabric; (*c*) braid and fabric packings encased in muslin wrapper.

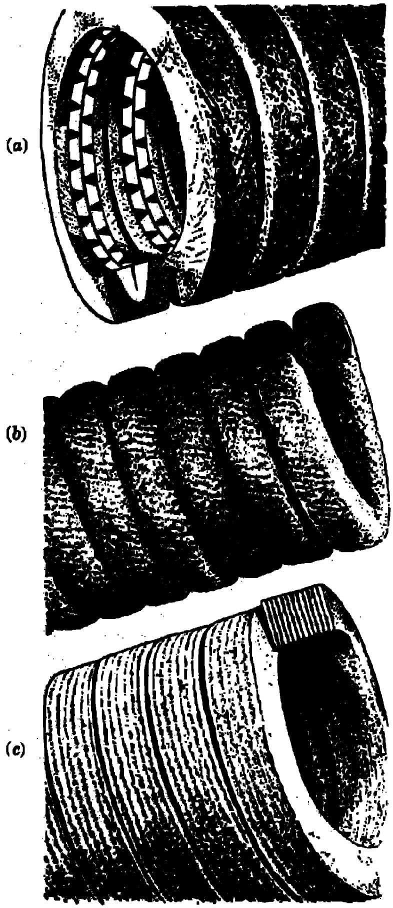


FIG. 19-29. Fabric packings (see Table 19-5, Type 7). (*a*) Asbestos channel with lead core; (*b*) rolled asbestos around rubber core; (*c*) cotton slab packing.

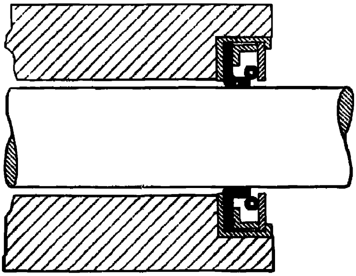
Table 19-5. Classification of Rod, Plunger, or Shaft Packings (*Cont.*)

A. Conventional rod, plunger, or shaft packings

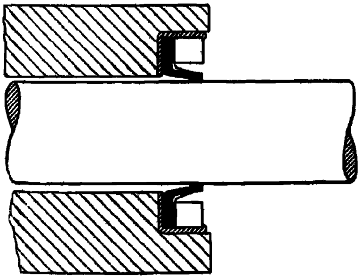
Type	Reinforcement	Binder	Physical shape or method of fabricating
8 Combinations of rubber or fabric rubber with braids (Component parts made up of Types 1 6, or 7)			1. Braid with rubber back 2. Braid with rubber core 3. Braid contained in channel of rubberized fabric 4. Braided and duck-rubber rings encased in skeleton cotton braid or muslin wrapper
9. Solid metal or carbon	1 Cast iron 2 Bronze 3. Babbit 4. Graphite	0. None	1. Split rings 2. Segmental cut rings

B. Radial seals

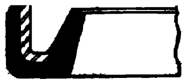
Seal element	Spreader or spring	Enclosure	Type
1. Neoprene 2 Buna N 3. Leather 4. Felt 5. Combination	0. None 1. Garter 2. Lug	0. None 1 Metal case 2 Embedded metal reinforcing ring 3. Phenolic resin outer ring 4 Frictioned fabric outer ring	1. Solid ring 2. Split or cut open



(a) Metal case-garter spring



(b) Metal case-lug spring



(c) Embedded metal-no spring

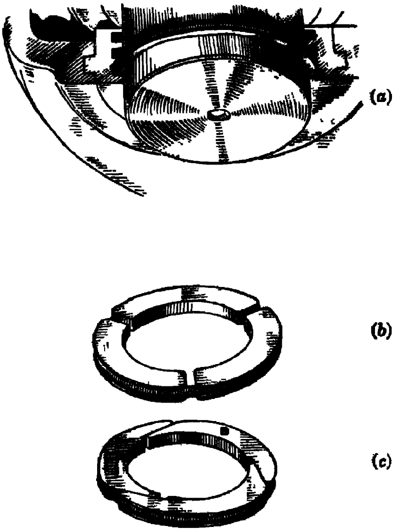


FIG. 19-31. Metal-rod packings (see Table 19-5, Type 9). (*a*) Piston-ring seal; (*b*) segmental ring—radial cut; (*c*) segmental ring—tangential cut.

FIG. 19-32. Radial seals (see Table 19-5*B*).

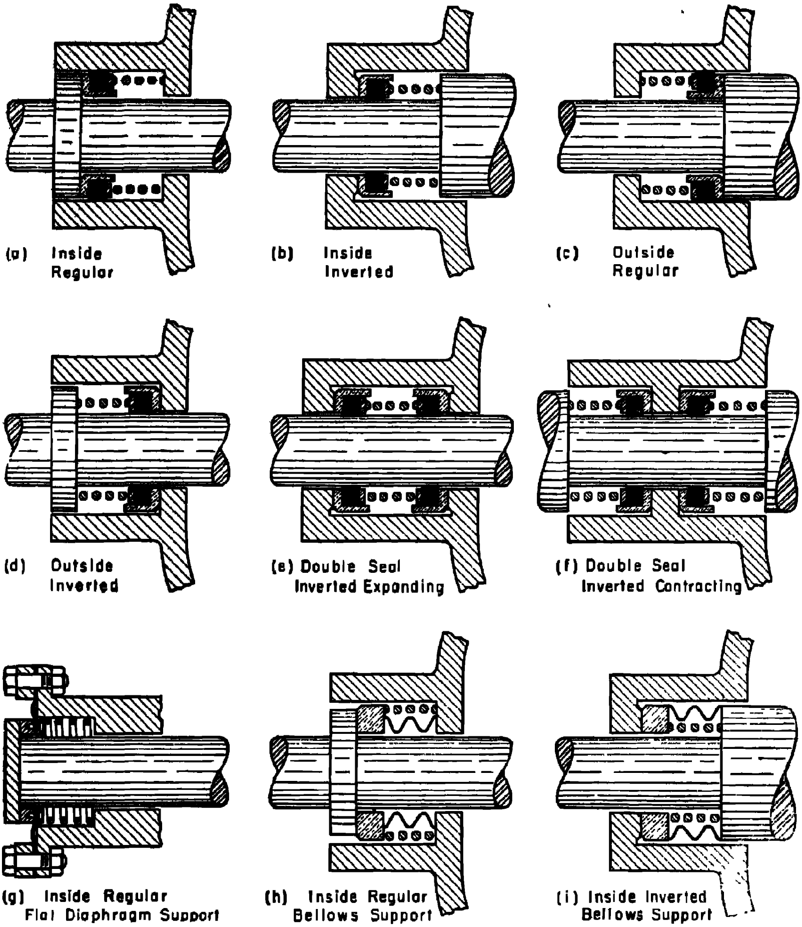

(a) Inside Regular
(b) Inside Inverted
(c) Outside Regular
(d) Outside Inverted
(e) Double Seal Inverted Expanding
(f) Double Seal Inverted Contracting
(g) Inside Regular Flat Diaphragm Support
(h) Inside Regular Bellows Support
(i) Inside Inverted Bellows Support

FIG. 19-33. Axial seals (see Table 19-5*C*).

Table 19-5. Classification of Rod, Plunger, or Shaft Packings (*Cont.*)

C. Axial seals

Type	Arrangement	Seal faces		Suspension	
		Stationary	Rotating	Type	Materials
1. Single inside 2. Single outside 3. Double expanding 4. Double contracting	1. Suspended seal stationary—"regular" 2. Suspended seal rotates—"inverted"	1. Carbon 2. Nonferrous metal (bronze) 3. Laminated phenolic 4. Teflon	1. Carbon 2. Nonferrous metal (bronze) 3. Laminated phenolic 4. Teflon	1. Sliding contact *a.* O ring *b.* V ring *c.* Wedge ring *d.* Packing 2. Flat diaphragm	As per Table 5*A* As per Table 5*A*

Table 19-5. Classification of Rod, Plunger, or Shaft Packings (*Cont.*)

C. Axial seals

Type	Arrangement	Seal faces		Suspension	
		Stationary	Rotating	Type	Materials
5. Tandem		5. Cast iron or steel 6. Cast iron or steel—hard-faced 7. Stainless steel 8. Stainless steel—hard-faced 9. Silicon iron	5. Cast iron or steel 6. Cast iron or steel—hard-faced 7. Stainless steel 8. Stainless steel—hard-faced 9. Silicon iron	3. Bellows	As per Table 5*A*

Table 19-6. Packing Recommendations for Utilities and General Services*

Service	Type of packed joint								
	Pumps, blowers, fans, and similar rotating shafts			Valve stems, expansion joints, floating heads of heat exchangers, etc.			Reciprocating rods of engines, pumps, compressors, hydraulic presses, and fluid motors		
Temperature	Up to 220°F	220 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	500 to 800°F
1. Water (boiler feed, condensate, drinking water, sea water, calcium chloride, brine)	645	....	....	645	657	...	625		
	Axial seal	....	....	....	....	...	7115		
	....	....	....	....	....	...	4151		
2. Oil (fuel, hydraulic, heat-transfer, lubricating)	558	558	548	558	558	548	7125	7225	
	559	559	Axial seal	559	559	...	4191		
	Axial seal	Axial seal							
3. Gasoline	648	....	....	648	....	...	7125		
	228	....	....	228					
	Axial seal								
4. Steam	9402	551–0	Labyrinth	645	657	657	......	551–0	561–0
	Labyrinth	558–0	....	....	7213	561	......	558–0	568–0
	Axial seal	9302	....	....	7215	568	......	7213	
	....	Labyrinth	....	....	....	...	......	9202–0	9202–0
	....	Axial seal							
5. Air	310	645	....	645	657	...	4151	7213	
	645	Labyrinth	....	....	....	...	7125–0	9202–0	
	Labyrinth	Axial seal							
	Axial seal								
6. Vacuum	558–0	....	....	645	....	...	9202–0	9202–0	
	559–0	....	....	....	....	...	558–0	558–0	
	Axial seal	....	....	....	....	...	559–0	559–0	
7. Ammonia	558–0	....	....	648	...	...	7112–0		
	559–0	....	....	7112	....	...	558–0		
	Axial seal	....	....	....	....	...	559–0		
	....	....	....	....	....	...	9102–0		
	....	....	....	....	....	...	9302–0		
8. Freon	558–0	....	....	648					
	559–0	....	....	104					
	Axial seal	....	....	240					
9. Dowtherm	....	548	548	...	548	548			

* As per classification Table 19-5*A*.

NOTE: "0" following packing recommendation denotes oil or grease seal.

Table 19-7. Packing Recommendations for Chemical Services*

Service	Type of packed joint								
	Pumps, agitators, and similar rotating shafts			Valve stems, expansion joints, floating heads of heat exchangers, etc.			Reciprocating rods of pumps and compressors		
Temperature	Up to 220°F	220 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	500 to 800°F
1. Water (neutral alkaline, or weakly acid solutions) Salt solutions and brines Ammonium nitrate Sodium carbonate Sodium chloride Sodium hypochlorite Sodium sulfite Trisodium phosphate Organic acids Dilute acetic Citric Lactic Carbonic Alkalies Caustic soda Caustic potash Aqua ammonia Slurries Miscellaneous Sugar solutions Molasses Dilute hydrogen peroxide	645 645–0 558–0 559–0 Axial seal	645 645–0 558–0 559–0 Axial seal	....	645 ...	657 ...	... ...	625 7115		
2. Food products Aqueous Fruit juices Beverages Pharmaceuticals Oils and emulsions Milk Mayonnaise	Axial seal	......	......	104	...	...	1043		
3. Mineral acids (with or without dissolved salts or suspended matter)									
Hydrochloric....	666 246 Axial seal	666 246 Axial seal	....	104 246 670	104 246 670	...	7325		
Hydrofluoric (aqueous)	246 Axial seal	246 Axial seal	....	104 246 670	104 246 670				
Hydrofluoric (anhydrous)	566–0 246 Axial seal	566–0 246 Axial seal	....	104 246 670	104 246 670				
Phosphoric Sulfuric Nitric (including mixtures with sulfuric)	666 246 Axial seal	666 246 Axial seal	....	104 246 670	104 246 670	..	666		

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* As per classification Table 19-5*A*.

Table 19-7. Packing Recommendations for Chemical Services (*Cont.*)

Service	Type of packed joint								
	Pumps, agitators, and similar rotating shafts			Valve stems, expansion joints, floating heads of heat exchangers, etc.			Reciprocating rods of pumps and compressors		
Temperature	Up to 220°F	200 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	500 to 800°F
4. Oils and solvents (with slight or no swelling effect on natural rubber or GR-S) Alcohols Methyl alcohol Ethyl alcohol Glycerine Ethylene glycol Aliphatic aldehydes and ketones Formaldehyde Acetaldehyde Acetone Methyl ethyl ketone Aliphatic amines Mono- or di- or triethylamine Mono- or di- or trimethylamine Aliphatic nitro compounds Nitromethane Nitroethane "Silicone" oils	645 648 Axial seal	... ...	... ...	104 645 648 670 240	... ...	... ...	7115 415		
(with slight or no swelling effect on Neoprene or Buna N)									
Aliphatic ethers Diethyl ether Dimethyl ether Cellosolve	648 Axial seal	... ...	... ...	648 104 670 240	...	...	7125		
Animal and vegetable oil Lard oil Cottonseed oil	558 559 Axial seal	558 559 Axial seal	... ...	558 559 ...	558 559 ...	... ...	7125 558 559	7225 558 559	
Aliphatic hydrocarbons									
Petroleum naphtha Gasoline	558 559 648 Axial seal	... ...	... ...	648 104 240 670	...	...	7125		
Kerosene Gas oil Fuel oil Lubricating oil Crude oil	558 559 Axial seal	558 559 Axial seal	548 Axial seal ...	558 539 ...	558 559 ...	548	7125 558 559	7225 558 559	548
Freon F-11, F-12, F-122	Axial seal	... ...	... ...	104 240 648	...	...	7125		

Table 19-7. Packing Recommendations for Chemical Services *(Cont.)*

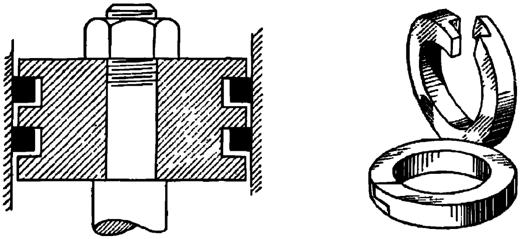
Service	Type of packed joint								
	Pumps, agitators, and similar rotating shafts			Valve stems, expansion joints, floating heads of heat exchangers, etc.			Reciprocating rods of pumps and compressors		
Temperature	Up to 220°F	220 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	500 to 800°F
4. Oils and solvents (*Cont.*)									
(with strong swelling effect on all rubbers)	648 Axial seal	.	. . .	104 240 648			648		
Phenols									
Phenol-anhydrous (carbolic acid)									
Cresol									
Aromatic aldehydes and ketones									
Benzaldehyde									
Acetophenone									
Cyclohexanone									
Aromatic ethers									
Dibenzyl ether									
Esters									
Methyl acetate									
Butyl acetate									
Aromatic or terpene hydrocarbons									
Benzene									
Toluene									
Xylene									
Coal tar									
Naphthalene									
Manufactured gas drip oil									
Turpentine									
Pinene									
Aromatic amines									
Aniline									
Mono- or di- or triethylaniline									
Mono- or di-butylaniline									
Di-phenylamine									
Benzidine									
Aromatic nitro compounds									
Nitrobenzine									
Di-nitrobenzine									
Mono- or di-nitro toluene									
Chlorinated hydrocarbons									
Carbon tetrachloride									
Chloroform									
Monochlorbenzene									
Trichlorethylene									
Freon F-21									
Carbon bisulfide									

Table 19-7. Packing Recommendations for Chemical Services (*Cont.*)

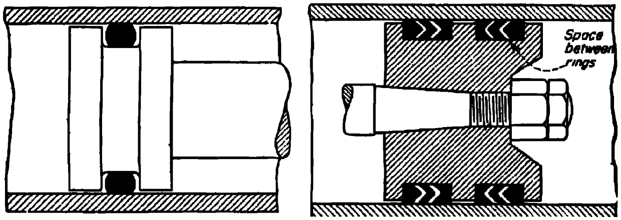
Service	Type of packed joint								
	Pumps, agitators, and similar rotating shafts			Valve stems, expansion joints, floating heads of heat exchangers, etc.			Reciprocating rods of pumps and compressors		
Temperature	Up to 220°F	220 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	500 to 800°F	Up to 220°F	220 to 500°F	500 to 800°F
5. Special liquids									
Liquid chlorine Liquid bromine Sulfur chloride	....	....	....	674 104					
Acetic acid (conc.)	645 Axial seal	... . .	... .	645					
Acetic anhydride . .	Axial seal	....	... .	645					
90 per cent hydrogen peroxide......	....	.. .	..	104					
Tin and titanium chloride...........	....	... .	... .	104					
6. Dry gases (with little effect on gasket materials)									
Acetylene Carbon monoxide Carbon dioxide Ethane Ethylene Hydrogen Helium Methane Natural gas Nitrogen Propane Producer gas Sulfur dioxide Hydrogen chloride	645–0 558–0 559–0	645–0 558–0 559–0	645–0 548–0	645	645	548 568 ...	9102–0 558–0 559–0	9102–0	
7. Special gases									
Oxygen...........	Labyrinth with inert gas seal.	....	....	104 240	... ..	...	9402		
Chlorine and bromine Sulfur chloride vapor Hydrogen fluoride (HF)	240 674	....	....	246 674	...	...	9402		

Table 19-8. Conventional Piston Packings

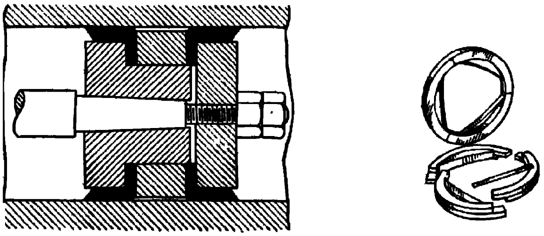
Type	Materials		Physical shape or method of fabricating
	Reinforcement	Binder	
1. Plain	0. None	1. Ebonite (natural rubber or GR-S) 2. Soft natural rubber 3. Neoprene 4. Buna N 5. Teflon	1. Rectangular cross section 2. Round cross section (O rings) 3. Nested V rings 4. Cups
2. Leather	1. Combination tannage leather 2. Chrome leather	1. Wax 2. Phenolic resin 3. Synthetic rubber	1. Nested V rings 2. Cups
3. Fabric	1. Cotton duck 2. Fine cotton fabric (sheeting or drill) 3. White-asbestos fabric	1. Semihard natural rubber or GR-S 2. Soft natural rubber or GR-S 3. Neoprene 4. Buna N 5. Phenolic resin	1. Rectangular cross section 2. Nested V rings 3. Cups 4. Segmental cut rings
4. Solid metal, carbon, or graphite	1. Cast iron 2. Bronze 3. Graphite	0. None	1. Split ring 2. Segmental cut rings



(a) Split piston rings (b) Split piston ring with step joint



(c) O ring (d) V rings



(e) Cups (f) Segmental cut

Fig. 19-34. Piston packings (see Table 19-8, Types 1 to 4).

Table 19-9. Piston-packing Recommendations for Utilities and General Services

Service	Type of packed joint	
	Inside packed pistons of pumps, compressors, hydraulic presses, fluid motors, etc.	
Temperature	Up to 220°F	220 to 500°F
1. Water (boiler feed, condensate, drinking water, sea water, calcium chloride brine)	2112 3211 3113	
2. Oil (fuel, hydraulic, heat transfer, lubricating)	1042 2121 2122 3133 4101 3151 2132	3151 4101
3. Gasoline	3151	
4. Steam	......	4101
5. Air	4101–0 2122–0 3133–0	4101–0
6. Vacuum	4101–0 2122–0 3133–0	4101–0
7. Ammonia	4101–0	

Table 19-10. Piston-packing Recommendations for Chemical Services

Service	Type of packed joint	
	Inside packed pistons of pumps, compressors, hydraulic presses, fluid motors, etc.	
Temperature	Up to 220°F	220 to 500°F
1. Water (Neutral, alkaline, or weakly acid solutions) Salt solutions and brines Ammonium nitrate Sodium carbonate Sodium chloride Sodium hypochlorite Sodium sulfite Trisodium phosphate Organic acids Dilute acetic Citric Lactic Carbonic Alkalies Caustic soda Caustic potash Aqua ammonia Slurries Miscellaneous Sugar solutions Molasses Dilute hydrogen peroxide	3211 3113	

Table 19-10. Piston-packing Recommendations for Chemical Services (*Cont.*)

Service	Type of packed joint Inside packed pistons of pumps, compressors, hydraulic presses, fluid motors, etc.	
Temperature	Up to 220°F	220 to 500°F
2. Oils and solvents		
(With slight or no swelling effect on natural rubber or GR-S)	3211 3113	
Alcohols		
Methyl alcohol		
Ethyl alcohol		
Glycerine		
Ethylene glycol		
Aliphatic aldehydes and ketones		
Formaldehyde		
Acetaldehyde		
Acetone		
Methyl ethyl ketone		
Aliphatic amines		
Mono- or di- or triethylamine		
Mono- or di- or trimethylamine		
Aliphatic nitro compounds		
Nitromethane		
Nitroethane		
Silicone oils		
(With slight or no swelling effect on Neoprene or Buna N)	3151 3113	3151
Aliphatic ethers		
Diethyl ether		
Dimethyl ether		
Cellosolve		
Animal and vegetable oil		
Lard oil		
Cottonseed oil		
Aliphatic hydrocarbons		
Crude oil		
Petroleum naphtha		
Gasoline		
Kerosene		
Gas oil		
Fuel oil		
Lubricating oil		
Freon		
F-11, F-12, F-122		
(With strong swelling effect on all rubbers)	3151	3151
Phenols		
Phenol-anhydrous (carbolic acid)		
Cresol		
Aromatic aldehydes and ketones		
Benzaldehyde		
Acetophenone		
Cyclohexanone		
Aromatic ethers		
Dibenzyl ether		
Esters		
Methyl acetate		
Butyl acetate		
Aromatic or terpene hydrocarbons		
Benzene		
Toluene		
Xylene		
Coal tar		
Naphthalene		
Manufactured gas drip oil		
Turpentine		
Pinene		

Table 19-10. Piston-packing Recommendations for Chemical Services (*Cont.*)

Service	Type of packed joint	
	Inside packed pistons of pumps, compressors, hydraulic presses, fluid motors, etc.	
Temperature	Up to 220°F	220 to 500°F
2. Oil and solvents (*Cont.*)		
Aromatic amines	3151	3151
Aniline		
Mono- or di- or tri-ethylaniline		
Mono- or di-butylaniline		
Di-phenylamine		
Benzidine		
Aromatic nitro compounds		
Nitrobenzine		
Di-nitrobenzine		
Mono- or di-nitro toluene		
Chlorinated hydrocarbons		
Carbon tetrachloride		
Chloroform		
Monochlorbenzene		
Trichlorethylene		
Freon F-21		
Carbon bisulfide		
3. Dry gases		
(With little effect on packing materials)	4101 4102	4101 4102
Acetylene		
Carbon monoxide		
Carbon dioxide		
Ethane		
Ethylene		
Hydrogen		
Helium		
Methane		
Natural gas		
Nitrogen		
Propane		
Producer gas		
Sulfur dioxide		
Hydrogen chloride		

DIAPHRAGMS

A packing diaphragm spans a gap between a moving and a stationary member, permitting limited motion in the former. It makes a fluid-tight contact with both members, with or without supplementary gaskets.

Since a diaphragm functions in the same manner, except for absence of leakage, as a sliding-contact packing, there are diaphragm equivalents for all the various types of sliding-contact packings. The diaphragm in the air motor (Fig. 19-35) may be looked upon as the equivalent of a reciprocating piston packing. Such motors are used to operate valves, switches, or perform any other simple mechanical act such as to apply brakes, press garments, or operate a recording instrument. Fluid pressure, in the case of a motor valve, may be derived from the fluid in the line or from air or other fluid supplied by a pilot valve which is itself operated by the fluid in the line or otherwise. If the

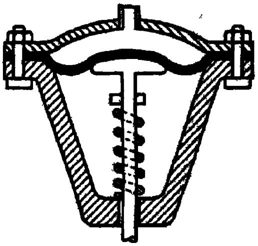


Fig. 19-35. Air-motor diaphragm.

fluid is too destructive to the diaphragm material, a suitable siphon of harmless fluid may be interposed. Thus, in the case of steam-operated diaphragm motors, direct contact of the diaphragm with steam is avoided by having a loop in the connecting pipe which fills with condensate.

A list of diaphragm packings is given in Table 19-11.

Table 19-11. Classification of Diaphragm Packings

Type	Materials		Physical shape or method of fabricating
	Reinforcement	Binder	
1. Plain	0. None	1. Natural rubber (or GR-S) 2. Neoprene 3. Buna N	1. Sheet (or diaphragms cut from sheet) 2. Molded with dished or troughed contour 3. Molded with cylindrical corrugations (bellows)
2. Fabric	1. Cotton 2. Nylon 3. Glass	1. Natural rubber 2. Neoprene 3. Buna N 4. Silicone rubber	1. Sheet (or diaphragms cut from sheet) 2. Molded with dished or troughed contour
3. Metal	1. Carbon steel 2. Stainless 3. Bronze 4. Aluminum	0. None	1. Sheet (or diaphragms cut from sheet) 2. Flat corrugated 3. Cylindrical corrugated (bellows)
4. Leather	1. Sheepskin	1. Animal or vegetable oil	1. Hides (or diaphragms cut from hides) 2. Molded with dished or troughed contour

Sheet-metal diaphragms are used for diaphragm motors at high pressures. Flat corrugated metal is used principally for aneroid barometers and thermostats. Metal bellows (Fig. 19-36) are widely used in refrigerator control valves. They are available in a limited number of sizes and need to be gasketed or soldered to adjacent parts. They have the advantage of long travel, which, however, must not exceed the elastic limit of the metal, or early failure will occur by fatigue. Flat leather diaphragms are largely used in gas meters and regulators because of their resistance to benzol. Flat fabric–rubber diaphragms are used for general service. A few precautions to be observed in their use are shown in Fig. 19-37. Flat fabric diaphragms are capable of only about 2 in. travel per ft of diameter (1 in. either side of the plane of the rim), and in an effort to increase this, diaphragms are sometimes bought premolded to various contours of which the simple dished contour (Fig. 19-38) is the best. The advantages of molded diaphragms over flat diaphragms in respect to travel is very slight, however, and is frequently lost because of concealed damage to the fabric in the molding operation. Plain rubber is used when pressure differentials are very low.

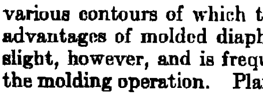
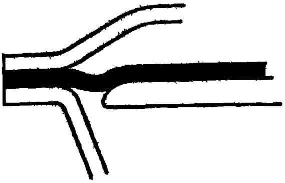


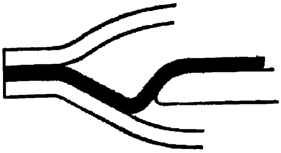
Fig. 19-36. Metal bellows.

The cover compound on most fabric diaphragms is based on natural rubber or GR-S, but oil resistant synthetics are used for services against oil or against air containing oil, as in "garage-type" air-compressor switch motors or in the unloading mechanism of large air compressors. Covers of oxidized oil (oilcloth), nitrocellulose, etc., were used at one time for solvent service but have been largely displaced by the synthetic rubbers. Covers of silicone rubber are of interest where extremely hot air is involved.

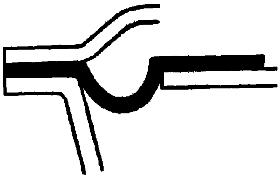
Corresponding to the piston pump is the diaphragm pump. Pump diaphragms are usually special because of the severe service and possibility of early fatigue failure. Small diaphragm pumps are used to pump fuel to the automobile engine (Fig. 19-39). A variant is the liquid-piston diaphragm pump (Fig. 19-40) in which the diaphragm serves merely as a separator and is under no stress. Plain rubber, therefore, can be used. Similar diaphragm separators are employed with air accumulators on oildraulic systems which function alternately as pumps and as motors. There are also



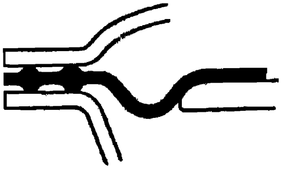
Don't make free space narrow



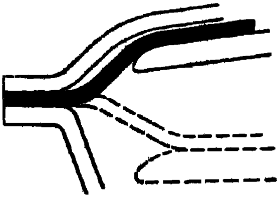
Don't make case too shallow



Don't leave sharp angles



Don't use beaded rims
(or beaded flanges)



Don't use all the slack

IG. 19-37. Precautions in the se of diaphragms.

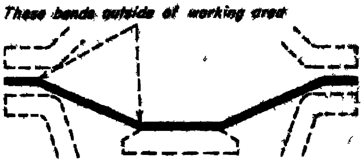


FIG. 19-38. Molded diaphragm.

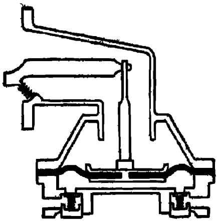


FIG. 19-39 Diaphragm pump (automobile)

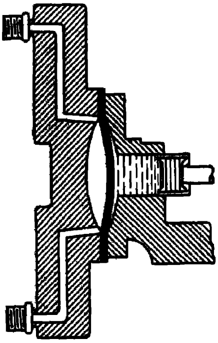


FIG. 19-40. Liquid-piston diaphragm pump.

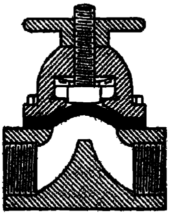


FIG. 19-41. Saunders valve.

diaphragms used as air-compressor pistons on portable spray outfits. This service is severe not only because of the speed of operation but also because of oxidation.

Diaphragms are also used as valve-stem packings, usually in the form of bellows, as in the so-called "packless" radiator valves. In the Saunders valve (Fig. 19-41), the diaphragm functions simultaneously as a valve-stem packing and as a valve disk. Diaphragms of many different materials are available to combat various fluids.

Axial seals employing diaphragm suspensions, both flat and cylindrical, have been mentioned under Rotary Shaft Packings. The domestic refrigerating-machine crankcase seal was, until recently, the principal market for the metal bellows. Some of the bellows used on water-pump seals, as in the familiar automobile-radiator pump, are of plain rubber, the pressures being insufficient to necessitate any reinforcement.

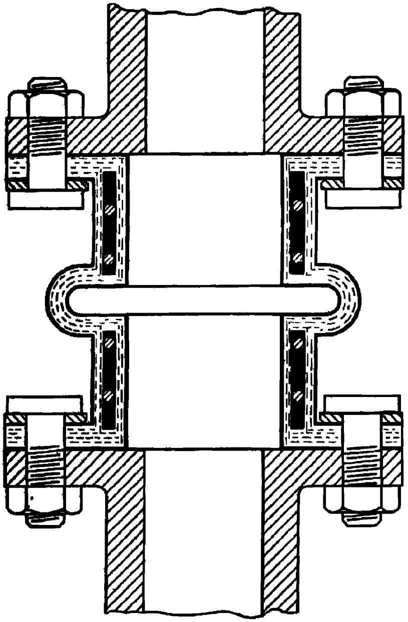


FIG. 19-42. Bellows expansion joint.

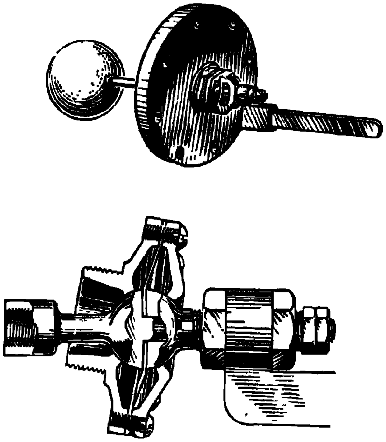


FIG. 19-43. Diaphragm-packed float valve.

Other uses for packing diaphragms are (1) the bellows expansion joint (Fig. 19-42), replacing the usual sliding-contact type, and (2) the diaphragm-packed float valve (Fig. 19-43). These permit rocking around an axis in addition to straight push and pull. The familiar Bourdon tube may be looked upon as a motor diaphragm adapted to produce rotary motion. Bellows with helical corrugations have been employed for the same purpose.

PACKING TESTING AND EVALUATION

Packing tests include field tests, bench tests, and laboratory tests.

Field Tests. In the last analysis, there is no satisfactory way to evaluate a packing except to observe its performance under service conditions. The packing bill of any large process industry is sufficient to warrant employing an engineer with nothing to do but to supervise packing performance. Such an individual should have at his disposal a card index with a card for each important packing application, covering complete dimensional data, operating conditions, and a record of the life of each successive packing installed. An attempt should be made to determine the causes of packing failure by examination of removed material and by periodic examination of shafts, cylinder liners, etc., for scoring, excessive clearances, misalignment, and weave or gyration. Although such supervision is installed primarily to differentiate between packings of various suppliers, its greatest benefit consists in the prolonged life conferred on all packings by closer attention to the mechanical condition of equipment, to the installation of seals and other desirable accessories, and to instruction of the operating force. In smaller organizations the functions of packing engineer, lubricating engineer, and power-transmission engineer are frequently combined.

Packings for original equipment are tested as part of the "green run" to which the equipment is subjected on the assembly floor of the supplier's plant. Unfortunately these "green runs" are frequently of too short duration to tell much about

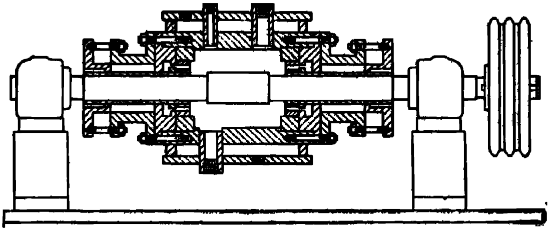


Fig. 19-44. Cross section of two-cavity bench tester.

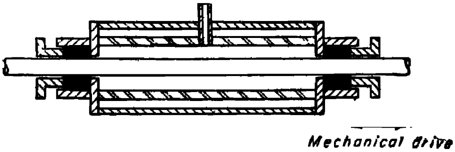


Fig. 19-45. Balanced-type tester (one type).

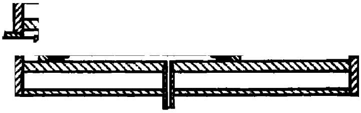


Fig. 19-46. Balanced-type tester (another type).

packing performance, especially as most packings do not exhibit their best performance until after several hours or days of continuous running. Performance should therefore be checked by tests run to destruction on a small portion of the product.

Bench Tests. For those who have the time and money to spend in accurate packing evaluation, satisfactory results free from personal error can be secured on laboratory bench machines set up to duplicate service. Figure 19-44 shows a cross section of a two-cavity bench tester suitable for all types of rotary shaft packings. Any medium at any pressure and temperature can be introduced into the interior, any speed of drive can be employed, and any degree of shaft runout can be induced by wear sleeves of predetermined eccentricity. Leakage is collected and measured, and packing adjustments as necessary are made to maintain a predetermined leakage rate. Stuffing boxes can be modified to permit tests on radial or axial seals. In Figs. 19-45 and 19-46 are shown "balanced-type" testers suitable for testing reciprocating rod packings and piston packings, respectively. In Fig. 19-47 is shown a device for continuously opening and closing valves—it can be employed to test valve-stem packings or valve disks. In Fig. 19-48 is shown a device for flexing diaphragms

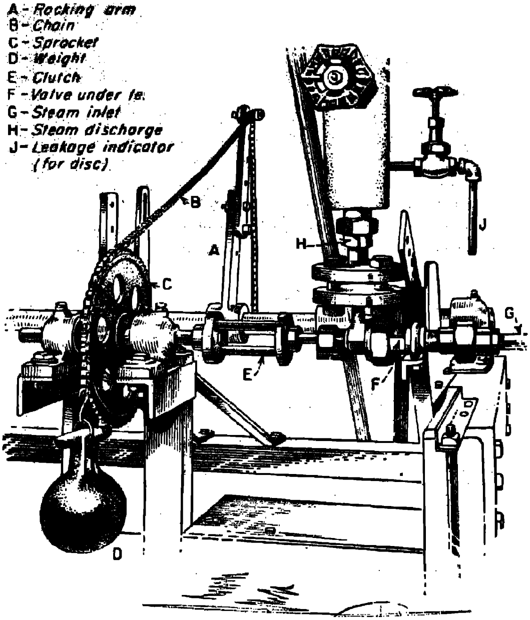


FIG. 19-47. Device for continuously opening and closing valves.

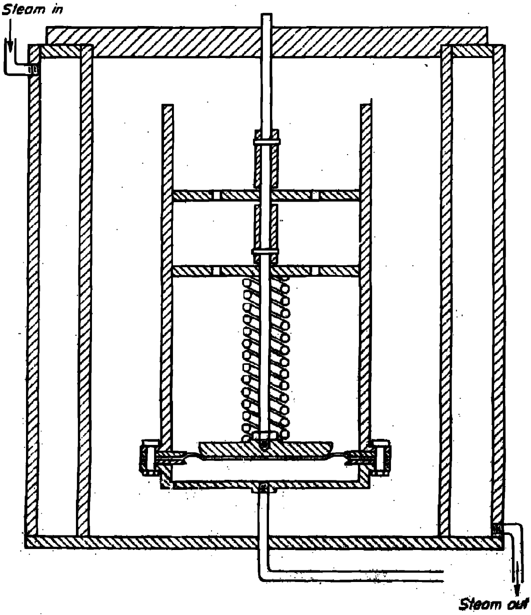


FIG. 19-48. Device for flexing diaphragms to destruction.

to destruction. The device is so constructed that the diameters of piston and cylinder, the radii of the corners, the stroke, the nature and pressure of the actuating medium, and the ambient temperature can all be modified at will. All these dynamic testers can be made to indicate packing friction or record it continuously. In Fig. 19-49 is

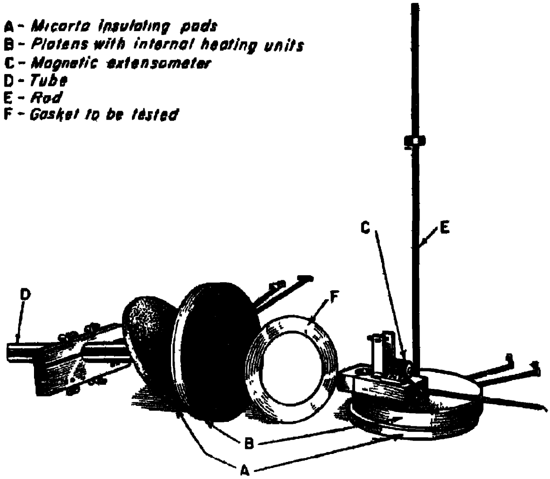

A - Micarta insulating pads
B - Platens with internal heating units
C - Magnetic extensometer
D - Tube
E - Rod
F - Gasket to be tested
C
E
D
F
B
A

Fig. 19-49. Flange tester for gaskets.

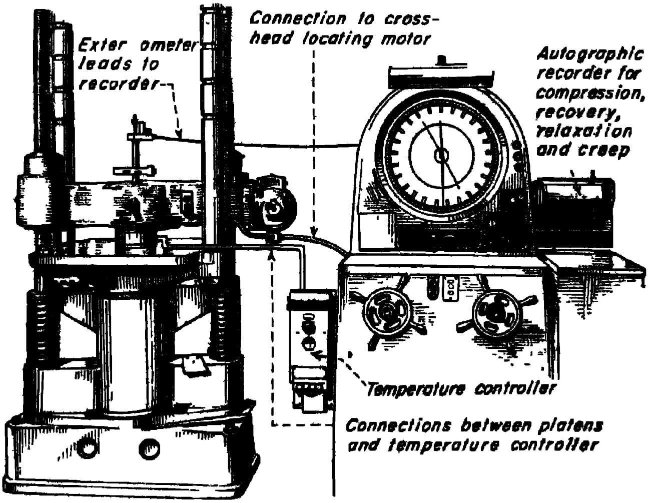

Extensometer leads to recorder
Connection to cross-head locating motor
Autographic recorder for compression, recovery, relaxation and creep
Temperature controller
Connections between platens and temperature controller

Fig. 19-50. Flange tester installed in a heavy-duty test machine.

shown a flange tester for gaskets, and in Fig. 19-50 it is shown installed in a heavy-duty test machine. This device can be used to record gasket compression, creep, and relaxation under practically all conditions of fluid medium, pressure, temperature, and condition of flange surfaces. In addition to their value for evaluating competitive packings, machines such as above described are invaluable as research tools for exploring the effect of varying operating conditions on packing performance.

Laboratory Tests. Laboratory tests for plain rubber packing include the following:

Tensile and Elongation Tests. Tensile and elongation tests are conducted on dumbbell strips on a test machine of the type shown in Fig. 19-51. Details for conducting the test are given in ASTM-D-412. Where the dimensions of the packing do not permit cutting the standard test specimen, approximate value can be obtained from nonstandard specimens. For instance, O rings can be tested over spool jaws.

Set Tests. Set tests include tension-set and compression-set determinations. Both involve the exposure of a strained specimen to a predetermined temperature for a fixed time and the measurement of the resultant recovery. Tension-set tests are conducted on dumbbell (or equivalent) specimens in accordance with ASTM-D-412. Compression-set tests are conducted on cylindrical specimens which in certain cases may be plied up from thinner material. Method is given in ASTM-D-395.

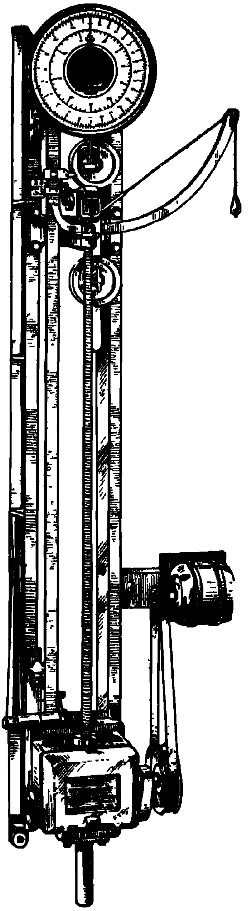


FIG. 19-51. Tensile and elongation test machine.

Hardness Tests. Hardness tests involve measurement of the penetration of a point of fixed dimensions under a predetermined load. Spring-loaded portable instruments are generally known as "durometers" and record penetration on an inverted scale, ranging from 100 for a rigid object down to about 20 for a very soft rubber. Method for calibrating and using these is given in ASTM-D-676. The ASTM penetration tester (sometimes known as the Olsen tester) and the plastometer (Pusey & Jones) are bench machines employing dead-weight loading. They are less convenient than the durometers but more accurate in that they do not require calibration. Standard methods for employing these are given in ASTM-D-314 and D-513, respectively.

Hot-air Aging Tests. Hot-air aging tests involve exposure of a tensile or hardness specimen to circulating hot air, the effect being measured by resultant changes in tensile, elongation, or hardness. Methods are given in ASTM-D-573. Closely related are the oxygen-bomb and air-bomb aging tests which are described in ASTM-D-572 and D-454.

Solvent or Oil-immersion Tests. Solvent or oil-immersion tests involve immersion of the test specimen in a specified solvent or in lubricating oil for a predetermined time and temperature and measuring the resultant changes in volume, tensile strength, elongation, or hardness. Methods are given in ASTM-D-471.

Modification of above tests applicable to compressed asbestos sheet are given in ASTM-D-733. Packings constructed from plied-up fabric can be tested for ply adhesion in accordance with ASTM-D-413. Standard methods are also available for testing paper, leather, yarn, metals, etc. In addition, any of the recognized chemical methods for the separation and identification of ingredients may be applied to packings of any type.

A point to be observed in connection with all laboratory tests is that they do not by themselves supply a measure of packing serviceability. They are employed legitimately as a means of demonstrating the uniformity of successive shipments of a packing which has been originally approved on the basis of field or bench tests.

SECTION 20

MATERIALS HANDLING

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CONVEYORS, ELEVATORS, PALLETS, AND FORK TRUCKS

BY R. F. BERGMANN

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INTRODUCTION

The handling of materials, either in their raw state or as semifinished or finished products, is a large and important problem in modern industry. Efficient handling of materials is the cornerstone in the layout of plants in the mass-production industries. Within a plant the movements of material should be as few and as short as conditions will permit. Some layouts lend themselves to the use of gravity for moving the materials, decreasing the amount of machinery required.

There are many different devices for elevating and conveying materials. Solving a particular problem requires the selection of a device that experience has shown to be suitable and one whose initial and maintenance costs are consistent with the amount of work done.

BULK MATERIALS

There are many difficulties which arise in the handling of bulk materials which are not present in the conveying of units, pallets, or packaged goods. Study should be given to the following factors: possible degradation of the material, discoloration and impurities, abrasive effects, and dust produced with respect to its injurious effects on workers and the possibility of explosion and fire.

The handling of bulk materials has been divided into two parts: unloading and conveyance. For a brief comparison of the types of elevators, see Table 20-1, and of conveyors, see Table 20-2.

Table 20-1. Bucket Elevators

	Centrifugal discharge	Perfect discharge	Continuou bucket	Gravity discharge	Pivoted bucket
Carrying paths	Vertical	Vertical to inclination 15 deg from vertical	Vertical to inclination 15 deg from vertical	Vertical and horizontal	Vertical and horizontal
Capacity range, tons per hr, material weighing 50 lb per cu ft	78	34	345	191	255
Speed range, fpm	306	120	100	100	80
Location of loading point	Boot	Boot	Boot	On lower horizontal run	On lower horizontal run
Location of discharge point	Over head wheel	Over head wheel	Over head wheel	On horizontal run	On horizontal run
Handling abrasive materials	Not preferred	Not preferred	Recommended	Not recommended	Recommended

Table 20-2. Conveyors

	Belt conveyor	Apron conveyor	Flight conveyor	Drag chain	En masse conveyor	Screw conveyor	Vibratory conveyor
Carrying paths	Horizontal to 18 deg	Horizontal to 25 deg	Horizontal to 45 deg	Horizontal or slight incline, 5–10 deg	Horizontal to 90 deg	Horizontal to 15 deg, may be used up to 90 deg but capacity falls off rapidly	Horizontal or slight incline, 5 deg above or below horizontal
Capacity range, tons per hr, material weighing 50 lb per cu ft	2 160	100	360	20	100	150	100
Speed range fpm	600	100	150	20	80	100 rpm	40
Location of loading point	Any point	Any point	Any point	Any point	On horizontal runs	Any point	Any point
Location of discharge point	Over end wheel and intermediate points by tripper or plow	Over end wheel	At end of trough and intermediate points by gates	At end of trough	1 Any point on horizontal runs by gate 2 Head of vertical runs	At end of trough and intermediate points by gates	At end of trough
Handling abrasive materials	Recommended	Recommended	Not recommended	Recommended with special steels	Not recommended	Not preferred	Recommended

UNLOADING

Railroad-car Unloaders

Track Hoppers. Track hoppers are used principally for materials shipped in bottom-dump cars, where requirements are not over 2 to 2½ tons per min of material weighing 50 lb per cu ft. The concrete foundation may be shaped to form the hopper, or separate steel hoppers may be installed below the track level. For effective dumping with a minimum of spill, it is recommended that the hopper be not less than 12 ft wide. A hopper 12 ft long will accommodate a 50-ton two-door car; while for a 70-ton four-door car (and for more effective discharge of a 50-ton car), the track hopper should be 22 ft long. To reduce depth of pit required and the cost involved for a 22-ft-long track hopper, it generally is better to use two 12- by 12-ft units placed end to end.

The speed of discharge of a car will vary greatly with the condition of the material at the time the car is unloaded. Fines mixed in with the load will often pack down during transit requiring the load to be loosened by barring or by striking the side of the car to vibrate the load loose. Packed and partially frozen material can easily be removed by means of mechanical shaker units. Where the load is wet or frozen, the time of unloading may be as much as doubled, even with additional laborers on the job. Mechanical devices such as traveling augers, shakers, and air-hammer spuds may be used to accelerate the dropping of frozen material. Under good conditions a car may be unloaded in 10 to 30 min, depending upon the size of car, size of hopper, characteristics of the material, and number of laborers assisting the operation.

Cars may be spotted over the hopper by an industrial locomotive of the diesel, steam, fireless-steam, compressed-air, or electric type or by a capstan car puller. The car puller is ideal for intermittent operation since the electric-motor drive need be operated only while the car is being spotted. Car movement is facilitated if the track is on a slight downgrade.

The capacity of the elevator away from the track hopper should be great enough to provide sufficient room in the hopper for the dumping of the succeeding car, but not so large that the elevator will run empty an appreciable portion of the time.

Crane Unloaders. A crawler, locomotive, or truck crane, available from other duties, is sometimes used with a clamshell or orange-peel bucket to unload gondolas. Flat-bottom cars should be available, for the bucket cannot readily unload hopper-bottom cars. A track hopper leading to a pit at the side of the track is used where the type of car cannot be controlled, and the crane may handle either from the car or from the pit.

Cranes are particularly adapted for temporary installations where unloading may be done into trucks, overhead bins, or storage piles. Some hand labor is required in the cleaning out of cars. Materials can also be received in bulk containers. In this case the units are usually removed by means of a monorail or bridge crane.

Rotary Car Dumpers. In cases where the required capacity is greater than that which can be handled with a track-hopper installation, a rotary car dumper (see Fig. 20-1) is used. While the initial cost is considerably greater than a track hopper, the operational cost per ton is low. Two or three unskilled laborers are all that is required, and the device is so simple that one of these laborers can handle the operational machinery.

The mechanism consists of two portal end rings with a cradle supported between them. The end rings are rotated on pairs of supporting wheels at each end of the dumper. Cable-operated clamps lock the car to the rails by pressure on the top of the car during the dumping operation.

The usual installation employs a switch engine to spot the cars on the dumper. Cars leaving the dumper are pushed onto an inclined track by the incoming car and are gravity-propelled to a storage yard. The capacity per hour will depend largely on the switching facilities available and may run as high as 30 cars an hour. A full dumping cycle takes approximately 1½ min. A frozen load will require slightly longer. These units cannot be used where material is received in a frozen state unless provisions are made for melting down the mass before dumping. The dumping time is not affected by sticking of the bottom gates. Units will handle cars of capacities up to 120 tons.

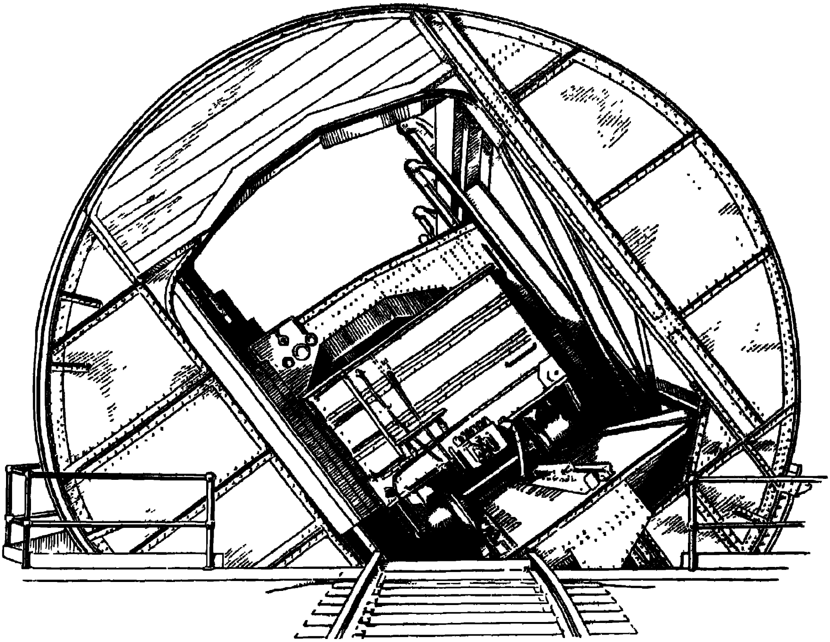

Fig. 20-1. Rotary car dumper.

Power Shovel. Where bulk material is to be unloaded from boxcars and the unloading-capacity requirements are moderate, a power shovel will efficiently perform the operation (see Fig. 20-2). The shovel is drawn into the car by the operator, whereupon a slight slackening of the rope from the shovel to the motor-operated drum serves to trip the mechanism engaging a clutch. The drum winds the cable pulling the shovel to the car door, at which point the clutch is automatically disengaged. At the door the material is discharged from the shovel into a hopper feeding the conveyors required to carry off the material.

From the description it can be seen that the power-shovel operation is fatiguing and hazardous. It, therefore, is suggested that a gasoline-powered scoop shovel be considered for these operations.

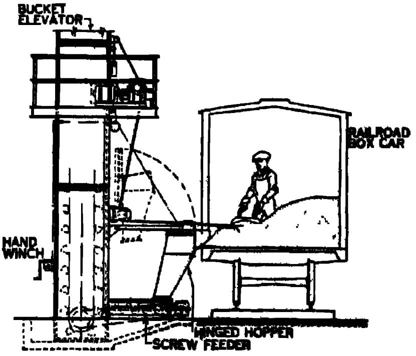

Fig. 20-2. Unloading box car by power shovel.

Pneumatic Unloaders. The pneumatic system, operating on the same principle as the vacuum cleaner, is particularly suitable for dusty materials deleterious to operators. The car-door bulkheads can remain in place, and the operator works from the outside except for final cleaning. A cyclone separates the material at the storage point, no elevators or conveyors being required, with the air usually discharged through a bag filter. Pneumatic units can be used efficiently with materials received in covered hopper cars with considerable savings in labor cost for unloading.

The unloading capacity will depend on the physical properties of the material in addition to the amount of air pumped and the difference in air pressure between the two ends of the system. The capacity will generally be greater than that of a power shovel, and the horsepower requirements will be higher. With high air pressures, abrasive materials may cause excessive wear at turns, and degradation of fragile materials may result. However, proper materials of construction, such as rubber lining or any of the many abrasion-resisting alloys, can solve these problems.

Packed or dense materials, such as cement, can be unloaded through the use of a Fuller-Kinyon pump. In the portable self-contained unit, a horizontal disk feeder delivers the material to a choke screw feeder in which it is aerated and fed into the conveying line, the air and material mixture flowing somewhat like a liquid.

Box-car Dumpers. For large capacities of grain or soybeans a box-car dumper is usually employed (see Fig. 20-3). It consists of a structural-steel cradle mounted on rollers at four points so as to permit endwise tipping in either direction to 40 deg from the horizontal and a car supporting platform pivotally mounted to permit sidewise tipping to an angle of 15 deg in one direction. The car is switched onto the supporting platform, then two end clamps engage the car couplers. The grain-door opening frame is brought into contact with the car door and locked against backward movement. The car is then tipped sidewise against the frame. This frees the car doors. The cradle is then tipped endwise, first in one direction, then in the other, removing the majority of the grain. A baffle plate is inserted diagonally across the car floor on the last endwise tip to remove the grain that might otherwise run past the door opening.

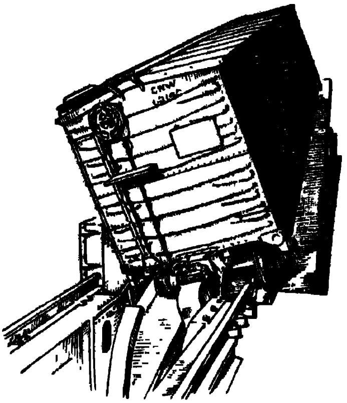

Fig. 20-3. Box-car dumper

The car platform returns the empty car to a horizontal position, the door opener is withdrawn, the end clamps are disengaged, and the platform is automatically locked. The empty car is then ready to be moved off the platform.

The time required for the complete cycle of operation is 4½ min. Under favorable conditions eight cars per hour can be unloaded with two men, or four cars per hour with one man.

Boat Unloaders

Installation of boat unloaders is very specialized, with each unit usually being designed for the particular job. Because the average plant does not receive bulk material direct from boats, only a brief mention of the types available is given.

Clamshell Buckets. For average capacity requirements or intermittent service, a clamshell bucket working from a monorail-system gantry crane or cantilever bridge with tramway hoist will prove satisfactory. The unit may discharge into cars along the wharf, into stockpile, or into a conveying system.

Hulett Unloader. The Hulett unloader consists of a gantry frame with a vertical leg having both vertical and rotating movements. A clamshell type of bucket operates at the lower end of this leg. The operator rides in the lower end of this leg where he is close to the work. Discharge from the bucket is into a hopper and weighing larry from where it may go into railroad cars or conveyor system. Capacities with a 17-cu-yd bucket are from 1,000 to 1,500 tons per hr of iron ore weighing 150 lb per cu ft.

Self-unloading Boats. With self-unloading boats, discharge from the boat to the wharf is over a belt conveyor mounted on a pivoted arm which may be swung to either side of the boat and raised or lowered. Cargo comes from the hold on a bucket elevator or belt conveyor which in turn is fed either by belt conveyors or by drag scrapers operating in tunnels at the bottom of the holds. Unloading capacities are in the range of from 1,000 to 2,000 tons per hr.

Pneumatic Unloaders. Grains are unloaded at the rate of 6,500 bu per hr by suction systems.

CONVEYANCE

Bulk Elevators

Skip Hoists. Skip hoists (Fig. 20-4) may, in general be classified into one of three types:

1. Single bucket without traveling counterweight
2. Single bucket with traveling counterweight
3. Two buckets operating reciprocally

The first type, in the simplest form, will consist of a rectangular bucket running on vertical or inclined tracks (or a combination of the two) with a hoisting cable and hoisting machine. Loading at the foot is usually from a hopper, but a conveyor intermittently operated is sometimes used. At the top of its travel, the bucket is usually tipped to discharge the material.

The use of a counterweight heavy enough to equal the dead weight of the bucket plus half that of the load will reduce the maximum motor load more than half, but the motor will be loaded in both directions of travel. A common method of connecting the counterweight is to have the counterweight cable wound on the same drum as the bucket cable but in the reverse direction.

In nonautomatic operation the stopping of the bucket at the top and bottom is controlled by the hoist operator. In the semiautomatic device the operator starts the hoist, which is automatically stopped at the top of the bucket travel, and after a time lag sufficient to allow the bucket to empty completely, the hoist is automatically reversed to return the skip to the starting position. It must now be restarted by the operator before another round trip is made.

In fully automatic operation, the skip continues to repeat its cycle until stopped by the operator. This requires some method of automatic loading, which may be one of four types:

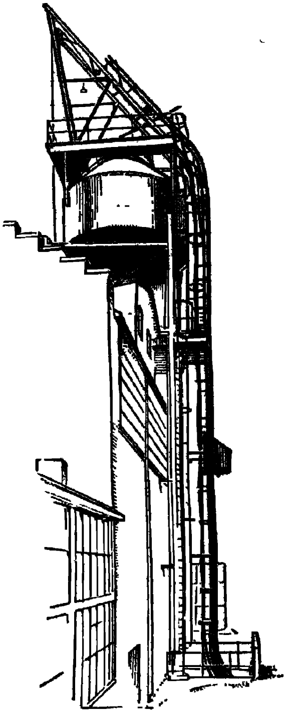

Fig. 20-4. Skip hoist.

1. The measuring type, in which the bucket is loaded with a predetermined volume of material before it recommences its upward trip
2. The weight type, in which the bucket is loaded with a predetermined weight of material before it starts its upward trip
3. The full-bucket type, in which the bucket will start upward only after it has been filled
4. The time-lag type, in which the bucket remains at the bottom a sufficient length of time to allow it to fill, overflowing of a bucket being prevented by the damming effect of the material as it reaches its angle of repose (this has the disadvantage that, if the hopper should empty or the spout jam, the bucket will continue to run whether it is loaded or not)

In the stock-piling of material a variation of the skip hoist is used in which the bucket, at the top of its travel on the rails, is lowered until it touches the top of the stock pile. This releases gates in the bottom of the bucket to dump the load, and there is no free fall of material with accompanying degradation.

The skip hoist may have a bucket capacity of 20 to 200 cu ft and will operate at a cable speed of 40 to 450 fpm. In addition to these factors, the tons of material handled per hour will depend upon the distance of lift and the weight per cubic foot of the material. Table 20-3 gives the capacities with fully automatic operation.

The skip hoist is dependable, easy to operate, has few moving parts, and for high lifts or large capacities, the first cost will be less than for any other type of mechanical conveyor. It is particularly adaptable for the handling of abrasive or corrosive materials or those which are either fine or lumpy.

Bucket Elevators. Under this heading the following types of bucket elevators are discussed: centrifugal discharge, perfect discharge, gravity discharge, continuous bucket, and the pivoted bucket carrier. For a brief comparison of these types see Table 20-1.

Table 20-3. Capacities of Skip Hoists—Automatic Loading, Fully Automatic Operation

Speed of bucket, fpm	Lift, ft	Tons per hr, material weighing 50 lb per cu ft					
		Bucket size, cu ft					
		40	60	80	100	150	200
80 (single speed, one bucket)	40	50	75	95	120	185	245
	60	35	50	65	80	125	165
	80	25	40	55	65	100	135
	100	22	30	45	55	85	110
	120	18	25	35	45	70	95
	160	14	21	30	35	55	70
	200	11	17	23	29	45	55
140 (single speed, one bucket)	40	70	105	140	175	265	350
	60	50	75	100	125	185	250
	80	40	65	85	105	155	210
	100	35	50	70	85	130	170
	120	30	45	60	75	110	150
	160	22	35	45	55	85	110
	200	18	27	35	45	65	90
260 (two speeds, one bucket)	80	60	90	120	150	225	300
	100	50	75	100	125	190	255
	120	45	65	90	110	165	225
	160	35	55	75	85	135	185
	200	30	45	60	75	115	155
260 (two speeds, two balanced buckets)	80	115	175	235	205	445	590
	100	100	155	205	255	380	510
	120	90	135	180	225	335	450
	160	75	110	145	170	275	370
	200	60	90	125	155	235	310

Centrifugal-discharge Type. The centrifugal-discharge type (Fig. 20-5*a*) of bucket elevator usually consists of malleable-iron buckets spaced at intervals and attached to the chain or belt by special attachments. The material is delivered into a boot from which it is scooped up by the buckets, elevated, and discharged by being thrown from the buckets by centrifugal force while passing over the head sprocket. Proper discharge action depends upon the diameter of head wheel, the speed and spacing of the buckets, and the location of the discharge chute. This type of elevator is satisfactory for the elevation of material containing no large lumps where nominal capacity is required. For fine, dry, or abrasive materials, a belt is usually used in place of chains. Where continuous service is required, it is better to use a slower speed elevator of the continuous-bucket type. Fine materials of light density should never be handled in a centrifugal elevator.

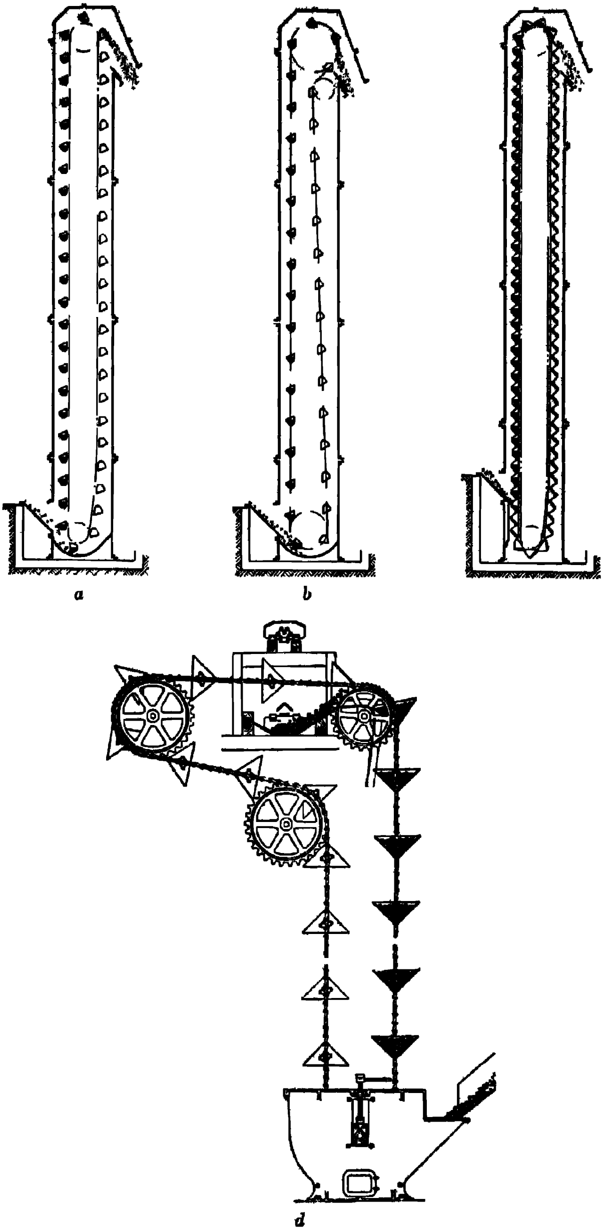

Fig. 20-5. Bucket-elevator types.

Perfect-discharge Type. The perfect-discharge bucket elevator (Fig. 20-5*b*) has buckets suspended at intervals between two strands of chain and operates at slower speeds than the centrifugal-discharge type. On the return side, just below the head wheels, are located a pair of idler sprockets to deflect the chains and completely invert the buckets. These idler sprockets may be omitted when the elevator is operated on an incline. Because of the slower speeds, fragile materials may be elevated without the breakage that would occur with the centrifugal-discharge type, and the operating parts wear longer.

Continuous Type. The continuous bucket elevator (Fig. 20-5*c*) consists of a continuous series of steel buckets on one or two strands of chain or a belt. The buckets are so shaped that the front of each will act as a deflector for the material being discharged from the succeeding bucket as it passes around the head wheel. Thus clean discharge is effected at the slow operating speeds (from 80 to 125 fpm). An inclination of the elevator will permit the discharge chute to be placed under the deflecting bucket, reducing spill.

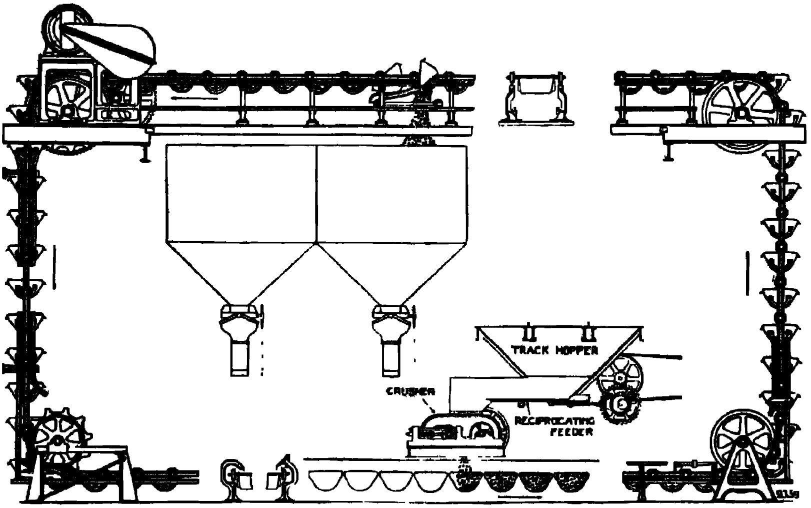

FIG. 20-6. Pivoted-bucket carrier.

A feeding leg is usually used, particularly with lumpy or abrasive material, in which the material is fed to the elevator two or three buckets above the foot wheel, eliminating the wear and strain of digging through an accumulation of material at the foot of the elevator boot. Elevators constructed in this manner are used extensively for the lifting of crushed stone, ore, coal, or other material of a bulky or gritty character and for certain light and fluffy materials.

The amount of material transported per hour by a continuous bucket elevator is relatively high, but the initial cost is higher than with the centrifugal- or perfect-discharge types, for more buckets are used, and since the loads are heavier, larger size machinery is required. However the lower maintenance of the continuous elevator will generally justify the additional cost.

Gravity-discharge Type. The gravity-discharge bucket elevator (Fig. 20-5*d*) was designed and is extensively used for elevating and conveying problems involving vertical and horizontal paths of operation or at inclinations up to 45 deg. The steel buckets, approximately V-shaped in cross section, are fastened between two strands of chain at right angles to the pitch of the chain. The material is scooped up from the boot as the buckets pass around the foot wheel and is elevated to the desired height where the buckets turn around the head wheel and trough-turn to a horizontal position of travel. Here the buckets push the material along a steel trough from which it may be discharged through gates or openings in the bottom of the trough.

By regulating the gates on the discharge run, the material may be delivered into any one of a number of bins. A number of bins may automatically be filled in series, for when the first bin is full, the material will obstruct the opening, and the following material will be carried on to the succeeding bin.

The slow speed (approximately 100 fpm) of the gravity-discharge type lessens degradation, for the pickup in the boot is slow, and the discharge may be down an inclined chute. This type is satisfactory for handling anthracite lump or run-of-the-mine coal, but ashes or gritty material will cause excessive wear to the horizontal troughs.

Pivoted-bucket Carrier. The pivoted-bucket carrier (Fig. 20-6) consists of a series of buckets pivotally suspended between two strands of roller chain and maintained in the carrying position by gravity. This permits a single carrier to transport vertically, horizontally, or on an inclined path. The buckets are usually constructed with a lip which overlaps the succeeding bucket while on the horizontal path and are fed while in this position. Discharge must be made on a horizontal run and is performed by a portable tripper overturning the bucket. This allows the discharge point to be varied along the length of the horizontal run. The buckets are either righted by gravity or by a separate device.

With this type of carrier, the power required is kept to a minimum as the friction losses are kept down by supporting of the bucket line on self-oiling rollers of large diameter. Elevating and conveying problems may be carried out through the use of one machine, and only one driving unit is required. Long life is attained through the slow operating speed (from 40 to 80 fpm) with smooth, quiet operation. Abrasive materials may be transported without undue wear. The system is economical to install and operate where the capacity and travel are not too great.

BULK CONVEYORS (See Table 20-2)

Flight Conveyors. The flight or pusher type of conveyor (Fig. 20-7) was developed as a means for distributing coal or other nonabrasive materials horizontally or at angles up to 45 deg. The material is received in a trough, pushed along by the flights or scrapers attached at intervals to a chain or chains, and discharged through gates in the bottom or at the end of the trough. Conveying is done on the bottom run.

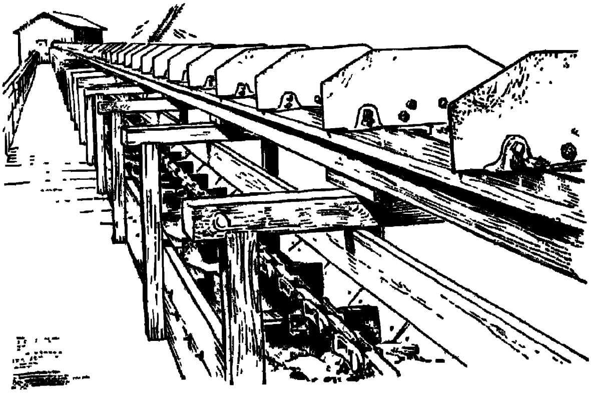

Fig. 20-7. Flight or pusher-type conveyor.

Flight conveyors are commonly used in boilerhouses and coal yards for the handling of coal which may be in the form of lumps or fines. The transporting of abrasive materials will shorten the life of the conveyor to a considerable extent. The maximum usable inclination is approximately 45 deg. With horizontal conveyors capacities may be up to 360 tons an hr of coal at a speed of 150 fpm.

Drag-chain Conveyors. Drag-chain conveyors are similar to flight conveyors in operation except that the flights are omitted, the conveyor consisting of nothing more than wide chain, of a suitable type, sliding in a trough and carrying the material with it. It operates at slow speeds (up to 20 fpm). A chain operating in a trough 20 in. wide and 8 in. deep, with a depth of material of 6 in., will carry 4.75 tons per hr of material weighing 50 lb per cu ft at a speed of 5 fpm. At other speeds and weights the capacities will be in direct proportion. Because of the slow speed, it is possible to handle abrasive materials with reasonable life if the chain and trough are constructed of hardened materials.

Conveying en Masse. The mechanism of the en masse conveyors (Fig. 20-8) appears much like a flight conveyor enclosed in a casing. It consists of a series of skeleton or solid flights (or a combination of the two) connected together with articulated joints to form a continuous chain. It operates over suitable pulleys, or pulley, and is enclosed within a dust-tight casing. A wide variety of operating paths varying from horizontal to vertical may be obtained. The feed may be at one or more points on either the upper or lower strands of a substantially horizontal run, while the discharge points may be on horizontal or at top of vertical runs.

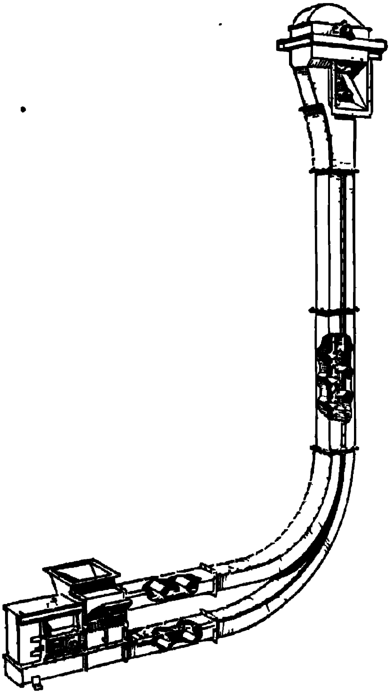

Fig. 20-8. En masse conveyor.

The space requirements for installation of the casing are small. The cross-sectional area required is less than one-fourth that of a flight conveyor, approximately one-fifth that of a bucket elevator, and one-third of a belt conveyor. Vertical casings are not self-supporting. While higher capacities are obtainable, it is recommended that, for requirements of over 100 tons per hr (50 lb per cu ft of material), another type of conveyor or elevator be used.

This conveyor is most suitable for handling pulverized or granular materials. Small lumps can be handled if they are not too large to enter or discharge. Corrosive, abrasive, or sticky materials should be avoided for the average installation.

Apron Conveyors. The apron conveyor is made up of overlapping steel pans which are carried between two strands of roller chain traveling on suitable tracks on both the top and bottom runs (see Fig. 20-9). Movable or stationary retaining sides may be used to permit a greater depth of material to be carried. Apron conveyors are used extensively for handling materials such as coal, ash, clinkers, sand, crushed stone, etc. A horizontal apron, 48 in. wide, with an average depth of 6 in. of coal and a speed of 50 fpm, will handle 100 to 110 tons per hr. Apron conveyors may be installed on inclines up to 25 deg.

Apron conveyors make excellent feeders from track hoppers. The feeder usually does not have sides on the pans but uses stationary skirt plates to retain the material on the conveyor. When used as feeders, they usually operate at a slower speed than when used as conveyors. A 36-in.-wide apron with 18-in.-high skirt plates will handle 45 tons per hr of coal traveling at 10 fpm.

The standard apron conveyor will withstand greater shock in the loading process than will other types of conveyors; however, in applications such as are encountered in the mining field, a more rugged conveyor is required. An apron conveyor of modified design will permit the carrying of lumps up to 6 ft in diameter and weighing a ton, and these lumps may be dropped onto the conveyor from a considerable height. The chains, rollers, sprockets, and overlapping pans are usually made of manganese steel. The chains are placed under the pans, away from the material, and the broad wearing surfaces of the chain links form substantial tracks for the closely spaced large-diameter rollers supporting the carrying run. On short feeders the return run is unsupported between the head and foot shafts. On medium- or long-center feeders, suitably spaced rollers carry the return run.

Reciprocating Plate Feeders. Reciprocating plate feeders are rather inexpensive units and are very satisfactory as a medium for feeding incline belt conveyors starting under track hoppers.

Screw Conveyors. The helical screw conveyor (Fig. 20-10) is particularly adaptable to the conveying of small-sized material such as grain, cement, cottonseed, pulverized coal, sand, etc. Where cleanliness or corrosion resistance is a factor, as in the food or chemical industries, the component parts may be made of special metals (brass, stainless steel, aluminum, etc.). In cases where abrasive materials are to be handled, hardened parts are used, and the screw may have a thickened wearing edge, replaceable if desired, or it can be stellited.

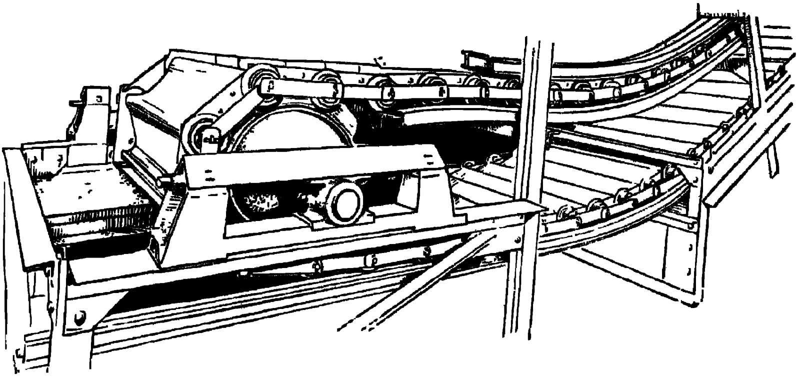

Fig. 20-9. Apron conveyor.

The diameter of the screw will vary from 3 to 24 in. and may be right hand, left hand, or a combination of the two (to move material either toward or away from the point of juncture). Where sticky materials are to be carried, a ribbon screw is used which has a clear space between the central shaft and the helix, eliminating the build-up of material along the shaft. If it is desired to have the conveyor act also as a mixer, separate paddles may be added to the screw shaft, or in the case of the ribbon conveyor, a second ribbon can be added inside the first but with opposing helix

Up to approximately 15 deg from the horizontal there is little decrease in the quantity of material conveyed, but above that point there is an appreciable decrease, the amount of which will depend upon the material, the air trapped in the material, and the pressure at the loading point. Installations up to the vertical are made with the pitch of the screw being reduced in such cases. Discharge may be over the end of the conveyor trough or through openings or gates in the bottom of the trough. The screw conveyor is especially applicable as a feeder. For free-flowing materials it will act as a shutoff when stopped. It can be installed within relatively small space limitations.

Vibrating Conveyors. Vibrating conveyors consist of a single-piece steel (or other material) deck to serve as the carrier with the vibratory motion imparted by a mechanical, electric, or hydraulic drive. On one device of the mechanical type, the deck is supported on pairs of legs hinged at each end. The lower ends of the legs are ahead (considering the direction of travel of the material) of the upper ends (Fig. 20-11).

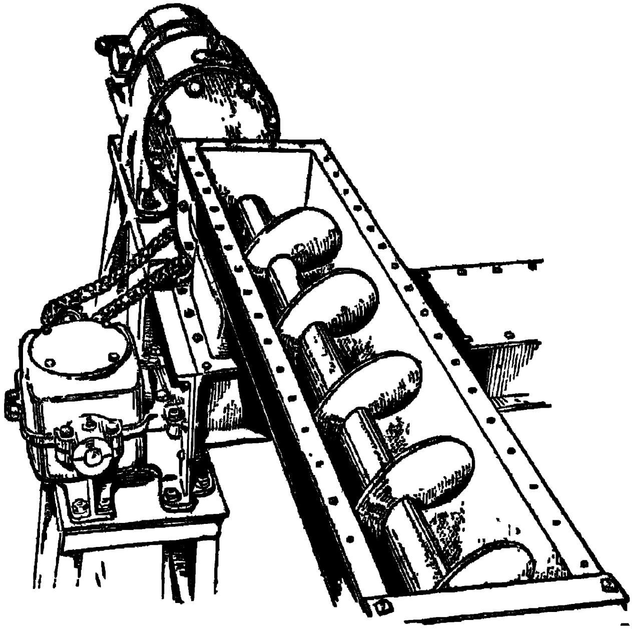

FIG. 20-10. Helical screw conveyor.

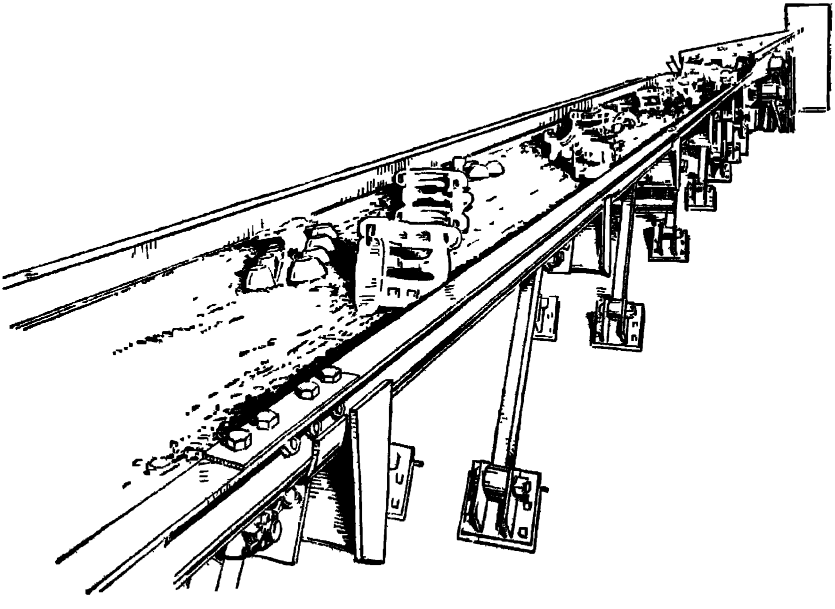

FIG. 20-11. Vibrating conveyor

Thus, as oscillatory motion is imparted to the deck by a short throw crank, an upward and forward thrust is obtained, and the material will "float" along, being held in suspension most of the time by the motion of the trough. The rate of movement will be from 20 to 50 fpm.

Another type of vibrating conveyor has the deck suspended or supported on coil springs or cushions. An electric driver or drivers, of the magnet or solenoid type, is attached to the underside of the deck at an angle of about 20 deg from the horizontal. As a series of a-c or interrupted d-c pulses is applied to this "motor," a vibration of the deck is obtained both vertically and horizontally, causing the material to "flow" along the deck. These conveyors are manufactured in various sizes handling up to 200 tons per hr, level operation. The speed of material movement may be varied by changing the amplitude of the vibrations through control of the amount of electric current supplied.

A vibrating conveyor is ideally suited for the handling of materials that would produce unsatisfactory results with most other types of conveyors, such as metal turnings, borings and chips, steel scrap, castings, shakeout sand, and material that is abrasive, wet, oily, hot, and jagged or irregular in shape. For dust control, the conveying decks may be enclosed pans, rectangular or circular. Flake material may be handled up a slight incline. The conveyor may be combined with screens or grizzlies for selective separation of the material.

Belt Conveyors. The principal elements of belt conveyors are a belt, usually rubber covered (see Power Belting, Fasteners, Pulleys, and Shafting in Sec. 14), two end pulleys, the driving machinery, and usually a series of intermediate supporting pulleys. The intermediate pulleys, or idlers, may consist of one, three, or five separate rollers. In the case of multiple rollers, the outer ones are inclined to trough the belt, increasing its carrying capacity without spill. Roller or ball bearings on the idlers are used to decrease the friction losses and to lessen the maintenance required. The carrying idlers should be from 3 to 5 ft apart, depending upon the capacity required, the size of lumps, and the weight of material per cubic foot. The spacing of the return idlers may be 9 to 10 ft apart. Pneumatic idlers are used to lessen the effects of shock at loading points where it cannot be avoided.

To reduce the wear on the belts, the loading should be in such a manner that the material striking the belt will have an initial velocity approximately equal to that of the belt and in the same direction. Where a mixture of lumps and fines is handled, it is best to have the fines loaded on the belt first to provide a cushion for the loading of the lumps. Stationary skirtboards, a slight distance above the belts, are often used at the loading point to eliminate spill.

Discharge may take place over the head pulley, or stationary trippers may be installed at predetermined points along the path of travel. Where it is desired to unload at a number of points, a movable tripper may be installed which will allow discharge at any point along the travel of the belt but not closer than approximately 20 ft from the loading end. Headroom may be a factor with tripper installations since they require from 4 to 7 ft above the conveying belt. For slow-speed belts, installed flat, a strike-off board may be used to act as a plow in removing the material from the belt.

Belts operate at speeds from 300 to 600 fpm (higher when carrying grain) and may be from 12 to 60 in. wide. Because of their high speed and continuous delivery of material, the quantity carried per hour will be relatively high. A 60-in. belt carrying coal (50 lb per cu ft) at 600 fpm will carry 2,160 tons per hr. They can be operated at inclines up to approximately 18 deg, depending upon the physical characteristics of the material.

Owing to the large continuous capacity, with low power consumption, the cost per ton for belt conveying is relatively low. Belt conveyors are simple in construction and relatively noiseless in operation. A number of interesting bulk-material handling problems have been solved through the use of belts, such as the Shasta Dam construction on the Sacramento River, California. Concrete aggregates were carried 9.6 miles on twenty-six 36-in.-wide belt conveyors, operating at 550 fpm with a capacity of 1,100 tons an hr.

Portable Loaders and Unloaders. A number of portable loaders and unloaders are made, using belts, and are particularly adaptable for intermittent operation where the total volume per day is small. A boxcar loader is produced which is wheeled to the car door where it can be supplied by another conveyor or from a chute. The belt is run at high speed which serves to throw the material as far as either end of the car, distributing it for proper loading.

A larger unit, mounted on a permanent base, beside or between loading tracks, has hinged and pivoted arms supporting the belt mechanism to allow the loader to be swung into the car. A slow-speed belt is used, the arms having sufficient movement to reach either end of the car and deposit the material without throwing. Degradation is kept at a minimum by the gentle deposit of the load.

A belt made of a band of 0.65-carbon cold-rolled steel, 0.03 or 0.04 in. thick, is available in widths up to 32 in. These belts operate flat, usually handling materials that are injurious to rubber and fabric belts. Woven-wire belts are used on conveyors that wash, dry, heat, or cool material. Their open structure permits liquid to drain from the material and permits air to circulate freely. This wire can be of a metal that will resist corrosion or great heat.

Bulldozers, Carryalls, and Drag-line Scraper Buckets. Bulldozers, carryalls, and drag-line scraper buckets are generally considered for building up outdoor stock piles and for reclaiming same.

PACKAGE HANDLING

In package handling the material is in unit form: cartons, boxes, barrels, sacks, or pieces in their semifinished or finished state. The characteristics of the units and the purpose of moving them determine the device to be used for any particular problem (see Figs. 20-12 and 20-13).

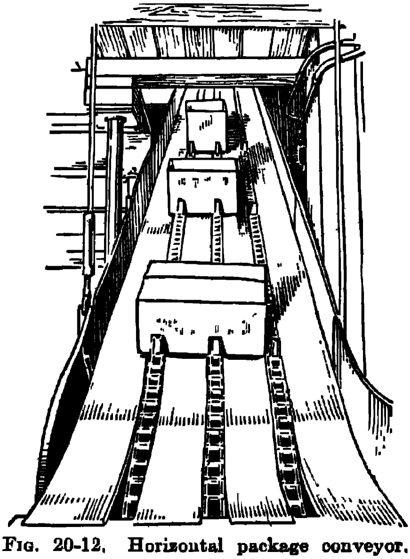

FIG. 20-12. Horizontal package conveyor.

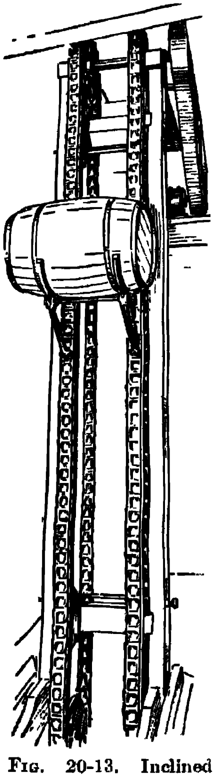

FIG. 20-13. Inclined package conveyor.

Elevators. A belt or a pair of chains, operating vertically or on an incline, may have rigidly attached protruding arms suitable for the package. In loading for upward travel, the package is placed on a finger skid. The ascending arm passes through the skid and picks up the package. The discharge is usually over the head wheel where a fingered chute will catch the package while permitting the arm to pass through. If it is desired to discharge on the ascending run, the arms can be arranged to discharge on contact with suitable mechanism at the desired unloading level.

An elevator may have carrying trays pivotally suspended between two strands of chain, with the center of gravity of the trays and load low enough to maintain them in an upright position. The carrying trays consist of transverse fingers which mesh with similar fingers in the loading and discharge skids. The load is placed on the skid from which it is picked up by the ascending tray, carried over the head wheel, and discharged on the descending side at the first point a skid is encountered.

A reciprocating lift can be used for elevating drums on end, cartons, tote pans, etc. This device can be made to function on a semi- or fully automatic basis. It also can be arranged to lower packages by merely providing the necessary electric controls.

Portable Elevators. A number of different types of portable elevators, stackers, or tiering machines are made with the lifting mechanism either motor or hand operated. Various methods of guiding the vertical path of the lifting tray are used. One type has a turntable base to allow use in restricted quarters, such as narrow aisles. The maximum size of a standard unit is approximately 2½ tons with an 18-ft lift.

Portable stackers are made for the handling of smaller units. Some of these inclined stackers use a belt with or without transverse cleats. Others have chains with flights to push the loads up an inclined chute.

Gravity Roller Conveyors. Gravity roller conveyors, consisting of ball-bearing mounted tubes or pipes, are generally used to convey packages short distances and around curves. The permissible distance is determined by practical loading height and by the normal pitch required of approximately ½ in. per lin ft. Horizontal runs of gravity can be used to move heavy masses by hand as the rolling resistance will be from 1 to 3 per cent. Roller curves also are combined to form a lowering spiral which serves as satisfactory supply to filling machines or to a loading dock. For long horizontal distance, where a supply of containers is to be stocked for ready use, the roller conveyor can be made "live" by an actuating drive belt placed under the carrying roller (known in the industry as "live roller conveyor").

Spiral Chutes. Spiral chutes, made in the form of a vertical helix with a guardrail, are universally used as a cheap and practical means for lowering most types of packages except steel drums.

Conveyors. Belt conveyors, operating horizontally or inclined, may be used for the handling of large or small packages. For small, light packages, the carrying run of the belt may be slid along a flat surface to provide the necessary support. For heavier loads, rollers are used to carry the load. The rollers will usually be smaller and closer together than those used with bulk-material conveyors. To eliminate bumping, the package on the belt should be supported by no less than two rollers at any time. Packages may be handled on inclines up to approximately 25 deg through the use of crepe-top belts. The belt speeds are generally less than when handling bulk material.

Slat conveyors have transverse wooden or metal slats connected to and moved by one or two endless chains. The weight may be carried by the slats sliding on a pair of tracks, but the better practice is to have the slats fastened to roller chains; the weight is rolled, and friction is less.

A push-bar conveyor uses a continuously moving chain to which are attached flights or pushers to provide the impetus for the movement of packages. The amount of inclination that can be attained with this type is limited only by the overturning of the packages.

A number of wheeled carts can be attached to an endless chain. This system is commonly used in foundries. The flasks are loaded on the carts at the molding machine, poured, and transported to the shakeout in a continuous movement. From the shakeout the empty flasks may be returned to the molder on the same cart.

Chain conveyors are used with the article carried directly on single or multiple runs of chain sliding in guide troughs. Stationary skirtboards may be employed to prevent side movement of the article on the chain. Where inclines or declines are required, raised lugs on the chain prevent slipping of the article. Normal chain speeds will be approximately 30 fpm.

Overhead conveyors consist of small wheels operating on one or two rails with suitable carriers suspended therefrom. The wheels are connected by an endless conveyor chain driven by a sprocket or a second short drive chain with protruding dogs (known as a "caterpillar drive") to engage and transmit motion to the conveyor chain. Overhead conveyors may follow a wide variety of paths: up or down steep inclines, around short-radius horizontal curves. The conveyor is usually suspended from above and does not require floor space.

Overhead supported telphers or tramrails are also used. By the use of a number of tracks and switches these machines are not limited to one fixed pattern of operation but have some flexibility in moving material from one department to another.

Hoists and Grabs. Hoists and grabs are generally used in conjunction with jibs, monorails, or cranes to raise and transport carrier racks, charging hoppers, and miscellaneous objects. The hoists can be chain actuated for light loads or can be powered by air, hydraulic, or electric means for handling heavier or more frequent loadings. Many forms of standard and special grabs have been developed for engaging most of the objects or shapes encountered in industry.

SKID AND PALLET HANDLING

A pallet is nothing more than a portable platform on which objects are placed to facilitate their handling. The universal use of pallets was given impetus by Army and Navy shipping requirements during the Second World War.

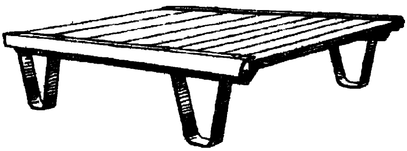

Fig. 20-14a. Skid.

The pallets used today by industry for the easy handling of one or multifold units in a single load are constructed with a base raised a short distance from the floor to allow the tines of a fork truck to be inserted underneath to raise the load. Similar to the industrial-type trailers are the "live" skids which are mounted on wheels, usually of the rigid-caster type. This allows short movements to be made without engaging the tine of a fork truck, but they are carried or pulled by a fork truck during long movements

More common are the skid (Fig. 20-14a) which is supported on legs and the pallet (Fig. 20-14b) which rests on a flat base. In addition to being less expensive, pallets have the advantage that they, with loads, can be stacked in tiers to accomplish space saving in storing or warehousing operations. The standard size of a pallet is 4 by 4 ft. They may be manufactured of wood, metal, or a combination of the two.

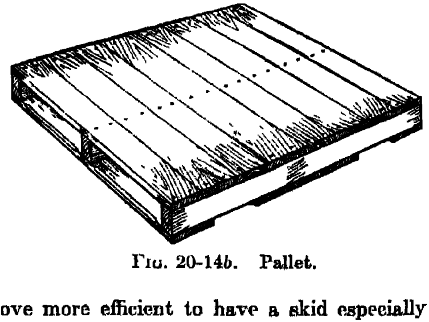

Fig. 20-14b. Pallet.

In the handling of large individual units or where plant requirements are such that special "skid boxes" are impractical, the flat skid may be the most suitable. Figure 20-15 illustrates such a metal skid being handled with a fork truck. For other applications it may be more suitable to have a container box, racks, or multiple trays mounted on the skid. In many cases it will prove more efficient to have a skid especially designed for the conveying problems encountered in the particular plant (Fig. 20-16)

The use of pallets not only affords the advantage of ease and flexibility of handling materials throughout the plant but allows rapid loading and unloading of trucks and boxcars. For plants having only intermittent loading and unloading problems, a fork truck need be freed from its other duties for only a short interval to complete the job so that plant investment is not tied up in a mechanical device that is idle a considerable portion of the time.

Pallets or skids used for shipping may be any of the general type. With the flat-skid type it may be advisable in many cases to secure the load to the skid with metal straps, while the use of adhesives between the individual packages on the load may reduce the number of straps required. If container-type pallets used for shipping are to be returned for reuse, they should be constructed so they can be "knocked down" to effect a space saving that will allow a large number to be returned in one load.

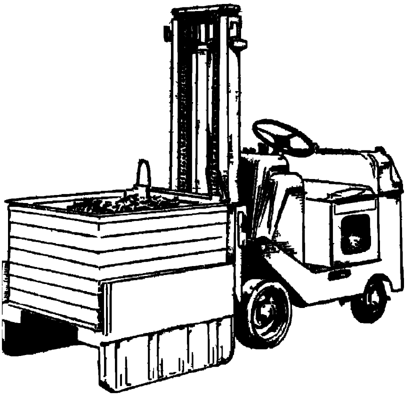

Fig. 20-15 Metal skid being handled by fork truck

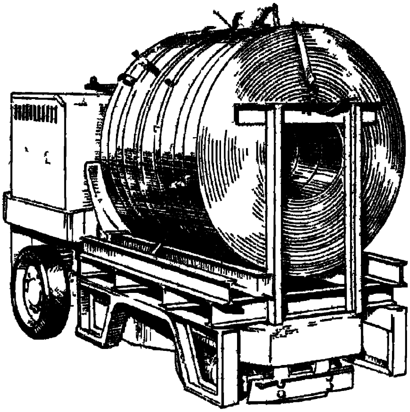

Fig. 20-16 Skid of special design to suit the object to be handled

The prevailing method of pallet or skid handling is by the use of fork trucks. These are produced for hand operation, or they may be driven by gasoline engine or electric motor, battery supplied. Powered models are available up to capacities of 9 tons and lifts up to approximately 25 ft. In many cases it will be economical to use one or more of the package conveyors, previously discussed, alone or in conjunction with fork trucks, for specific handling operations. Loads consisting of quantities of long items (pipe, lumber, steel, etc.) may be handled on larger bolsters through the use of straddle trucks, as shown in Fig. 20-17.

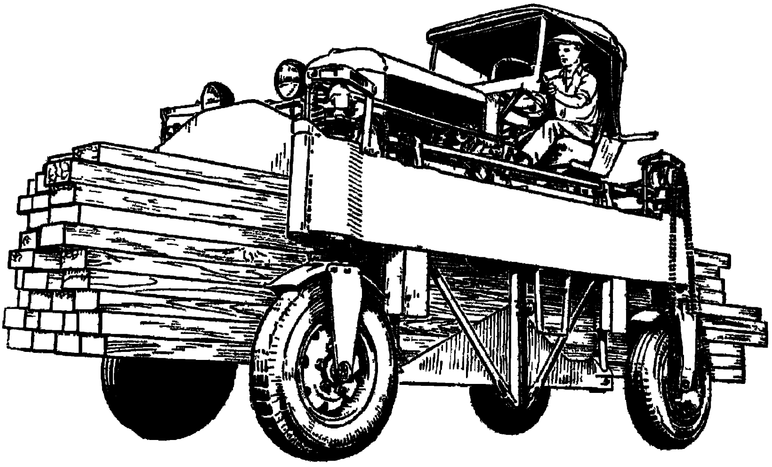

Fig. 20-17 Straddle truck for the handling of pipe, lumber, steel, etc.

CONVEYOR AND CONVEYOR-BELT ENGINEERING

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INTRODUCTION

Before starting the design of a belt conveyor, the following information must be available:

1. Kind of material to be handled and its weight per cubic foot, maximum size, condition of the material, such as wet, dry, hot, oily, acid, etc.
2. Quantity to be handled in terms of maximum short tons per hour
3. Distance to be conveyed and the height to elevate or lower

With this information, the necessary belt width required for these conditions can then be determined. The width is considered in conjunction with belt speed in feet per minute (Table 20-10), the maximum material size (Table 20-14), and the angle of incline (Table 20-16). Also see Tables 20-12 and 20-15.

Other factors such as type of feed, balancing conveyor drives for uniformity of motors, speed reducers, standardization of belt widths and plies, breakage of material, and spillage must also be considered.

The next step is to decide on the type and location of the belt-driving mechanism and the take-up.

The foregoing information is necessary to calculate

1. Total power the belting must transmit. See Solving for the Total Horsepower the Belting Must Transmit, page 1420, which gives the necessary steps to follow.
2 Convert this total power, horsepower, into terms of maximum belting working stress in pounds. This starts with Converting Total Horsepower Belting Must Transmit into Maximum Belt Stress in Pounds (T_M). Nine different conveyor examples, Figures 20-18 to 20-26, inclusive, follow this heading.
3. Decide on the correct belt-carcass construction (see page 1426).
4. Determine cover thicknesses to be used (see pages 1428 and 1429).
5. Equipment over which the belting must operate and suggestions for same (see pages 1429 and 1430).

SOLVING FOR THE TOTAL HORSEPOWER THE BELTING MUST TRANSMIT

Solving for the total horsepower the belting must transmit is divided into three main factors.

1. Horsepower to overcome friction in all the moving parts that are belt driven. This is factor X.

$$X = (G \times F_x \times S \times L_c)/33{,}000$$

2. Horsepower required to move the load horizontally (without lift or drop). This is factor Y.

$$Y = (L_c \times F_y \times C)/990$$

3. Horsepower required to elevate the load or the power generated by the load moving downhill. This is factor Z.

$$Z = (C \times H)/990$$

(This power is plus for upgrade and minus for down.)

NOTE: If there are belt-driven conveyor accessories such as trippers, feeders, cleaners, etc., more power will have to be added for them (see page 1436 for extra power to add for trippers).

The total power transmitted by the belt then equals horsepower and is the algebraic sum of the above three factors.

$$\text{Hp} = X + Y + Z \qquad \text{or} \qquad \text{Hp} = X + Y - Z$$

as the case may be.

In the above three formulas

G = weight, lb per ft, of center distance over the belting of all moving parts that are belt driven, plus the weight of 2 ft of belting (see Tables 20-17, 20-18, 20-19, and 20-20)

F_x, F_y = friction factors (see Table 20-21)

C = *maximum* rate of feed to the belting, short tons (2,000 lb) per hr (see Table 20-12)

S = belt speed, fpm

L = center-to-center distance measured over the belting between the head and tail pulleys

L_c = corrected center distance ($L_c = 0.55L + 115$)

H = total vertical lift or drop (including tripper height) measured from point of loading onto the belt to the point of discharge from the belt, ft

NOTE: The total horsepower the belting must transmit does not include the power losses in the driving motor or motors or in the reducing unit (see page 1436 for instructions in determining the correct motor size needed to compensate for these losses).

CONVERTING TOTAL HORSEPOWER BELTING MUST TRANSMIT INTO MAXIMUM BELT STRESS IN POUNDS (T_M)

Converting total horsepower belting must transmit into maximum belt stress in pounds (T_M) is done by using two fundamental formulas for power transmission by belting:

$$\text{Hp} = [(T_1 - T_2)S]/33{,}000 \qquad \text{or} \qquad T_1 - T_2 = (\text{hp} \times 33{,}000)/S = T_E \qquad (1)$$

$$(T_1 - T_c)/(T_2 - T_c) = 10^{0.00758fa} \qquad \text{or for conveyors} \qquad T_1/T_2 = 10^{0.00758fa} \qquad (2)$$

where $T_1 - T_2 = T_E$ = the effective tension, or the difference between the theoretical tight-side tension T_1 and the theoretical slack-side tension T_2 at the conveyor drive. In the case of tandem- or dual-motor drives, an intermediate belt tension T_3 exists between the two driving pulleys

S = belt speed, fpm

Hp = total power the belting must transmit

$T_c = Wv^2/g$ = belt tension due to centrifugal force which for conveyor speeds is so small that it can be neglected (W = belt weight per lineal foot, lb; v = belt velocity, fps; and $g = 32.2$)

f = coefficient of friction between belting and the drive pulley or pulleys. 0.25 is used between rubber-belting covers and bare-steel or cast-iron pulleys and 0.35 between rubber-belting covers and rubber lagging. These figures are used for either wet or dry conditions. If rubber lagging is frequently wet, a grooved-design lagging should be used to prevent excess slippage between the belting and the lagging

a = total arc of contact between the belt and the driving pulley or pulleys, deg

For any given set of driving conditions, by solving simultaneously Eqs. (1) and (2) above, both T_1 and T_2 can be determined. That is assuming that horsepower, speed, and the other factors are known or assumed except T_1 and T_2.

Instead of doing this, it is much simpler to use Table 20-22. This table gives values to a drive factor K for any set of driving–take-up combinations, with bare or lagged pulleys.

Multiplying the effective tension T_E by this factor will give the theoretical slack-side tension T_2 needed in the belting at the drive to enable the belting to transmit the required horsepower without slippage occurring. Thus

$$T_1 - T_2 = T_E = (\text{hp} \times 33{,}000)/S \qquad T_2 = KT_E,$$

and substituting this value of T_2 in Eq. (1),

$$T_1 = T_E + T_2 \qquad \text{or} \qquad T_1 = (K + 1)T_E$$

For many drives T_1 equals the maximum belt tension T_M. However, there are cases where T_1 is not the maximum tension and where T_2 is considerably more than the theoretical value needed to prevent slipping between the belting and the drive pulley or pulleys. These variations are due to drive location and depend upon whether the belt conveyor is an elevating or a lowering one. The effect of belt weight on a slope becomes a very important factor particularly on long-center, high-lift, or decline conveyors.

This belt-slope weight equals HW, where H is total lift or drop in feet, and W is belt weight per lineal foot. But the return idlers have a retarding action which tends to lessen this weight or pull. The idler retarding force equals $0.4(G \times F_x \times L_c)$ (see Table 20-18 for figuring belt weights).

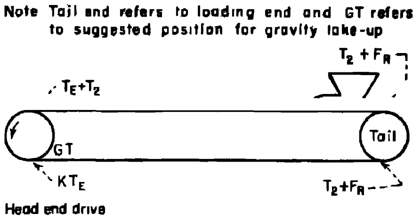

Note Tail end refers to loading end and GT refers to suggested position for gravity take-up
$T_2 + F_R$
$T_E + T_2$
GT
Tail
KT_E
$T_2 + F_R$
Head end drive

FIG. 20-18. Approximately horizontal conveyor with drive at or near head end.

$T_E = (\text{hp} \times 33{,}000)/S$
$T_2 = KT_E$ = theoretical slack-side tension
$T_1 = T_E + T_2$ or $(K + 1)T_E$
$T_M = T_1$

To simplify the calculations for various types of conveyors, typical examples are given by Figs. 20-18 to 20-26, inclusive. These show drive and take-up location and the steps necessary to arrive at the maximum belt working stress T_M.

NOTE: This notation applies to Figs. 20-18 to 20-26, inclusive. Since it is usually necessary to increase the take-up weight over the theoretical amount, it is suggested that 2 per cent of the T_1 value be added to the theoretical maximum tension T_M to estimate the maximum operating tension.

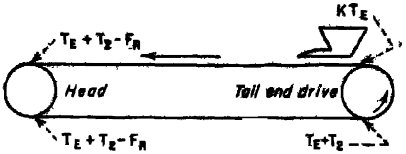


Fig. 20-19. Approximately horizontal conveyor with drive at or near tail.

T_E = (hp × 33,000)/S

$T_2 = KT_E$ = theoretical slack-side tension

$T_1 = T_E + T_2$ or $(K + 1)T_E$

$T_M = T_1$

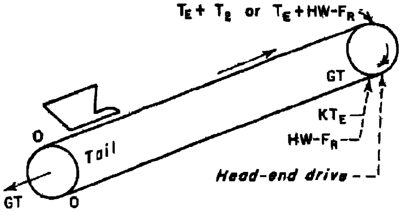


Fig. 20-20. Elevating conveyor with drive at or near head end.

T_E = (hp × 33,000)/S

$T_2 = KT_E$ = theoretical slack-side tension

$T_1 = T_E + T_2$ or $(K + 1)T_E$

The effect of return-side belt tension = $HW - F_R$. If T_2 is greater than $HW - F_R$,

$$T_M = T_1 = T_E + T_2$$

If T_2 is less than $HW - F_R$,

$$T_M = T_E + HW - F_R \quad \text{or} \quad T_1 - T_2 + HW - F_R$$

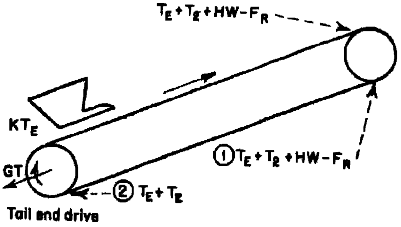


Fig. 20-21. Elevating conveyor with drive at or near tail end.

T_E = (hp × 33,000)/S

$T_2 = KT_E$ = theoretical slack-side tension

$T_1 = T_E + T_2$ or $(K + 1)T_E$

The effect of return-side belt slope tension = $HW - F_R$. If HW is greater than F_R, then maximum tension is at ①, and

$$T_M = T_E + T_2 + HW - F_R \quad \text{or} \quad T_1 + HW - F_R$$

If HW is less than F_R, then maximum tension is at ②, and

$$T_M = T_1 = T_E + T_2$$

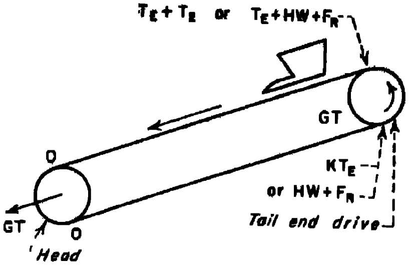


FIG. 20-22. Lowering conveyor with tail-end drive (holdback).

Suppose that in the horsepower calculation for a lowering conveyor, $X + Y - Z$ came out to be a negative number, which meant that Z was greater than $X + Y$. The fully loaded belt would tend to run down the slope, and holdback force is necessary at the drive.

$$T_E = (\text{hp} \times 33{,}000)/S$$
$$T_2 = KT_E = \text{theoretical slack-side tension}$$
$$T_1 = T_E + T_2 \qquad \text{or} \qquad (K + 1)T_E$$

The effect of return-side belt tension $= HW + F_R$. If T_2 is greater than $HW + F_R$,

$$T_M = T_1 = T_E + T_2$$

If T_2 is less than $HW + F_R$,

$$T_M = T_E + HW + F_R \qquad \text{or} \qquad T_1 - T_2 + HW + F_R$$

Caution: In most lowering conveyors, the T_E and T_2 tensions required for the empty conveyor are greater than for the loaded. Therefore the T_K used for the maximum-tension calculation should be the T_E calculated on basis of the horsepower X or the horsepower $X + Y - Z$, whichever is the larger.

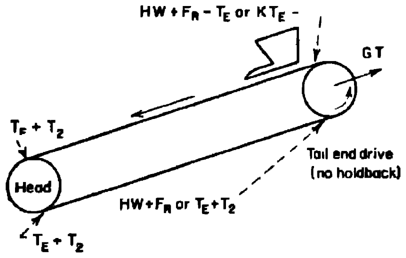


FIG. 20-23. Lowering conveyor with tail-end drive (no holdback).

Suppose that in the horsepower calculation for a lowering conveyor, $X + Y - Z$ came out to be a positive value, which meant that $X + Y$ was greater than Z. The loaded belt will then not tend to run down the slope but will always require some power, and no holdback will be necessary.

$$T_E = (\text{hp} \times 33{,}000)/S$$
$$T_2 = KT_E = \text{theoretical slack-side tension}$$
$$T_1 = T_E + T_2 \qquad \text{or} \qquad (K + 1)T_E$$

The effect of return-side belt slope tension $= HW + F_R$. If T_2 is greater than $HW + F_R - T_E$,

$$T_M = T_E + T_2 \qquad \text{or} \qquad T_1$$

If T_2 is less than $HW + F_R - T_E$,

$$T_M = T_E + HW + F_R - T_F = HW + F_R$$

Caution: In most lowering conveyors, the T_F and T_2 tensions required for the empty conveyor are greater than for the loaded. Therefore the T_E used for the maximum tension calculation should be the T_E calculated on the basis of the horsepower X or the horsepower $X + Y - Z$, whichever is the larger.

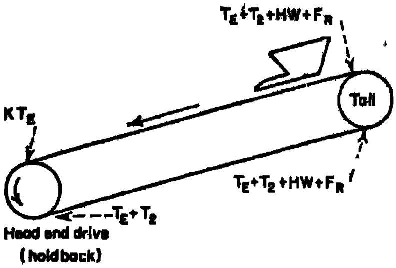


FIG. 20-24. Lowering conveyor with head-end drive (holdback).

Suppose that in the horsepower calculation for a lowering conveyor, $X + Y - Z$ came out to be a negative value, which meant that Z was greater than $X + Y$. The fully loaded belt would tend to run down the slope, and holdback is necessary at the drive.

$$T_E = (\text{hp} \times 33{,}000)/S$$
$$T_2 = KT_E = \text{theoretical slack-side tension}$$
$$T_1 = T_E + T_2 \quad \text{or} \quad (K + 1)T_E$$

The effect of return-side belt slope tension $= HW + F_R$.

$$T_M = T_E + T_2 + HW + F_R \quad \text{or} \quad T_1 + HW + F_R$$

Caution: In most lowering conveyors, the T_E and T_2 tensions required for the empty conveyor are greater than for the loaded. Therefore the T_E used for the maximum tension calculation should be the T_E calculated on the basis of the horsepower X or the horsepower $X + Y - Z$, whichever is the larger.

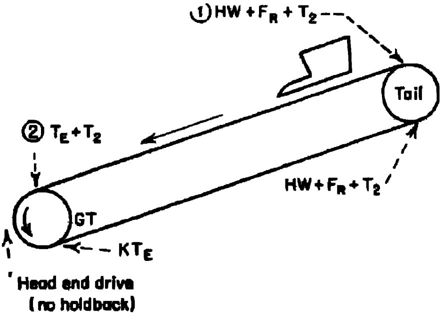


FIG. 20-25. Lowering conveyor with head-end drive (no holdback).

Suppose that in the horsepower calculation for a lowering conveyor, $X + Y - Z$ came out to be a positive value, which meant that $X + Y$ was greater than Z. The loaded belt will not tend to run down the slope but will always require some power, and no holdback will be necessary.

$$T_E = (\text{hp} \times 33{,}000)/S$$
$$T_2 = KT_E = \text{theoretical slack-side tension.}$$
$$T_1 = T_E + T_2 \quad \text{or} \quad (K + 1)T_E$$

The effect of return-side belt-slope tension $= HW + F_R$. If $HW + F_R$ is less than T_E,

$$T_M = T_1 = T_E + T_2 \text{ and is at ②}$$

If $HW + F_R$ is greater than T_E,

$$T_M = HW + F_R + T_2 \text{ and is at ①}$$

Caution: In most lowering conveyors, the T_E and T_2 tensions required for the empty conveyor are greater than for the loaded. Therefore the T_E used for the maximum tension calculation should be the T_E calculated on the basis of the horsepower X or horsepower $X + Y - Z$, whichever is the larger.

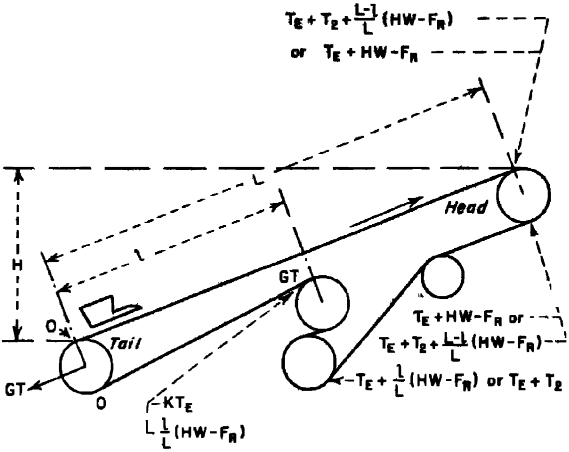

$T_E + T_2 + \frac{L-l}{L}(HW - F_R)$
or $T_E + HW - F_R$
L
l
H
GT
Head
Tail
O
O
GT
KT_E
$\frac{l}{L}(HW - F_R)$
$T_E + HW - F_R$ or
$T_E + T_2 + \frac{L-l}{L}(HW - F_R)$
$T_E + \frac{l}{L}(HW - F_R)$ or $T_E + T_2$

FIG. 20-26. Elevating conveyor with tandem or dual motor drive located part way down the slope on the return side.

$$T_E = (\text{hp} \times 33{,}000)/S$$
$$T_2 = KT_E = \text{theoretical slack-side tension}$$
$$T_1 = T_E + T_2 \quad \text{or} \quad (K + 1)T_E$$

The effect of return-side belt-slope tension $= (l/L)(HW - F_R)$. If T_2 is greater than $(l/L)(HW - F_R)$,

$$T_M = T_E + T_2 + (L - l)/L(HW - F_R)$$

If T_2 is less than $(l/L)(HW - F_R)$,

$$T_M = T_E + HW - F_R$$

DETERMINING TYPE AND NUMBER OF PLIES NECESSARY

The type and number of plies necessary must be

1. Sufficient to withstand the maximum belt operating tension T_M
2. Sufficient for the proper support of the load between idlers
3. Of sufficient body to withstand the impact at the loading point
4. To a minimum number for the belt width being considered so that there will be proper troughability when belt is empty or lightly loaded

The permissible working stresses for the various conveyor-belting fabrics and constructions, as given in Table 20-4, are based on actual strengths of the material used and the application of a suitable factor of safety. This factor of safety takes into consideration the higher stress in the belting when it is being started from rest either loaded or empty.

Determining Number of Plies Necessary for Maximum Tension

For fabric or cord belts through the No. 100 construction,

$$\text{Number of plies} = \frac{T_M}{\text{belt width (in.)} \times \text{permissible stress (lb/in./ply)}}$$

For steel-cord belts,

$$\text{Construction} = T_M/\text{belt width (in.)}$$

With the steel-cord constructions, the belting will have sufficient body to support the load and resist impact, and in addition these belts will trough satisfactorily in 24-in. or wider widths. In the case of the fabric and other belt constructions, it is necessary to consider the maximum and the minimum number of plies that can be used for any given belt width (see Table 20-23).

To get the best service from any conveyer belt and to take advantage of the strength that is built into it, it is desirable to make the belting endless by means of vulcanized splices rather than with metal fasteners. Table 20-4 gives separate columns of figures for metal or vulcanized splicing. In the case of the cord belt constructions, to take full advantage of the qualities built into the belting, it is preferable to use vulcanized splices. However, by proper balancing of base plies of fabric and the cord plies, it is possible to use the cord belting with metal fasteners. Note, however, if metal lacings are employed, the No. 50 type should be used, not types No. 70 or No. 100. See Table 20-5 on balancing No. 50 cord construction for vulcanized and metal splicing.

Table 20-4. Permissible Working Stresses

Belt duck weight, oz	Permissible stress, lb per in. per ply	
	Vulcanized splices	Metal fasteners
28	31	26
32	36	30
36	41	35
42	50	40
No 50 cord	50	40*
No 70 cord	70	40*
No. 100 cord	100	40*

* Refer to Table 20-5

Steel cord	*Permissible stress, lb per in of belt width*
No. 1000	1 000
No. 1500	1,500
No. 2000	2,000
No 2500	2,500
No. 3000	3,000

Note: It is suggested that, before using the No. 70, No. 100, or any of the steel-cord constructions, complete operating details be referred to the belting manufacturer and to the equipment manufacturer The initial cost of the investment in these latter constructions warrants the best in equipment design and operating maintenance, and such installations should not be considered as just ordinary conveyor installations.

Belt Flexibility

1. Sometimes a few heavy plies are better than a larger number of lighter plies.
2. Cord carcass is superior because of greater cushion and greater strength with no loss of troughability.
3. Twenty-eight-ounce fabric is indicated only for the lightest service.
4. High working stress, heavy load, large lumps, heavy impact indicate need for 42-oz fabric or cord construction.

Standardization

Wherever possible in a plant using a number of conveyors, it is desirable to standardize on one construction for each of the belt widths used. This will simplify the problem of keeping spare belting on hand. It is necessary in doing this that the belt decided upon be properly designed for the hardest service.

Table 20-5. Cord Conveyor-belt Ply Combinations

Total plies	Cord construction No. 50 with vulcanized splices		Cord construction No 50 when necessary to use metal fasteners	
	Fabric plies	Cord plies	Fabric plies	Cord plies
4	2	2	2	2
5	2	3	2	3
6	2	4	3	3
7	2	5	3	4
8	2	6	4	4
9	2	7	4	5
10	2	8	5	5

Note: With vulcanized splices, the permissible working stress in the above combinations is 50 lb and with metal splices, 40 lb.

DETERMINE COVER THICKNESSES

Before the Second World War the various rubber companies were making four qualities of belt-cover stocks. During the war these were reduced to two, and since that time they have standardized on three grades. (This is exclusive of special constructions for oil, heat, or other special services.)

Table 20-6 gives brand comparisons for the three standard constructions furnished by all of the major rubber companies.

In deciding on the cover thickness to be used consideration should be given the following:

1. Quality of rubber to be used
2. Thickness to be used
3. Type of breaker to be used if one is needed

For any given set of loading and material conditions a certain minimum of cover thickness must be used regardless of the length of the conveyor. It is easy to visualize that a $\frac{1}{32}$-in.-thick cover on a belt carcass would quickly be cut to pieces by sharp heavy rocks even if any point in the belt came under the loading chute only once every 15 min. A certain amount of cover wear occurs as the load moves along with the belt owing to slight shifting of such load as it goes over each carrying idler.

The lowest quality should be considered only where operating conditions are very easy, as when grain, powdered material, or very small pieces are being conveyed, or in package service. Medium and medium-heavy service conditions can usually be belted with the middle-quality belting. The top quality is indicated only in a few very severe operating conditions. A thick medium-quality cover will do a better job in service than a thin top quality.

Table 20-6. Conveyor-belt Brand Comparison

Company	No. 1 construction, 3,500 lb cover, 20 lb friction	No. 2 construction, 2,500 lb cover, 16 lb friction	No. 3 construction, 800 lb cover, 12 lb friction
B. F. Goodrich	Super Longlife	Longlife	Maxecon
U. S. Rubber	Matchless	Giant	Security
Goodyear	Stacker	Style M	Style W
Manhattan	Super Master	Master	Delhi
Hewitt	Maltese Cross	Ajax	Conservo
Boston Woven Hose	Bull Dog	Silver King	Iron Clad

NOTE: Many special constructions are available for extra-high-tension service. Goodrich and Goodyear make steel-cable carcass belts; U. S. Rubber makes an "Ustex" construction; Goodrich and Goodyear make cotton cord constructions; Manhattan makes what is called a "Homocord" belt; and several companies are experimenting with rayon, nylon, and other special fibers for the tension members in their high-tension belt constructions. Hewitt has made several "Raynile" belts.

Table 20-7 is a guide for thickness of top cover to use for various types of material.

Table 20-7. Thickness of Top Cover

Service	Top-cover gauge, in.	Type of material
Light	$\frac{3}{32}$-$\frac{1}{8}$	Nonabrasive materials such as wood chips, flue dust, loose cement, bone char
Ordinary or standard	$\frac{3}{16}$	Mildly abrasive materials such as sand, small pebbles, earth, or bituminous coal; rock, stone, or coal under 3 in.
Average severe	$\frac{1}{4}$	Abrasive materials such as anthracite coal, coke, and sinter; rock, stone, or coal up to 10 in.
Severe	$\frac{5}{16}$	Heavy abrasive materials such as ores of iron, copper, zinc, and lead; foundry refuse, limestone, and slag
Very severe	$\frac{3}{8}$	Heavy, sharp, abrasive materials such as trap rock, quartz, glass cullet, and iron pyrites, and exceptionally large lumps of any material

Except for the lowest quality belting and for very light service, it is good practice to include in the top-cover thickness either a regular breaker fabric ply or a transcord breaker ply. This not only greatly increases the adhesion of the cover to the belting, but it also assists, particularly in the case of the transcord breaker, in helping to distribute the shock of impact at the loading point over a greater cover area. Table 20-8 is a guide to its use.

Table 20-8

Cover gauge, in.	*Breakers*
$\frac{3}{32}$, $\frac{1}{8}$, or $\frac{3}{16}$	Regular leno breaker included
$\frac{1}{4}$, $\frac{5}{16}$	Include one transcord breaker next to carcass
$\frac{3}{8}$	Include one transcord breaker next to carcass and one floated $\frac{1}{16}$ in. below top surface of the belting

Back-cover Thickness

For light service and low-quality belting, $\frac{1}{32}$-in. back cover is used. For most other service conditions a $\frac{1}{16}$-in. back cover is used. This is especially true when vulcanized splices are used to make the belting endless, as it increases the life expectancy of the splices. For sticky conditions, or where belting may freeze to the idlers or the pulleys, or where the back cover is subjected to a lot of wear or abuse, $\frac{3}{32}$-in. back cover, including regular breaker, or heavier cover is indicated. For very severe conditions, especially where large lumps are handled and the belting is subjected to heavy impact, it is desirable to use a transcord breaker in the back cover.

NOTE: Regular breaker displaces $\frac{1}{32}$ in. of cover thickness, and the minimum cover thickness in which it can be included is $\frac{3}{32}$ in. The transcord breaker displaces $\frac{1}{16}$ in. of cover thickness, and the minimum cover thickness in which it can be included is $\frac{1}{4}$ in. However, while the breaker actually displaces that amount of cover, it is still mainly rubber and does not materially decrease the wear resistance of the cover by its over-all thickness.

EQUIPMENT OVER WHICH THE BELTING MUST OPERATE

Space does not permit a detailed discussion of this part of the conveyor problem. Certain points will be stressed since they govern to a large extent the satisfactory performance of the belting. A number of belt users fail to realize that the belting on the conveyor represents from 40 to 60 per cent of the initial cost of the conveyor installation and also that the belting usually requires replacement before any of the equipment over which it operates.

Types of Drives

There are three types of drives in common use:

1. **Plain Drive.** The plain drive can be either at the head or tail pulley and gives only a 180-deg arc of contact between the belt and the driving pulley.

2. **Snubbed Drive.** On the snubbed drive a snub pulley smaller than the driving pulley is installed to increase the arc of contact between the belt and the driving pulley. The degree of contact can be varied from 210 to a maximum of 260 deg provided the drive pulley is sufficiently large.

3. **Two-pulley Drive.** The common two-pulley drive is known as the tandem drive. Both pulleys are driven from one motor, and the two pulleys are geared together so that they both turn at the same number of rpm. A new drive involving two pulleys is known as the dual-motor drive where each pulley is driven by a separate motor. The motors can be so designed that the speed of the two pulleys will automatically compensate for changing belt length on the pulleys, commonly referred to as "creep."

The purpose of increasing the wrap between the belt and the pulley is to permit the belting to transmit the required horsepower with less initial tension, which results in less total tension in the belting when it is operating under load.

Types of Starting for Conveyor Drives

1. **Across the Line.** Owing to the fact that conveyors often have to be started fully loaded, it is necessary that motors with a high starting torque be used. This

type of starting is the hardest on the belting but can be used on conveyors that are lightly stressed.

2. Autotransformer Starting. In autotransformer starting a lower voltage is fed to the motor for a few seconds until the motor starts to turn over. Then the full voltage is applied to bring the motor up to speed. This is somewhat easier on the belting than the across-the-line type.

3. Resistance Type through a Number of Starting Points. The resistance type can be manually operated or automatic. This type permits a longer acceleration period which, if properly designed, reduces the starting stress on the belting. All belts that are started frequently and are stressed to their full rated strength should have this type of starting or a modified version of it.

4. The Use of a Fluid Coupling. The fluid coupling can be used on the drive shaft of the motor in conjunction with some form of types 2 or 3 starting.

Pulleys, Bearings, and Idlers

Pulleys. Pulley diameters should not be skimped on because of limited headroom unless it is absolutely necessary. Theoretically, it is the amount of wrap between belt and pulley that determines the amount of power which can be delivered before belt slippage occurs, and not the area in contact between belt and pulley. Theoretically, as much wrap can be obtained between a belt and a small pulley as between it and a large pulley. But the stiffness of the belting makes a difference in favor of the larger pulley. It also stands to reason that the plies in a belt are being distorted as they bend over a pulley, the outer plies being stretched and the inner ones put under compression. The amount of distortion increases with the thickness of the belting; increasing the pulley diameter permits this deformation to take place over a longer length. Naturally the higher the working stress in the belting at the time the distortion is taking place, the more damage to the belting. Such severe flexing usually shows up first in short splice life, particularly when the metal type of splice is used.

At the points in an installation where the belting is fully stressed, flat-faced rather than crowned pulley surfaces should be used so that the load in the belting can be equally distributed across the full width of the belting. It is emphasized that the drive, head, and tripper pulleys should all be of the same diameter because they are all working under the same belt stress. It has always been common practice to make the head and tripper pulleys smaller than the drive pulleys in a conveyor that uses a drive which is separate from the head pulley and which may or may not be equipped with a tripper (see Table 20-9 for minimum pulley diameters).

Bearings. On long center, horizontal, or inclined conveyors, it is good design to use either ball or roller bearings on all pulley shafts.

Idlers. It is suggested that consideration be given to the use of larger diameter idler rolls. The larger rolls permit lower rpm with resultant longer bearing life and are somewhat easier on the belt owing to having a greater moment arm for turning. They also give smoother movement of load with less load shifting from slight belt sag. Roll diameter of at least 6 in. should be used, and some of the better conveyor systems have proved the economy of 7- and 8-in.-diameter rolls on large heavily loaded conveyor systems.

The use of three pulley-type carrying idlers with all three rolls the same length and the two troughed rolls placed at 20-deg angle to the horizontal is almost universally accepted practice. There is a trend toward the use of the same design except that the center roll is made about a third longer and the troughing rolls proportionately shorter but still at the same degree of troughing. It has been proved that such an idler will carry about 5 per cent more load without spilling, will give a straighter running belt (because of better belt troughing possibilities), and will give longer belt life because of less concentrated bending of the belting at the junction of the horizontal and troughing rolls.

Idler Spacing. Because belt sag between idlers is a function of the load per foot of belting, the tension in the belting, and the distance between idlers, much better support can be given to the belting by varying the idler spacing. At the point of lowest belt tension (usually the tail end of the conveyor), the idlers should be placed close together, and at the point of high belt tension they should be placed farther apart. The total number used will be the same as though they were uniformly spaced for the entire length of the conveyor.

Recommended Pulley Diameters. Table 20-9 lists minimum pulley diameters for installations where the belt is designed for the plies to take the maximum fabric stresses as given in Table 20-4 If it is certain that the belt will never be stressed above 70 to 80 per cent of its maximum, then the minimum pulley diameters may be taken at tabulated value minus 6 0 in.

Table 20-9. Minimum Pulley Diameters, In.

No of plies	Drive, head or tripper pulley	Tail take-up or high tension snub	Bend pulley or low-tension snub	Drive head or tripper pulley	Tail, take up or high-tension snub	Bend pulley or low-tension snub
	28 oz			32 oz		
3	18	12	12	18	18	12
4	24	18	12	24	24	18
5	24	24	18	30	24	18
6	30	24	18	36	30	24
7	36	30	24	42	36	24
8	42	30	24	48	36	30
9	48	36	30	54	42	36
	36 oz, 42 oz, or cord No 50			Cord No 70 or No 100		
3	30	24	18			
4	30	24	18			
5	36	30	24	42	36	24
6	42	36	24	48	36	30
7	48	36	30	54	42	36
8	54	42	36	60	48	36
9	60	48	36	60	54	42
10	60	54	42	72	60	48

Steel cord

Class	Drive head or tripper pulley	Tail take-up or high-tension snub	Bend pulley or low-tension snub
1000	54	42	36
1500	60	48	36
2000	60	48	36
2500	60	54	42
3000	72	60	48

NOTE Use the following standard pulley diameters if at all possible 12, 18, 24, 30, 36, 42, 48, 54, 60, 72 84, and 96

Take-ups for Conveyors

There are two types of take-ups for conveyors in common use:

1. **Screw.** The screw type of take-up moves the head, tail, or take-up pulley and positively fixes the centers. It is usually used on short-center conveyors and necessitates higher stresses in the belting than are actually necessary for belt operation.

2. **Gravity.** The gravity type of take-up keeps a constant tension in the belt and

is free to come and go with change in belt length. The head or tail pulley can be mounted on a movable carriage with a cable and weights attached to the carriage, or the belt slack side may be led over two bend pulleys with a weighted pulley suspended between.

Advantages of a Take-up of the Gravity Type. 1. A take-up of the gravity type maintains a constant T_2 tension necessary for the drive to operate the belt without excess slipping occurring.

2. It helps to keep belt sag between idlers at a minimum, particularly at the point of low belt tension, thus preventing load disturbance and saving power.

3. Permits belt-length change due to stretch or shrinkage and at the same time keeps the tension in the belt constant.

4. Takes care of the instantaneous belt stretch which occurs when the belt is started and maintains belt tension.

5. Particularly if vulcanized splices are used, it is well to provide take-up movement equal to 2 to 3 per cent of the center distance between head and tail pulleys. This will provide ample movement to accommodate any shrinkage or stretch that may occur. It will provide plenty of movement for the instantaneous stretch which occurs when belt is started, and it will provide storage space for spare belting in case of damage to the belt. It also provides extra belting for splice replacement if needed.

Calculating Take-up Weight. Care should be used in estimating the amount of take-up weight needed for any particular conveyor design. The take-up weight is equal to the sum of the component forces in the direction of belt tensions at the take-up pulley, wherever its location may be. The point of location is very important also. For horizontal conveyors, it is usually best to have the take-up located as near to the drive as possible and in the slack side of the belt. For inclined conveyors it is usually best to have the take-up at the tail pulley for uphill conveyors and at the head pulley for downhill conveyors.

There is another factor to be considered in calculating take-up weight. That factor is the idler retarding force on the return strand of belting, and it is equal to $F_R = 0.4(G \times F_x \times L_r)$. Space does not permit explanation regarding how to figure take-up weight for each of the examples given by Figs. 20-18 to 20-26, inclusive.

Vertical Curves

Vertical curves refer to the radius of curvature necessary on those conveyors which change from the horizontal or from a given degree of incline to a steeper degree. Frequently too small a radius is used with the result that the belt under tension raises from the idlers and damages itself or spills its load. It is much better to be sure and use a large radius. This is especially true when dealing with high-tension belts such as the No. 70, No. 100, and the five steel-cable cord constructions.

Table 20-10. Recommended Maximum Belt Speeds, Fpm, for Horizontal and Elevating Conveyors

Material to be conveyed	Belt widths, in								
	12	18	24	30	36	42	48	54	60
Light, free-flowing material	300	400	500	600	600	600	700	700	700
Coal and similar lump material	300	400	500	550	550	600	600	650	650
Heavy ore and abrasive material	300	350	450	550	550	550	600	600	600

The most efficient conveyor-belt operation is usually with the narrowest belt at the highest speed at which good loading is obtained and without doing any damage to the material. Increasing the belt speed without changing the rate of loading tends to lower the necessary belt tension. This is sometimes done on long-center conveyors to decrease the stresses.

Table 20-11

Slope angle of declining conveyor at loading point, deg	*Recommended maximum belt speed, fpm*
7 or less	90% of value in Table 20-10
8-14	75% of value in Table 20-10
15 and above	50% of value in Table 20-10

Table 20-12. Belt Capacity

Width of belt, in.	Wt per cu ft of material, lb	Capacity, short tons (2,000 lb) per hr												Cross section of load, sq ft
		Belt speed, fpm												
		50	100	150	200	250	300	350	400	450	500	550	600	
12	35	5	10	15	20	25	30							0.095
	50	7	14	21	29	36	43							
	75	11	21	32	43	54	64							
	100	14	29	43	57	72	86							
	125	18	36	54	72	90	107							
	150	22	43	65	80	108	129							
18	35	12	24	36	48	60	72	83	95					0.227
	50	17	34	51	68	85	102	119	136					
	75	26	51	77	102	128	153	179	204					
	100	34	68	102	136	170	204	238	272					
	125	43	85	128	170	212	255	298	340					
	150	51	102	153	204	255	306	357	408					
24	35	22	44	66	88	110	131	153	175	197	219			0.417
	50	31	63	94	125	156	187	219	250	281	312			
	75	47	94	141	187	234	281	328	375	422	468			
	100	63	125	188	250	312	375	438	500	563	625			
	125	78	156	234	312	390	468	546	624	702	780			
	150	94	188	281	375	468	562	656	750	843	936			
30	35	35	70	105	140	175	210	245	280	315	350			0.667
	50	50	100	150	200	250	300	350	400	450	500			
	75	75	150	225	300	375	450	525	600	675	750			
	100	100	200	300	400	500	600	700	800	900	1,000			
	125	125	250	375	500	625	750	875	1,000	1,125	1,250			
	150	150	300	450	600	750	900	1,050	1,200	1,350	1,500			
36	35	51	102	152	203	254	304	355	406	457	508	558	609	0.967
	50	73	145	218	290	362	435	507	580	652	725	797	870	
	75	109	217	326	434	542	651	760	868	976	1,085	1,195	1,300	
	100	145	290	435	580	725	870	1,015	1,160	1,305	1,450	1,595	1,740	
	125	181	362	543	724	905	1,085	1,270	1,450	1,630	1,810	1,990	2,170	
	150	218	435	653	870	1,090	1,310	1,520	1,740	1,960	2,180	2,390	2,644	
42	35	71	142	213	284	355	426	497	568	639	710	780	852	1.353
	50	102	203	304	406	508	609	710	812	913	1,016	1,120	1,220	
	75	152	304	456	608	760	912	1,065	1,218	1,370	1,520	1,670	1,825	
	100	203	406	609	812	1,015	1,218	1,420	1,625	1,825	2,030	2,230	2,440	
	125	254	508	761	1,015	1,270	1,525	1,780	2,030	2,280	2,540	2,790	3,040	
	150	304	609	915	1,220	1,520	1,830	2,130	2,440	2,740	3,040	3,350	3,650	
48	35	96	193	289	385	481	577	673	770	866	963	1,060	1,155	1.833
	50	138	275	413	550	688	825	963	1,100	1,240	1,375	1,510	1,650	
	75	206	413	620	825	1,030	1,240	1,445	1,650	1,860	2,060	2,270	2,480	
	100	275	550	825	1,100	1,375	1,650	1,925	2,200	2,480	2,750	3,020	3,300	
	125	344	688	1,030	1,375	1,720	2,060	2,410	2,750	3,100	3,440	3,780	4,130	
	150	413	825	1,240	1,650	2,060	2,480	2,890	3,300	3,710	4,125	4,540	4,950	
54	35	125	250	375	500	625	750	875	1,000	1,125	1,250	1,375	1,500	2 383
	50	179	358	537	716	895	1,075	1,250	1,430	1,610	1,790	1,970	2,150	
	75	268	536	805	1,070	1,340	1,610	1,875	2,140	2,410	2,680	2,950	3,220	
	100	358	715	1,072	1,430	1,790	2,140	2,500	2,860	3,229	3,580	3,930	4,290	
	125	447	894	1,340	1,790	2,240	2,680	3,130	3,580	4,020	4,470	4,910	5,360	
	150	536	1,072	1,610	2,140	2,680	3,220	3,760	4,290	4,830	5,360	5,900	6,440	
60	35	157	315	473	630	788	945	1,100	1,260	1,420	1,575	1,730	1,890	3.000
	50	225	450	675	900	1,125	1,350	1,575	1,800	2,020	2,250	2,480	2,700	
	75	337	675	1,010	1,350	1,690	2,020	2,360	2,700	3,040	3,370	3,710	4,050	
	100	450	900	1,350	1,800	2,250	2,700	3,150	3,600	4,050	4,500	4,950	5,400	
	125	562	1,125	1,690	2,250	2,810	3,370	3,940	4,500	5,060	5,620	6,180	6,750	
	150	675	1,350	2,020	2,700	3,380	4,050	4,720	5,400	6,070	6,750	7,420	8,100	

Cubic feet per hour for each 100 fpm belt speed = 6,000 × load cross section in square feet.

Especially with light free-flowing material the factor determining the maximum speed is the windage which may blow the material from the belt. The average speed of grain belts is 800 to 1,000 fpm, with some operating up to 1,200 fpm.

The type of chute or mechanical feeder will often determine the speed at which a belt may operate satisfactorily. If the carrying-idler spacing is graduated and the belt tension is maintained correctly so that the load travels smoothly, higher speeds are often possible. If it is necessary to load onto a belt where it is at a slope angle of 10 deg or more, slower belt speed may be necessary to get full-capacity loading.

When declining conveyors are considered, the amount of slope at the loading point is usually the determining factor as far as maximum belt speeds are concerned. If the loading point is on a horizontal run, the belt speed for the declining conveyor can be as high as that shown in Table 20-10. Keeping full rated load on the belt for its width and speed helps downhill operation.

Where it is necessary that the loading of a downhill conveyor be on the slope, lower speeds than specified in Table 20-10 are recommended. These are shown in Table 20-11.

The capacity at any belt speed and with any weight of material can be found by using the constants shown in Table 20-13.

Table 20-13

Belt width, in.	*Capacity in short tons (2,000 lb) per hr*
12	0.00286 × fpm × wt per cu ft of material, lb
18	0.0068 × fpm × wt per cu ft of material, lb
24	0.0125 × fpm × wt per cu ft of material, lb
30	0.0200 × fpm × wt per cu ft of material, lb
26	0.0290 × fpm × wt per cu ft of material, lb
42	0.0406 × fpm × wt per cu ft of material, lb
48	0.0550 × fpm × wt per cu ft of material, lb
54	0.0715 × fpm × wt per cu ft of material, lb
60	0.090 × fpm × wt per cu ft of material, lb

NOTE: The capacity values given are obtained with standard 20-deg troughing idlers. Consider flat conveyor belts as giving one-half the capacities shown.

Table 20-14. Size of Maximum Lumps

Belt width, in.	Max lump size, in.	
	Uniform size	If mixed with 90 % fines
12	2	4
18	4	6
24	5	8
30	6	10
36	7	12
42	8	14
48	10	16
54	11	20
60	12	24

With a heavy belt, very close idler spacing, and skirt boards the entire length, much larger lumps have been carried than shown in Table 20-14. In some cases material has been carried where all three dimensions are equal to the belt width, but this is detrimental to belt and idlers.

There are a number of other considerations which may affect the slope angle. If slope is over 16 deg, if belt is running too fast, or if it is only partially loaded, there is danger of lumps falling back down the slope. This is liable to be a hazard for the belting, equipment, and workmen. In several cases, swinging bars with a stop against the downslope side have been used. The bars swing forward to let the load pass through but if lumps start rolling back, the bars are held against the stop and the backward flow of material is dammed. Several of these swinging bar units are used on a

Table 20-15. Approximate Unit Weights of Various Materials

Materials	Weight, lb per cu ft
Minerals	
Coal, anthracite (broken and loose)	55
Coal, bituminous (broken and loose)	45
Ashes of bituminous coal	35–45
Ashes (damp)	40–50
Cinders (coal and ash)	40–45
Cinders (blast furnace)	57
Coke (loose)	23–32
Iron ore	110–160
Bauxite (crushed aluminum ore)	80
Gypsum (irregular lumps)	82
Salt	75–80
Rocks	
Gneiss	96
Granite	97
Limestone	96
Sandstone	86
Marble	96
Quartz	94
Slate	95
Metals	
Cast iron (solid or compact weight)	450
Wrought iron (solid or compact weight)	480
Steel (solid or compact weight)	490
Brass (cast) (solid or compact weight)	505
Lead	711
Zinc	438
Masonry	
Cement (portland)	90–100
Cement mortar	120
Concrete (cinder) solid weight	115
Concrete (sand and gravel) solid weight	150
Brickwork solid weight	115
Earth	
Loam (loose)	76
Clay (loose and dry)	70
Mud	110
Sand (wet)	120
Sand (dry)	100
Gravel (wet)	125
Gravel (dry)	112

	Lb per cu ft	Lb per bu
Wood		
Wood chips	20	25
Wood pulp (damp)	30	37
Grain		
Wheat	48	60
Barley	38	48
Shelled corn	45	56
Flour	56	70
Oats	26	32
Rye	45	56

NOTE One bushel equals 1.244 cu ft The solid or compact weights of the various materials cannot be used in considering the capacity of a conveyor belt because the belt usually carries crushed material. A close approximation of the crushed weight of an ore or rock equals solid weight divided by 1.75. All the weights in Table 20-15 are for crushed material as would be carried by a conveyor belt except those specified as being for a solid. Wherever possible, the actual material weight as determined by test should be used

Table 20-16. Maximum Incline Angles

Material handled	*Max angle, deg*
Dry silica sand	15
Paper-wrapped packages with flat surfaces on belting with standard canvas or rubber cover	16
Free-flowing materials such as washed gravel, fine-sized coal, grain, etc	18
Size lump materials such as stone, lump coal, ore, etc., of less than ½-in. size	22
Granular materials such as sand, glass batch, or salt	22
Powdered materials such as lime, cement, sulfur, chemicals, etc	23
Mixtures of lumps and fines such as run-of-mine coal, crushed stone, etc	24
Tempered molding sand	24
Wood chips	27
Paper-wrapped packages with flat surfaces and soft bagged material or special "grip-top" belting	28

slope. If material is wet and sloppy, the angles shown in Table 20-16 must be decreased. If there is slip between material and belt, excess wear will occur at belt-cover gouges and fasteners.

TRIPPER HORSEPOWER AND REDUCTION-DRIVE LOSSES

Tripper Horsepower. As stated on page 1421, the horsepower necessary for tripper operation must be included with the $x + y + z$ horsepower factors to get total power necessary to be transmitted to the belt.

The horsepower necessary to lift the load at the tripper should be included in the lift-horsepower calculation for the conveyor. Therefore, when tripper is used, its lift in feet should be included in the H value in the Z formula. If exact lift at tripper is not known, add 4, 5, 6, 7, or 8 ft, depending on belt plies which are estimated to be necessary.

Table 20-17. Value of *G*

Belt width, in.	*G*		
	Light conveyors	Standard conveyors	Heavy-duty conveyors
14	9	14	
16	10	16	
18	11	17	
20	12	19	
24	15	22	32
30	18	26	45
36	21	32	58
42	24	45	71
48	29	52	84
54	35	61	97
60	38	71	110

If the value of *G* is to be calculated, belt, idler, and pulley weights must be known.

To compensate for the added moving parts when tripper is used, increase the G value by 5 per cent.

Therefore, if tripper is to be used and a 30-in. belt on heavy-duty conveyor is being considered, the H and G values might be increased as follows:

H should include 6 ft of tripper lift. Increase G by 5 per cent, or from 45 to 47.3.

If G, Table 20-17, is being calculated, then the weight of tripper pulleys should be included to get the correct value.

Horsepower for Drive Losses. *Type of Reduction Units.* For cast-tooth gears, add 10 per cent for each reduction.

For cut-tooth gears, belting, or roller chain, add 5 per cent for each reduction.

For self-contained spur- or helical-gear reducers, add 5 per cent for each reduction.

Considering an average reduction to be a gear-reducer and roller-chain combination and 320 hp calculated to be necessary to drive loaded belt at drive pulley or pulleys, then

$$320 \times (5\% + 5\%) = 320 \times 10\%, \text{ or } 32 \text{ horsepower}$$
$$\text{Estimated motor output or rating} = 320 + 32, \text{ or } 352 \text{ hp}$$

A motor and also gear, chain, or belt reductions are most efficient at their rated loads. The motor and reduction units should be chosen so that they will be operating as nearly as possible to full-load rating.

G (Weight of Moving Parts)

G may be taken from Table 20-17 if it is not convenient or possible to calculate. For large installations, it is best to obtain the details and calculate.

Calculation of G. Since G equals the weight of moving parts per foot of conveyor centers, its value may be calculated easily by taking the sum of the following four factors:

1. Total weight of the entire length of belting used. The entire belting length equals approximately twice the conveyor center length.
2. Total weight of pulleys which are turned by the belt.
3. Total weight of moving parts of all troughing idlers.
4. Total weight of moving parts of all return idlers.

Then divide this sum by the length of conveyor centers as measured along the belt.

Belt Weights. Weights (Table 20-18) are given on three grades of belting specified as first, second, and third quality. First quality are reserved for the very severest of service, and the weights are based on compounds made from natural crude rubber. Second grade are for average-severe service and third grade for medium or light duty.

Table 20-18

Duck weight, 42 in. wide × 36 in. long	Grade	Belt weights, approx net weight lb per in. width per ft length
28 oz	First	0.026 per fabric ply 0.017 per $\frac{1}{32}$ in. of cover
	Second	0.026 per fabric ply 0.017 per $\frac{1}{32}$ in. of cover
	Third	0.022 per fabric ply 0.017 per $\frac{1}{32}$ in. of cover
32 oz	First	0.028 per fabric ply 0.017 per $\frac{1}{32}$ in. of cover
	Second	0.028 per fabric ply 0.017 per $\frac{1}{32}$ in. of cover
	Third	0.024 per fabric ply 0.017 per $\frac{1}{32}$ in. of cover
36 oz	First	0.030 per fabric ply 0.017 per $\frac{1}{32}$ in. of cover
	Second	0.030 per fabric ply 0.017 per $\frac{1}{32}$ in. of cover
42 oz	First	0.032 per fabric ply 0.017 per $\frac{1}{32}$ in. of cover
	Second	0.032 per fabric ply 0.017 per $\frac{1}{32}$ in. of cover
No. 50 cord*	First	0.031 per ply 0.017 per $\frac{1}{32}$ in. of cover
	Second	0.031 per ply 0.017 per $\frac{1}{32}$ in. of cover
No. 70 and 100 cord*	First	0.031 per ply 0.017 per $\frac{1}{32}$ in. of cover
Steel cord, No. 1000*	First	0.345 for carcass per in. of belt width 0.017 per $\frac{1}{32}$ in. of cover
Steel cord, No. 1500*	First	0.385 for carcass per in. of belt width 0.017 per $\frac{1}{32}$ in. of cover
Steel cord, No. 2000*	First	0.425 for carcass per in. of belt width 0.017 per $\frac{1}{32}$ in. of cover
Steel cord, No. 2500*	First	0.465 for carcass per in. of belt width 0.017 per $\frac{1}{32}$ in. of cover
Steel cord, No. 3000*	First	0.505 for carcass per in. of belt width 0.017 per $\frac{1}{32}$ in. of cover

* These constructions are made by the B. F. Goodrich Company. For special constructions made by other manufacturers it will be necessary to get the correct belt weight from the manufacturer.

The approximate weights of terminal, snub, or drive pulleys are given in Table 20-19.

For any important installations where an exact value of G should be used, obtain the exact weight of moving parts from the equipment manufacturer.

F (Coefficient of Rotational Friction of Pulleys and Idlers)

Tests of actual conveyor systems have indicated that the greatest accuracy can be obtained in the horsepower calculation by using two values of F. One value should

Table 20-19. Pulley Weights

Pulley diameter, in.	To calculate approx pulley weight, lb	
	Cast-iron pulleys	Welded-steel pulleys
12	7 lb × face width, in.	3 lb × face width, in.
18	11 lb × face width, in.	6 lb × face width, in.
24	16 lb × face width, in.	10 lb × face width, in.
30	22 lb × face width, in.	13 lb × face width, in.
36	29 lb × face width, in.	17 lb × face width, in.
42	37 lb × face width, in.	
48	46 lb × face width, in.	
54	56 lb × face width, in.	
60	75 lb × face width, in.	
72	98 lb × face width, in.	
84	120 lb × face width, in.	
96	142 lb × face width, in.	

These values are approximate pulley weights. Use the pulley manufacturers' value of weight whenever available.

Table 20-20. Weight of Idler Moving Parts

Belt width, in.	Approx weights of moving parts of steel idlers, lb							
	Troughing-idler diameter, in.				Return idler diameter, in.			
	4.0	5.0	6.0	7.0	4.0	5.0	6.0	7.0
14	14	26	35	...	12	18	24	
16	15	27	37	...	13	20	26	
18	16	29	39	...	14	21	28	
20	17	31	41	...	15	23	30	
24	19	34	45	...	17	26	34	
30	.	39	52	...	..	31	40	
36	..	44	58	125	..	36	46	97
42	..	49	65	137	..	41	52	109
48	.	54	71	150	..	46	58	122
55	..	..	78	163	..		65	134
60	..	..	86	174	..	.	71	147

Special heavy-duty types of steel troughing idlers may weigh 30 per cent over the above figures and cast-iron idlers will weigh approximately 60 per cent more than above.

be used for calculating the horsepower necessary to drive the empty belt, and a slightly larger value of F gives greater accuracy for figuring the horsepower to carry the load horizontally.

Values given in Table 20-21 for the most ideal condition should *rarely be used.* Horsepowers calculated with F_x and F_y of 0.025 and 0.030 have seldom been attained in actual service. There are only one or two cases where this was accomplished, owing to

1. Highest grade antifriction idlers used
2. Graduated spacing for troughing idlers
3. All terminal and bend or snub pulleys mounted with antifriction bearings
4. Experiments made to determine belt tension necessary for least horsepower
5. Very superior maintenance practiced

Cold-weather operation tends to raise the coefficient of friction slightly. Tests have indicated that for 20°F operation the coefficient-of-friction values should be raised approximately 10 per cent. This increase has very little effect on the horsepower necessary for a conveyor unit which has most of its need of power in elevating

Table 20-21. Values for F

Type of bearing and condition of conveyor	F_x	F_y
Most ideal	0.025	0.030
High grade	0.030	0.040
Average to be used if no details known except that antifriction idlers are used	0.035	0.050
Plain bearing	0.050	0.075

the load. The effect is greater on long horizontal installations where most of the total power is for moving the empty belt and carrying the load horizontally.

Actual field tests have indicated slightly lower coefficients of rotational friction than most of the values in Table 20-21. However, owing to the many variables of equipment maintenance, operating temperatures, etc., Table 20-21 values are slightly on the safe side.

Table 20-22

Angle of belt wrap at drive, deg	Type of drive	Drive factor K*			
		Screw take-up		Gravity-weighted or flexible take-up	
		Bare pulley	Lagged pulley	Bare pulley	Lagged pulley
180	Plain	1.6	1.0	0.84	0.5
190	Snubbed	1.5	0.9	0.77	0.45
200	Snubbed	1.4	0.8	0.72	0.42
210	Snubbed	1.3	0.7	0.67	0.38
220	Snubbed	1.2	0.6	0.62	0.35
230	Snubbed	1.2	0.6	0.58	0.32
240	Snubbed	1.0	0.5	0.54	0.30
360	Tandem	0.8	0.4	0.26	0.125
380	Tandem	†	†	0.23	0.113
400	Tandem	†	†	0.21	0.095
420	Tandem	†	†	0.19	0.084
440	Tandem	†	†	0.17	0.074
460	Tandem	†	†	0.15	0.064
480	Tandem	†	†	0.14	0.056

* The drive-factor constant multiplied by effective tension T_E equals slack-side tension T_2.

† Not recommended.

NOTE: A 210 deg arc of contact on snubbed-head pulley drives and a 420 deg arc of contact on tandem or dual-motor drives are most frequently used. Belt wraps above 220 deg are not recommended for use with screw take-up but must be used in some cases for underground service. The trend is toward the use of snubbed-head pulley drives with 240 deg arc of contact on long-slope conveyors where a high belt-slope tension exists.

Then for plain, snubbed, or tandem drives

$$T_E = (\text{hp} \times 33{,}000)/S$$
$$T_2 = KT_E$$
$$T_1 = T_E + T_2 = (K + 1)T_E$$

With reference to the maximum plies as indicated in Table 20-23, a greater number than nine plies of 28- or 32-oz duck would trough in the 48-, 54-, or 60-in. belt widths, but if more than nine plies are needed, a heavier weight of duck should be used. Likewise, 10 plies in the 36- and 42-oz constructions should be considered maximum for good balance. More than 10 plies would trough in the 42-, 48-, 54-, and 60-in. widths with the cord constructions, but consider 10 plies as the maximum to be used. The steel-cord belts will trough well in all widths of 24 in. and over. Probably 24 in. will be the minimum width for use of steel cord.

Table 20-23. Maximum and Minimum Plies for Troughed Belts

Belt width, in.	Minimum plies to support load															Maximum plies for troughing				
	Light, free-flowing material		Fine coal or crushed stone up to 3 in. size					Lump coal and similar material					Heavy ore and similar material							
	28 oz	32 oz	28 oz	32 oz	36 oz	42 oz	Cord No. 50, No. 70, No. 100	28 oz	32 oz	36 oz	42 oz	Cord No. 50, No. 70, No. 100	32 oz	36 oz	42 oz	Cord No. 50, No. 70, No. 100	28–32 oz	36 oz	42 oz	Cord No. 50	Cord No. 70, No. 100
12	3	3	..	..	..	..	3	..	..	..	..	..	..	..	..	..	3	..	..	3	
18	4	4	4	4	4	..	4	4	4	4	..	4	..	..	..	..	4	..	..	5	
24	4	4	4	4	4	3	4	4	4	4	..	4	5	4	..	5	5	4	3	7	6
30	4	4	5	5	5	4	4	5	5	5	4	4	6	6	4	5	6	5	4	9	8
36	4	4	6	5	5	4	4	6	6	6	4	5	7	6	5	5	7	6	5	10	9
42	4	4	6	5	5	5	5	7	7	6	5	6	8	7	6	6	9	8	6	10	10
48	4	4	6	5	5	5	5	8	7	7	5	6	8	7	7	7	..	10	8	10	10
54	..	.	..	7	6	6	6	..	8	7	6	6	..	..	8	8	..	..	10	10	10
60	..	.	..	7	7	6	6	..	8	7	6	6	..	..	8	8	..	..	10	10	10

FREIGHT ELEVATORS

BY E. M. BOUTON

Manager of Engineering, Elevator Division, Westinghouse Electric Corporation

INTRODUCTION

Freight elevators are used for handling material in many different types of buildings, such as factories, warehouses, and department stores, in fact, in every type of building where material must be moved from one floor to another. The kind of material handled and the manner in which it is packaged and moved on and off the elevator vary greatly, as do the amount of material to be handled and the frequency of the service which is required. In general, as compared with passenger elevators, freight elevators handle much heavier loads, but the speed required of the car is not so great, owing largely to the fact that the loading and unloading is such a large percentage of the total cycle. The usual range of capacity of freight elevators is from 2,500 to 20,000 lb, although in special cases even larger capacities are required.

Speeds used vary from 50 to 200 fpm for most applications, although here, also, are found occasional requirements for speeds up as high as 500 fpm.

The appointments of the freight-elevator cars are much simpler than passenger elevators, there being no particular requirements for ornamentation. Ruggedness and safety are the most essential characteristics.

At the present time practically all freight elevators are of the electric type, although there are a few low-rise applications where the hydraulic lift may be employed to good advantage. Such cases are limited to a very low rise, not over 25 ft, and where the speed can be very low, usually not over 35 fpm.

Material handling in modern manufacturing operations has, in recent years, been receiving increasing attention. An analysis of manufacturing costs shows that the cost of moving material is one of the largest single elements in cost analysis. In the past much attention has been given to speeding up machining operations involving the use of high-speed steel so that cost of cutting operations has been greatly reduced. The use of spraying with special spray guns combined with conveyors has greatly reduced the cost of applying paint and finishes.

It is only recently, however, that the cost of moving material from one position to another has received serious consideration. During the Second World War, however, great impetus was given to the methods of material handling, resulting in increased use of power trucks and palletizing of material. The freight elevator must be considered one of the elements of a material-handling system, and many of the recent developments in material handling have definitely affected the design of freight elevators and have imposed new requirements which must be met by this equipment. Fortunately, considerable development has taken place in freight elevators and modern industrial freight elevators, if properly applied, will in many cases contribute greatly to the efficiency of the material-handling problems of a manufacturing plant.

ELEVATOR MACHINES

The *traction machine* came into use about 1900 and today has practically replaced the drum machine except for a few specialized applications. In the drum machine the hoisting ropes, or cables, used for hoisting the car are wound up on a drum, one end of the ropes being attached to the car and the other attached solidly to the drum. These machines have a number of serious limitations which the traction machine does not have. In the traction machine the ropes are attached to the car and pass over the machine sheave, or drum, the opposite end of the rope being attached to the counterweight. The car is driven by the friction, or traction, between the rope and the sheave. The traction machine is the safest type that has ever been evolved in that any danger of overwinding has been entirely removed. Should the car overtravel at the top, the counterweights will rest on the bottom of the pit, traction will be lost, and overwinding thus prevented. The traction machine has great flexibility in that the same sheave may be used for any height of building. Traction machines may be either geared or gearless. In the gearless machine the driving sheave is mounted directly on the motor shaft, the rpm of the sheave being the same as that of the motor armature. This type of machine is the one most suitable for high-speed passenger elevators but is used for freight elevators only in special applications. Where heavy loads are to be lifted, the high torque requirements of the motor make it prohibitive in cost so that for freight elevators the geared-type machine is used. Figure 20-27 illustrates a geared traction machine of the type that is most generally used for freight elevators. In this machine the gear is mounted on the drum shaft, and the drum shaft with its gear and drum are supported by heavy roller bearings. Since the entire weight of the car, counterweights, and load is supported by this shaft, these bearings are made extremely rugged. This gear meshes with the worm, the worm shaft being coupled directly to the motor shaft. The thrust developed by the gear is taken by substantial thrust bearing on the end of the worm shaft. In this machine the motor is flange mounted to the machine housing, which eliminated any problem of alignment of the motor shaft and the worm shaft. The essential characteristics of this machine are simplicity and ruggedness, making it well adapted to the requirements of freight elevators in modern industrial plants.

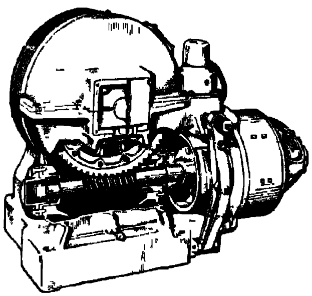


FIG. 20-27. Worm-geared traction elevator machine.

The methods of roping used with traction elevators are 1:1 (see Fig. 20-28), 2:1 (see Fig. 20-29), and 3:1 (see Fig. 20-30).

It is the usual practice to make the counterweight equal to the weight of the car plus approximately 40 per cent of the rated load in the car. Where the rise is more than 100 ft, it is the usual practice to hang compensating ropes from the bottom of the car to the bottom of the counterweight so that the weight of the car does not change appreciably at different positions in the shaft.

Roping with a 1:1 ratio is very generally used for passenger elevators where the load is comparatively light and the speed high. It is also used for freight elevators, but in this case the torque required by the drum shaft becomes very high for heavy loads, and a double reduction gear is required. This involves the introduction of spur gearing between the worm wheel and the sheave and complicates the design of the elevator machine. In order to use the simple worm-gear machine, 2:1 roping is very extensively used for heavy industrial freight elevators. With 2:1 roping the car speed is one-half the rope speed, but the load that can be lifted for a given pull on the rope is twice as great as 1:1 roping. Roping with a 3:1 ratio is occasionally used where very heavy loads must be lifted and the speed is low.

Traction drives may be further classified as single wrap and double wrap. The double-wrap traction is illustrated in Fig. 20-31 and has the same general characteristics as the single wrap shown in Fig. 20-28, but in this case an additional wrap is made about the driving sheave, thus increasing the traction for driving. It also doubles the load in the sheave bearings. The sheave groove in this case is known as U grooving, as illustrated in Fig. 20-31. This system is useful for high-speed freight elevators using a gearless machine. For freight elevators the single-wrap traction is generally used, in which case the grooves have an undercut, as shown in Fig. 20-28. This particular shape of groove gives high traction and good support for the rope.

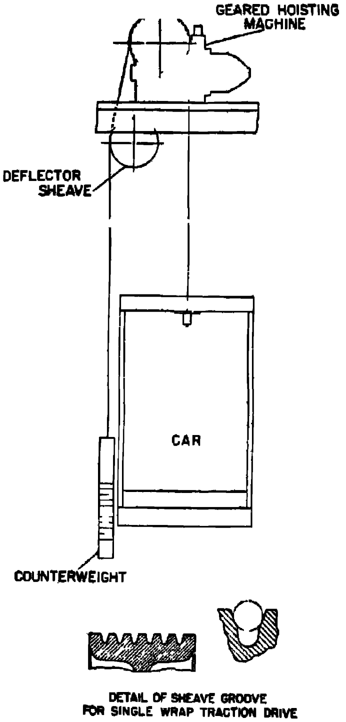


Fig. 20-28 Single-wrap traction elevator with 1:1 roping showing detail of sheave groove

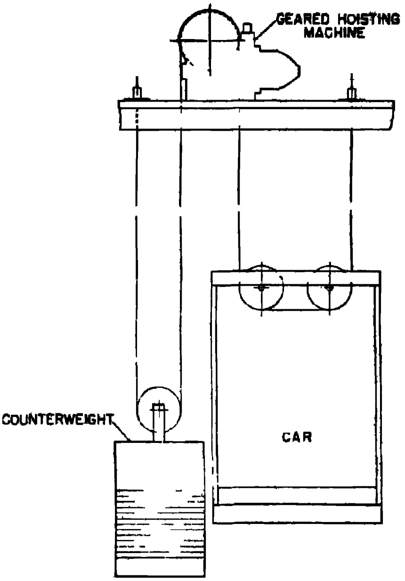


Fig. 20-29 Single-wrap traction elevator with 2:1 roping (Sheave grooves same as Fig. 20-28.)

Furthermore, the traction does not change as grooves wear, thus ensuring a long life of both sheave and rope.

The car is hoisted by means of wire rope. Elevator hoisting rope is a rather complicated structure, composed of a number of strands of wire which are wound upon a hemp center. The hemp center forms a support for the strands and carries a lubricant. The various strands must move freely with respect to one another to prevent internal wear. In order to obtain a satisfactory rope life, the ratio of sheave diameter to rope diameter should be kept as high as practicable. The American Standard Safety Code requires that the sheaves be at least 40 times the rope diameter. In practice, a ratio of 50:1 or more is usually used. Wire rope is manufactured in a number of constructions. For traction elevators it is desirable to have the outside wires in the strand large in order to prevent breakage due to wear from contact with the driving sheave.

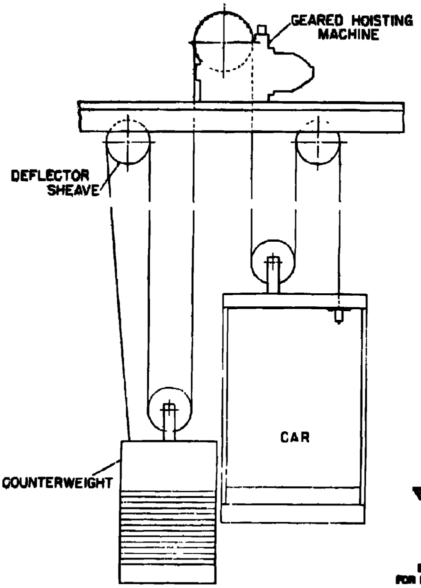


Fig. 20-30. Single-wrap traction elevator with 3:1 roping. (Sheave grooves same as Fig. 20-28.)

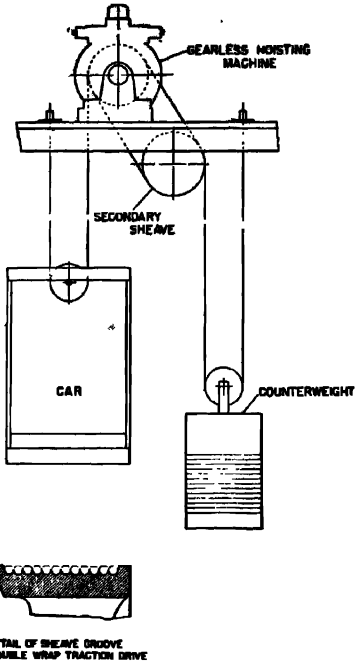

GEARLESS HOISTING MACHINE
SECONDARY SHEAVE
CAR
COUNTERWEIGHT
DETAIL OF SHEAVE GROOVE FOR DOUBLE WRAP TRACTION DRIVE

Fig. 20-31. Double-wrap traction elevator with 2:1 roping showing detail of sheave groove.

MOTORS AND CONTROL

Both a-c and d-c motors are used for elevator service. The simplest and most rugged type of motor is the single-speed squirrel-cage motor with high starting characteristics. Such motors are designed to give the maximum torque at starting. To obtain the high starting torque, a high rotor resistance is required so that the motor has considerable slip, and its speed regulation is not so good as that of the d-c motor. It is started by connecting it directly to the line, usually with a step or two of primary resistance, and is stopped by disconnecting it and bringing the car to rest by the friction brake on the elevator machine. No speed control can be obtained so that this type of motor is limited to low-speed elevators, 75 to 100 fpm, and where accurate stopping is not important and where self-leveling is not required. A modification of this motor is known as the two-speed motor, in which the motor has two sets of poles, thus giving two speeds, or has the primary reconnected on slowdown so as to change the number of poles. The starting characteristics of such a motor must be the same as for a single-speed motor, but by utilizing the greater number of poles the slower speed is available for stopping. Such motors have been built with pole changers with 6:1, 4:1 and 3:1 speed ratios and with reconnected primary with a 2:1 speed ratio. Such motors have been extensively used in the past as a means of obtaining higher car speeds or to obtain more accurate landing. In recent years, however, the development of variable-voltage control in smaller ratings gives much better performance, more accurate control, so that the use of two-speed a-c equipment has greatly decreased. Variable-voltage equipment may now be obtained in ratings as low as 10 hp so that there is very little need for an a-c motor for car speeds above 100 fpm.

In *variable-voltage* control a d-c elevator motor is used, and the speed and direction of rotation of the motor are obtained by controlling the field of a generator. In this case, there is a motor-generator set for each elevator. The generator may be driven by either an a-c or d-c motor, most of the drives being alternating current. The motor and generator are combined in one unit, or motor-generator set, which also includes an exciter. For speeds up to 200 fpm, which are usual for freight elevators, a compound-wound generator is used, compounding being obtained by a series field on the generator used for the purpose of compensating for load and giving good speed regulations both at high and low speed. The great advantage of variable-voltage control for any kind of elevator service is the smoothness with which the elevator operates and the accurate speed control which can be used for making landings. This good speed control is particularly necessary where automatic self-leveling is required, in that the car must be moved at a very low speed so as to obtain the required stopping accuracy.

Variable voltage is recommended for all freight elevators above 100 fpm and for lower speeds with 10,000 lb capacity or greater.

The control operating systems used for freight elevators are as follows:

1. Collective
2. Push button (single party)
3. Car-switch control
4. Constant-pressure push button

The system that is the most serviceable and best adapted to freight-elevator service is collective operation. With this system there are UP and DOWN buttons in the corridor on each floor and a set of buttons in the car corresponding to the floors. The car is started in response to the pressure of these buttons. With selective collective operation, which is the system most generally used, the car on its up trip responds only to up calls in the corridor and on the down trip responds only to down calls in the corridor. As it travels through the hoistway, it automatically stops for the calls received and in the natural order of the floors. All calls are stored so that, after completing a trip in one direction, the car will automatically start and answer all the calls in the other direction in the natural order of the floors served. While making a trip in either direction the car will also stop in response to any buttons that may be pressed within the car. When this system is equipped with an attendant service feature and annunciator in the car, it may be used with an attendant in which case the corridor buttons do not start and stop the car but register signals on the annunciator. The attendant controls the car by means of the car buttons. This system may be equipped with automatic self-leveling so as to maintain the car level with the floor during loading and unloading.

Where the service is less severe, push button operation of the single-party type may be satisfactory. With this system the car may be called to any floor by means of the corridor buttons or sent to any floor by means of buttons in the car. Since it responds to but one call at a time, it is limited to relatively infrequent service.

Car-switch operation is a system in which the car is controlled by a lever in the car and requires an attendant. This system may be equipped with automatic self-leveling. In order to eliminate the need of an attendant, a modification known as constant-pressure push-button operation is sometimes used. It has an UP button and a DOWN button in the car by means of which the car is started and stopped and buttons in the corridor which also control the car. The car is not automatically stopped but stops when the operating button is released. It is therefore necessary, when calling the car from a corridor button, to observe the position of the car through a vision panel to see that it is at the floor level. It is suitable only for slow-speed elevators where the service is not severe.

MECHANICAL EQUIPMENT

The *shaft*, or *hoistway*, in which an elevator runs is a very important element in its safe operation. It forms part of the building structure and contains the guide rails on which both the car and counterweight travel. The enclosing walls and the landing doors should be fire resistant. The building structure itself must furnish the proper support for the elevator and sufficient structure to which to attach the elevator guide rails. This is particularly important for freight elevators where power trucks are run on and off the car and where the car platform is subject to eccentric landing. At the bottom of the shaft, known as the pit, are located the terminal stopping devices, consisting of control limit switches, a governor-rope sheave, and the car buffer.

Car buffers may be either of the spring or oil type. Most elevators below 200 fpm use spring buffers, which covers a large number of freight elevators. For higher speeds an oil buffer is used. Oil buffers are designed to stop the car from governor tripping speed with an average retardation of gravity, and the American Safety Code for Elevators requires that the maximum instantaneous retardation shall not exceed 2½ times gravity.

The most suitable location for the elevator machine is directly above the hoistway, as, in this case, the roping is simpler, and the load on the building structure is less than when the machine is mounted in the basement. The building structure must be designed to support the weight of the loaded car and its counterweights, plus the weight of the machinery when located overhead.

Every elevator is equipped with an undercar safety which grips the guide rails in case of broken hoist ropes or in case of overspeed. This safety is actuated by a speed governor located overhead. There are two classes of safeties generally used:

1. Instantaneous safeties
2. Sliding safeties

Where the car speed does not exceed 100 fpm, the instantaneous, or roller-type, safety is used. In the instantaneous safety a roller is wedged between the safety jaw and the elevator guide rail, and the car is brought to rest with practically no slide. Where the speed exceeds 100 fpm, sliding safeties in a variety of forms are used. A very common type of safety is known as the wedge clamp. When the governor trips, the governor rope is brought to rest, causing a drum on the safety to turn, thereby applying the safety jaws to the rail. An improvement on the wedge-clamp safety is to make the jaws flexible so that the operation of the safety drum compresses a spring in the jaw mechanism, thus giving a uniform braking action which it is possible to adjust to any desirable value.

Every elevator should be equipped with terminal slowdown and stopping switches which will act independently of the operator to slow the car down and bring it safely to rest at the limits of travel.

All elevator doors should be equipped with electric contacts so that the car cannot be moved with the doors open. The doors should also be equipped with locks so designed that they cannot be opened on the outside unless the car is at the floor. Devices in which the lock and the contact are combined in such a way that the car cannot be moved unless the door is both closed and locked are known as interlocks. A device in which the contact can be made while the door is still unlocked, even though the locking mechanism is in a position to lock the door when the car leaves, is known as a contact and lock. American Standard Safety Code for Elevators describes these devices and establishes rules for their design, approval, and use.

APPLICATIONS OF FREIGHT ELEVATORS

In selecting a freight elevator for a factory, warehouse, or other use, a number of factors have to be considered. The requirements vary over a wide range from lightly loaded, infrequently used elevators to heavily loaded industrial elevators which are constantly kept in use in handling materials. The character of the material to be handled is the first consideration, taking into account its shape, density, and its distribution on the elevator platform. The way in which material is brought to the elevator has a great bearing upon the design of the elevator and the type to be used. Consideration must be given to the amount of material to be handled in a given time and the number of floors over which it is to be distributed. From a consideration of these factors, the general type of the elevator, its capacity, its platform size, the speed at which it is to be run, and the operating system and control may be intelligently determined.

The size and shape of the elevator platform must be studied in relation to the material loaded upon it and carried by it. The nature of this material has a great bearing upon the platform size for a given load, and the density of loading in pounds for a square foot vary greatly. A large platform area may be necessary to handle bulky objects even though the weight of these objects is light. On the other hand, the material may be dense, and the loading per square foot very high, as in the case of elevators carrying castings, plate glass, and many forms of palletized material. This great variation in density of materials leads to necessity for careful consideration of the use of the elevator over a long period of time. Large elevator platforms designed for light loading may be severely overloaded and may even become a hazard if their use is changed to carry more dense materials.

One of the most important considerations in selecting or specifying an elevator is the method of loading. This also is related very closely to the general material-handling methods used in the plant in which the elevator is to be installed.

Freight-elevator loading may be classed as follows:

1. Industrial-truck loading
2. Motor-vehicle loading
3. General freight loading
4. Passenger loading

Industrial-truck Loading. A method of material handling that has become very generally used in modern industrial plants and warehouses is that of palletizing material and moving these pallets from place to place with power trucks.

Palletizing came into very general use during the Second World War, and today material is stored on pallets, warehoused, shipped, and sometimes consigned by the pallet load. In factories a form of platform frequently called a skid is often used. These skids may be of the box type and are loaded with bolts and nuts and other hardware. Palletized material is well adapted to stacking in tiers. Stock rooms are frequently arranged to store material on pallets or skids, thus greatly reducing the amount of handling required. The power trucks used to handle these pallets and skids are of the forked type and are powered, usually by storage batteries and in some cases by gasoline motors. These trucks are very heavy, frequently weighing more than the load which they handle. In the usual design of such a truck, the load is balanced by the weight of the operating machinery, storage batteries, etc. Thus when fully loaded, most of the weight of the truck and its load are supported on its two front wheels. This results in a very high concentration of loading, and when elevators are loaded by this means, the platform design must take this into consideration.

A large elevator will handle several truck or pallet loads of material, and the usual practice is to spot these palletized loads on the elevator platform by driving the truck onto the elevator, depositing the load and driving it off again. This process is repeated several times until the elevator is fully loaded. Figure 20–32 is an elevation of a freight elevator being loaded with palletized material by means of a power truck. It will be noted from the figure that, as the truck is driven onto the elevator, the front wheels are resting on the edge of the platform so that the weight of the truck and its load are concentrated at this point. Therefore, an elevator that is to be used with industrial-truck loading must be structurally strong enough to support these heavy loads at a considerable distance from the center of the platform. The arrows in Fig. 20–32 show the direction of the reactions that occur at the elevator guide shoes These reactions represent the forces with which the guide rail resists the moment induced by the load at the edge of this platform. Therefore particular attention must be given not only to the design of the elevator but also of the guide rails and guide shoes. It will also be noted from Fig. 20–32 that, just before the truck rolls onto the platform, it had to pass over the trucking sill which is part of the elevator door. Special consideration must be given to this part of the equipment also, and trucking sills for industrial-truck loading should be specified. Figure 20–33 shows a plan view of a power truck and palletized load as it is being run onto the elevator. It should be noted here that not only are the truck and its load supported on the edge of the platform but they are also considerably off center and concentrated on but one side of the car. This eccentricity of loading sets up reactions at the guide

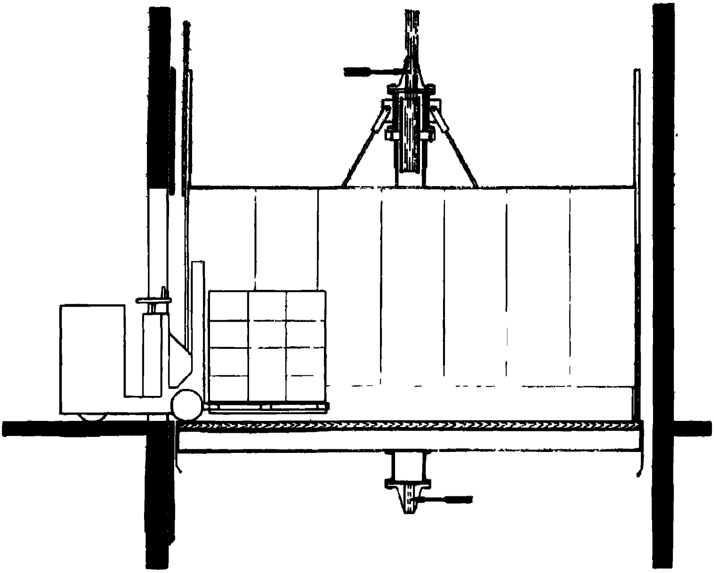


FIG. 20-32. A power truck and its palletized load entering an elevator. Most of the weight of the truck and its load are supported by the front wheels, which rest on the edge of the elevator platform. The arrows indicate the reactions of the guides.

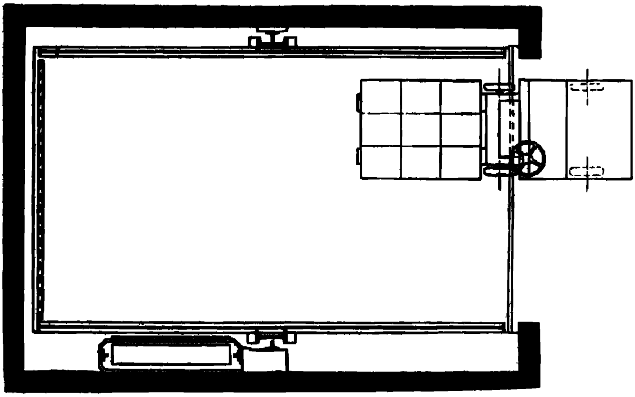


FIG. 20-33. Plan view of a power truck and palletized load entering an elevator. This shows that the load is not only concentrated at the edge of the platform but is displaced sidewise.

rails which are at right angles to those shown in Fig. 20-32 and this too must be given proper consideration.

It is not sufficient that the elevator alone be properly designed for this severe loading. It must be given consideration in the design of the building. Elevator guide rails can be no more rigid than the steel on which they are mounted, and a sufficient number of adequately designed guide-rail supports must be provided.

These modern trucks are operated at considerable speed and must be stopped by means of hand brakes while on the car. This gives rise to inertia force created by stopping the truck. This force must be absorbed by the guide rails and their supports.

Another factor which must be given consideration is the fact that in running these truckloads of material on and off the elevator, perfect accuracy in driving cannot always be expected. It is therefore, not unusual for truck drivers to strike the side of the car with a truck or the load, and elevator side guards must be designed to withstand this treatment. It is not unusual in factories and warehouses, where elevators were not originally designed for power-truck loading, to see the side guards bent and bowed from this cause. A practice which is quite generally followed and which has proved generally successful is to mount wooden rubbing strips on the sides of the car. A very commonly used form of rubbing strip is an oak plank approximately 2 in. thick, bolted to the side of the car. These rubbing strips stiffen the car sides and are readily removable when they eventually become worn.

When palletized material is loaded onto an elevator by means of power trucks, consideration should be given to the total load on the platform. Figure 20-34 shows the palletized load being deposited on the car. In this case the total load on the platform is equal to the total load to be hoisted by the car, plus the weight of the power truck. In many cases the elevator is called upon to hoist not only its load but also the truck by means of which the car is loaded. In this case the total capacity of the elevator must be equal to the weight of the net or payload plus the weight of the power truck.

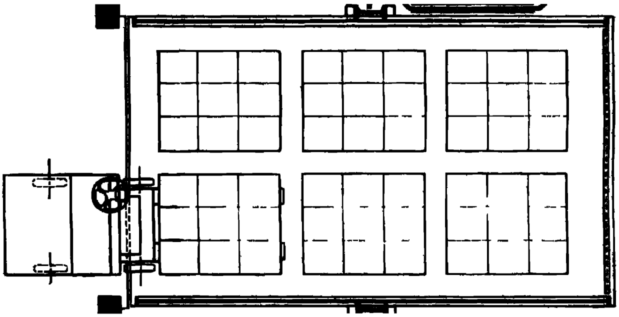


FIG. 20-34. Plan view of a loaded elevator as the last truck load is being deposited on the platform. The total load on the elevator is the weight of the load plus most of the weight of the truck.

Motor-vehicle Loading. In many large warehouses and some industrial plants, it is the practice to drive loaded automobile trucks directly onto the elevator. The truck and its load are then hoisted or lowered to a different level and driven off again. Special consideration must be given to this type of loading, but in general it is less severe than the case of industrial-truck loading. It is necessary that the platform be designed to support at least half the weight of the loaded truck on the edge of the platform, and it is necessary that the capacity of the elevator be equal to the weight of the automobile truck plus its load.

General Freight Loading. There are many uses for freight elevators having a capacity of 8,000 lb or less in which the building is not adapted to the use of power trucks, and such elevators may be designed on the basis of hand loading. In this case material is moved on and off the elevator either by hand or by means of hand trucks. The weight of these will not, in general, exceed the weight of one-quarter the capacity of the elevator. Many such elevators are used in small warehouses, department stores, and even for some applications in industrial plants where power trucks or automobile trucks are not used. In such cases general freight loading may be specified.

Passenger Loading. The passenger load in multiple-story manufacturing operations must be given serious consideration. The population density on the various floors for some kinds of manufacturing may be higher than for office buildings. In such cases, it is desirable to have a bank of passenger elevators to take care of this traffic. There is considerable interfloor traffic, consisting of supervision, time-study people, manufacturing engineers, etc. A centrally located group of passenger elevators can usually take care of this interfloor traffic most effectively. It is seldom desirable to have the freight elevators centrally located because their location depends on the manufacturing operation being performed. The freight elevators, as part of a manufacturing operation, cannot be interrupted for interfloor passenger traffic. However, it is frequently desirable to utilize freight elevators for passenger service during the morning and evening rush periods. It is often considered impractical to provide a sufficient number of passenger elevators to take care of the incoming personnel in the morning and take them out again at night.

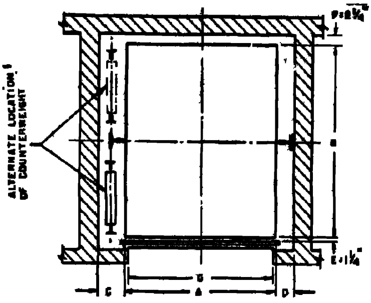
However, if the freight elevators are to be used as passenger elevators during the rush period, they should, from a safety standpoint, be considered as passenger elevators, and the rules of the American Standard Safety Code for Elevators as regards platform loading should be followed. This sometimes gives the elevator a larger capacity rating than it would have if considered purely as a freight elevator. In such cases, the elevator capacity should be selected on the basis of its passenger rating. Because of the tripping hazard, horizontal sliding doors are considered safer for passenger service than are center-parting counterbalanced doors. Counterbalanced doors are best suited for freight elevators because of their trucking sills. A method that has been used to obtain the required safety and still utilize the counterbalanced door is to interlock the hoistway doors with the car gate so that the car gate will not open until the hoistway doors are fully opened. This can be easily installed where power-operated doors are used.

It is difficult to establish any rules for selecting a freight elevator that apply in all cases, as the elevator must be considered as a part of over-all material-handling operation. A few general considerations, however, may be pointed out in connection with all heavy industrial or warehouse elevators.

1. The capacity in pounds and the size and shape of the car should be selected after a careful study of the material to be handled, taking into account its bulk, weight, and the way in which it is packaged.
2. Consideration should be given not only to the material handled by the elevator but also to the forces imposed on the car in loading and unloading if power trucks are used. If the truck is to be hoisted with the material, the rated capacity of the elevator should be made equal to the weight of the truck plus its load.
3. The walls of the car should be kept as smooth as possible by recessing the operating mechanism, such as car-switch or push-button station, and avoiding projections that may become damaged.
4. The sides of the car should be protected by rubbing strips.
5. Suitable trucking sills should be provided on hoistway doors capable of taking the severe loading imposed by power trucks.
6. Careful consideration should be given to the building structure, particularly at the points at which guide rails are supported.
7. The control and operating system should be selected on the basis of obtaining the most efficient service and the greatest flexibility. Collective operation with provision to by-pass, with an attendant service feature and an annunciator, is the most flexible system as it can be used either with or without an attendant.
8. Self-leveling is usually required to correct the stretch of the hoist ropes as heavy loads are run on and off the car. Variable-voltage control gives the best performance for large freight elevators and the most accurate self-leveling.
9. Power-operated hoistway doors and car gates should be used where the service is severe, particularly with large doors.

It is possible, where the service has been carefully studied and the conditions known, to predetermine quite accurately the amount of material that can be handled by an elevator. Certain basic data must be assembled for making this analysis. It is necessary to know the following:

1. The number of floors which the elevator is to serve.
2. The height of each floor.
3. From the above the travel of the elevator for each trip can be determined.
4. An estimate must be made of the number of stops which the elevator will make on each trip, based upon how material is to be handled or picked up.



LOAD	DIMENSIONS				
	A	B	C	D	G
2500	5'-4" 6'-4"	7'-0" 8'-0"	14¼"	8"	5'-0" 6'-0"
3000	5'-4" 6'-4"	7'-0" 8'-0"	14¼"	8"	5'-0" 6'-0"
4000	6'-4" 8'-4"	8'-0" 10'-0"	16¼"	8"	6'-0" 8'-0"
5000	8'-4" 8'-4"	10'-0" 12'-0"	16¼"	8"	8'-0"
6000	8'-4" 8'-4"	10'-0" 12'-0"	16¼"	8"	8'-0"
8000	8'-4" 8'-4"	10'-0" 12'-0"	17½"	10"	8'-0"
10000	8'-4" 10'-4"	12'-0" 14'-0"	18"	10"	8'-0" 10'-0"

FIG. 20-35. Standard elevator sizes. With front and rear openings dimensions *E*, *G*, and *H* apply at both ends

Note A: Platform loading per square foot must comply with local code requirements.

5. Select car size and capacity to handle the particular material involved. Figures 20-35, 20-36, and 20-37 show the dimensions of freight elevators that have been generally used in industry.
6. Select the car speed, based upon the height of the building.

The following is an indication of what should be used.

Table 20-24

Floors	*Speed, fpm*
3	75–100
4–5	150
6–10	200

If one of the floors exceeds 20 ft in height, the next higher speed is desirable.

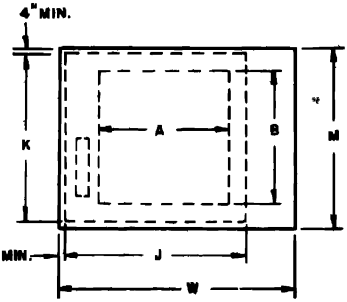
The time required to make each trip of the elevator is made up of the following:

1. *Running Time.* Running time is equal to the distance traveled divided by the car speed.
2. *Accelerating Time.* Accelerating time consists of the additional time required to accelerate and retard the car for each stop, multiplied by the number of stops made.

3. *Door Time.* Door time is equal to the time required to open and close the car-gate and hoistway doors, multiplied by the number of stops.

4. *Loading Time.* Loading time is the time required to load and unload the car and varies greatly with the type of material to be handled and the method of horizontal material handling used.

Suppose that it is desirable to know how many pounds of material per hour can be handled with an elevator having a capacity of 10,000 lb and a car speed of 150 fpm. This particular elevator has variable-voltage control with automatic self-leveling and has power-operated counter-balanced doors 8 ft high. It is to handle palletized material, and the platform size is such that it can be loaded with four pallets, each weighing 2,500 lb, or three pallets weighing 3,000 lb.



MIN MACHINE ROOM DIMENSIONS (NOTES A & B)

LOAD	RHEOSTATIC CONTROL						VARIABLE VOLTAGE CONTROL					
	RATED SPEED F.P.M.						RATED SPEED F.P.M.					
	50		75		100		100		150		200	
	M	W	M	W	M	W	M	W	M	W	M	W
2500	9'-9"	8'-11"			10'-3"	11'-0"			9'-6"	12'-0"	9'-6"	12'-0"
3000	9'-9"	10'-2"			10'-3"	11'-0"			9'-6"	12'-0"	9'-6"	12'-0"
4000			13'-9"	11'-1"	10'-3"	12'-3"			9'-6"	13'-0"	9'-6"	13'-0"
5000							13'-9"	11'-1"	13'-9"	11'-1"	15'-4"	14'-0"
6000							13'-9"	11'-1"	13'-9"	11'-1"	15'-4"	14'-0"
8000							13'-9"	11'-4"	15'-4"	14'-0"		
10000							15'-4"	14'-3"				

FIG. 20-36. Machine-room requirements.

Note A: Dimension *M* based on minimum standard length car with single entrance and regular counterbalanced doors. With deeper cars consult manufacturer.

Note B: Dimension *W* based on minimum standard width car. With wider car consult manufacturer.

It will be necessary to know from the analysis of the cycle about what the average load will be and how the material is to be distributed over the floors. Let us assume that the elevator is serving a five-floor warehouse. Let us also take the case where the material is brought up to the elevator with power trucks and that in the particular cycle considered most of the material will be taken from the third and second floors and delivered to a loading platform on the first floor and the distance between floors is 17 ft.

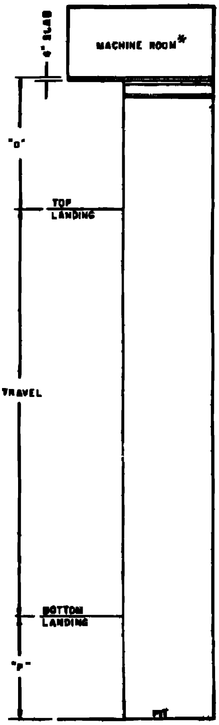
The elevator, then, will travel a distance of 34 ft in each trip from the first to the third floor and will make a stop at the second floor in the down direction only. Let us assume that the average load will be three pallets, averaging 2,800 lb in weight. Then the following calculations can be made:

Running time	= (2 × 34 × 60)/150	= 27 sec
Accelerating time	= 3 × 2½	= 7½ sec
Door time	= 3 × 9	= 27 sec
Loading time at 15 sec per pallet	= 3 × 15	= 45 sec
Total		106½ sec

Add 20 per cent, giving a round-trip time of approximately 128 sec.

Trips per hour	= 3,600/128 sec	= 28
Capacity load	= 3 × 2,800 lb	= 8,400 lb
Capacity per hour	= 28 × 8,400 lb	= 235,200 lb

In the above calculations the item that requires the greatest study of actual conditions is the loading time. It is obvious that this will vary over wide limits, depending upon the number of power trucks used, the distance from which the material must be brought up and the method of loading employed. Where there is a continuous flow of material, very satisfactory results can be obtained with two



MIN. PIT DEPTH "P" (NOTE A)

LOAD	RATED SPEED F.P.M.		
	50 TO 100	150	200
2500	4'-6"	5'-0"	5'-0"
3000	4'-6"	5'-0"	5'-0"
4000	4'-6"	5'-2"	5'-2"
5000	4'-6"	5'-2"	5'-2"
6000	4'-6"	5'-2"	5'-2"
8000	5'-0"	5'-2"	5'-6"
10000	5'-0"	5'-2"	5'-6"

MIN TOP CLEARANCE "O" (NOTE B)

LOAD	RATED SPEED FPM				
	50	75	100	150	200
2500	16'-10"		*16'-4"	*17'-4"	*17'-4"
3000	17'-7"		*17'-1"	*17'-4"	*17'-4"
4000		17'-1"	17'-1"	18'-1"	18'-1"
5000			17'-1"	17'-7"	18'-7"
6000			17'-4"	17'-7"	18'-7"
8000			17'-7"	19'-10"	
10000			19'-10"		

Fig. 20-37. Top clearance and pit requirements.

Note A: *P* based on 8 ft 0 in. door height. With 9 ft 0 in. door height, *P* = 5 ft 0 in., where table shows less than 5 ft 0 in. With 10 ft 0 in. door height *P* = 5 ft 6 in., where table shows less than 5 ft 6 in.

Note B: Dimension *O* based on 8 ft 0 in. clear car height. Increase dimension *O* by any increase above 8 ft 0 in. clear height.

Add 2 ft 0 in. to *O* dimension for car size 5 ft 4 in. by 7 ft 0 in.

Add 1 ft 0 in. to *O* dimension for car size 5 ft 4 in. by 7 ft 0 in.

* Height, 7 ft. 6 in. from floor to ceiling or trolley beam over machine.

elevators, so that one elevator is being loaded while the other is being unloaded. In this case the material that the elevator will handle can easily be made equal to the amount of material that can be brought up with the power trucks.

ELEVATOR MAINTENANCE

Elevators may be well designed and carefully selected for a particular service, but the best performance and long life can be obtained only if they are properly maintained. The objects of elevator maintenance are

1. To provide a maximum of uninterrupted service
2. To ensure long life of the equipment
3. To ensure safe operation

An elevator is an assembly of many difficult kinds of mechanism, some of which are quite intricate. Therefore, good adjustment, cleanliness, and adequate lubrication are prime essentials. Certain parts are subject to natural wear and tear and should be replaced from time to time before the failure causes an interruption to service or creates a hazard.

Good maintenance may be referred to as protective maintenance, as distinguished from mere trouble shooting, which corrects a fault only as trouble develops to a point where it causes faulty operation or a shutdown. Under protective maintenance, wearing parts, such as contacts, guide shoes, contact or shunts, and other similar parts are regularly inspected and replaced when it appears that they are approaching the end of their reliable life.

There are certain requirements that must be followed if good maintenance is to be obtained. One of the most important of these requirements is to have a maintenance staff with a thorough knowledge of the equipment. Unskilled persons may often do more harm than good by improperly diagnosing a fault and applying a wrong corrective. The symptoms of trouble should be recognized quickly and the basic source of the trouble immediately corrected. Lack of knowledge and skill leads to long shutdowns while trouble is being located and to improper steps taken in the correction of it.

To ensure a minimum of shutdowns, a regular schedule of maintenance is desirable. This schedule lists all the work that is to be done such as oiling, cleaning, inspecting, adjusting, etc. These steps should be regularly and periodically taken. Regular inspections at scheduled times should be made.

It is useful to make a log of all the troubles that occur. It is seldom that maintenance is perfect, and some trouble and occasional shutdowns may occur. If these troubles are recorded, together with their true cause, and a statement of what was done to correct it, a record will eventually be had which will be very useful as a guide for inspection and replacement.

It has been pointed out that the replacement of wearing parts should be made before they become so badly worn as to cause a shutdown or an unsafe condition. Every elevator should be equipped with a mileage recorder and a stop recorder in order that there be available a record of just how much service is being obtained. This service record, together with the trouble log, may be used as a basis for a systematic replacement of wearing parts before they become troublesome. In the replacement of parts experience with a particular installation, if systematically recorded, will furnish the best guide.

It is needless to say that, to accomplish all this, an adequate organization is necessary. Where there are a large number of elevators, a staff of maintenance men under good supervision may be employed continuously for elevator maintenance. Where there are only a few elevators, it usually becomes necessary either to maintain them by personnel who also maintain a great deal of other equipment about the plant or to have the maintenance done by others. Practically all elevator manufacturers keep a regular organization for maintaining elevators of their own manufacture. These manufacturers will enter into maintenance contracts of various types depending upon the nature of the installation. In considering the maintenance of an elevator or a group of elevators, such maintenance contracts should receive serious consideration.

SECTION 21

INDUSTRIAL PROCESS AIR CONDITIONING

BY ALFRED B. WASON

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INDUSTRIAL PROCESS AIR CONDITIONING

BY ALFRED B. WASON

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INTRODUCTION

Industrial process air conditioning conveys so many different concepts in the mind of the industrial engineer that the expression requires clarification. The industrial engineer is primarily concerned with the atmospheric conditions affecting the manufacture, preparation, process, storage, and preservation of material, equipment, and commodities. The fundamental basic factors which control these conditions are temperature, relative humidity, air circulation, and air purity, of which one or more may be the governing factor or factors. The optimum atmospheric conditions to be maintained depend upon the physical and chemical characteristic properties of the product, generally hygroscopic, and the nature of the process of manufacture.

Industrial process air conditioning may be defined as the art of simultaneously controlling the temperature, relative humidity, ventilation, cleaning, and movement of the air within an enclosure intended for the manufacture of a product sensitive to atmospheric conditions.

PSYCHROMETRY

Psychrometry is a branch of physics relating to the measurement or determination of the physical properties of atmosphere air, with particular reference to its moisture-vapor properties.

Fundamental Definitions, Laws, and Principles

To understand industrial process air conditioning, it is essential to have a general knowledge of the atmosphere surrounding the earth, which we call air, its most important physical properties, and the relations between these properties under variable conditions, and further, a practical knowledge of the instruments commonly used for measuring these properties. The composition of air is a physical mixture of vapors, gases, and dust which may be considered divided into four separate classifications, as follows:

1. The major constituents of air by volume are nitrogen, 78.1 per cent; oxygen, 21 per cent; and a small volume, 0.9 per cent, of inactive gases, principally argon. These constituents are present in the air in almost constant proportions.
2. Air includes accidental or foreign gases, such as toxic gases, odors, sulfur dioxide, ammonia, and carbon dioxide, which affect the air required for ventilation to more or less a degree in process air conditioning.
3. Air contains water vapor, moisture in gaseous state, which is the most important factor in considering atmospheric-sensitive products for manufacture.
4. Air contains solid-matter dirt, carbon particles, smoke bacteria, or any foreign particles generally termed dust which are suspended in the air.

The effects which the various components of the air have on industry are tremendous, particularly in the economics of manufacture and in personnel and healthful conditions. Especially is this true of water vapor present in the air. Fortunately, the above factors are to a large extent controllable by means of suitable equipment, properly operated and regulated to maintain the desired air conditions as required for manufacture.

Physical Properties of Air

Atmospheric air has six physical properties most applicable to process air conditioning. These are density or weight, pressure, temperature, heat content, humidity or moisture vapor, and velocity.

Density. Density is defined as the weight of a unit volume of a substance. The density of air is measured in pounds per cubic foot under specified conditions of temperature and pressure. Thus the density of air at 68°F and atmospheric pressure is 0.075 lb per cu ft.

Specific volume is the reciprocal of density. It is defined as the volume occupied by the unit weight of a substance under specified conditions of temperature and pressure and is measured in cubic feet per pound. Thus the specific volume of air at 68°F and atmospheric pressure is 13.3 cu ft per lb.

The atmospheric air totally envelops the earth, moving with it, and would remain stationary with reference to terrestrial objects except for local changes in temperature and moisture content, effecting change in density and producing winds, rain, snow, and storms. Pressure differences reflect these changes.

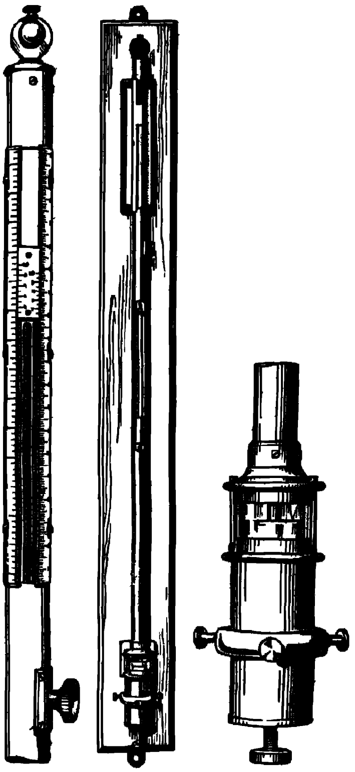


Fig. 21-1. Mercury barometer with enlarged reading scale and mercury well. Adopted by the United States Weather Bureau.

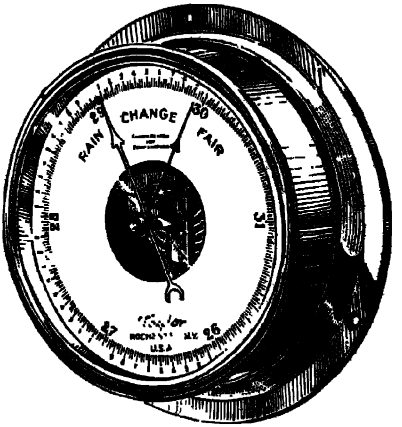


Fig. 21-2. Aneroid barometer.

Pressure. Air has appreciable density and constantly exerts an atmospheric pressure, termed "barometric pressure," which exerts a force which is equal to the weight of a column of air 1 sq in. in section through the whole layer, variously estimated at from 40 to 200 miles in depth, above which is interplanetary space. The normal or standard "barometric pressure" at sea level is 29.92 in. Hg, (slightly less at the equator), which corresponds to a pressure of 14.7 psi or, expressed another way, to the pressure of a column of water 34 ft high. As the altitude above the sea level increases, the atmospheric pressure decreases, which amounts to about ½ psi for each 1,000 ft rise for the first mile in height.

Barometer. The "barometer" is an instrument for measuring the pressure of the atmosphere. The standard instrument for this purpose is the mercury barometer which is designed to balance automatically the pressure of the atmosphere against the weight of a vertical column of mercury. A tube containing a column of mercury is inverted, and the open end is sunk in a cup containing mercury, which is exposed to the pressure of the atmosphere. The upper end of the tube is closed, and a vacuum exists between it and the column of mercury. The difference in height between the level of mercury in the cup and in the tube is the "height of the barometer"—a direct measure of atmospheric pressure (see Figs. 21-1 and 21-2).

Inches of mercury are readily convertible to pounds per square inch since 1 in. Hg is equivalent to 0.4912 psi, or conversely, 1 psi is equivalent to 2.036 in. Hg.

Gauge Pressure. Gauge pressure is the pressure measured above atmospheric pressure as a base, expressed in pounds per square inch (see Fig. 21-3).

Absolute Pressure. Absolute pressure is the sum of the gauge pressure and barometric pressure, or the total pressure above a perfect vacuum, expressed in pounds per square inch.

Vacuum. Vacuum is the pressure measured below atmosphere as a base, expressed in inches of mercury (see Fig. 21-4).

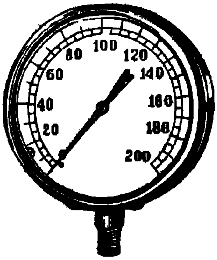


FIG. 21-3. Pressure gauge.

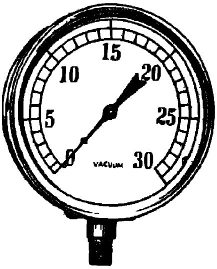


FIG. 21-4. Vacuum gauge.

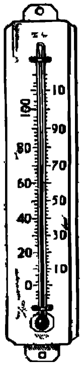


FIG. 21-5. Industrial thermometer

Temperature. Physics defines temperature as a measure of molecular activity, which is closely associated with the heat content of a substance and is measured by a conventional thermometer (see Fig. 21-5). It is generally termed the dry-bulb temperature. The temperature scales most generally used are Fahrenheit (F) and centigrade (C), the former being the most common.

$$C = \tfrac{5}{9}(F - 32) \qquad F = \tfrac{9}{5}C + 32$$

Absolute Temperature. Absolute temperature is obtained by adding 459° to the Fahrenheit temperature or 273° to the centigrade temperature.

The relationship of the three preceding physical properties of air—density, pressure, and temperature—and their performance may be further explained by the fundamental laws of perfect gases.

Law of Gases

Density of air varies with changes of temperature and pressure and performs practically in accordance with the fundamental laws of perfect gases which are

1. At a constant temperature the volume varies inversely as the absolute pressure (Boyle's law).

2. At a constant pressure the volume varies directly as the absolute temperature (Charles's law).

3. At a constant volume the pressure varies directly as the absolute temperature (Charles's law).

Combining the above laws, the general relationship becomes

$$PV = KT$$

where P = absolute pressure, psi
V = volume, cu ft/lb
K = a constant (for air 0.370)
T = absolute temperature, deg F

Density of air at atmospheric pressure, 29.92 in. Hg, at 68°F, and 50 per cent relative humidity is approximately 0.075 lb per cu ft. This figure is generally accepted for fan-rating-table performances.

Physical State of Matter. All matter exists in one or another of three different states as a solid, liquid, or gas.

Water (H_2O) can exist as a solid (ice), as a liquid (water), and as a gas (aqueous vapor or steam). The physical change in state of water with variations in temperature and pressure in combinations with air are the factors involved in psychrometry.

To illustrate, compare the relative amount of space occupied by a given weight of water in the three physical states, as shown in Table 21-1.

Table 21-1

Physical state, atmospheric pressure	Density, weight per cu ft, lb	Relative volume occupied by equivalent weight of water, cu ft
Solid (ice at 32°F)	57.5	1.09
Liquid (water at 60°F)	62.37	1.00
Gas (steam at 212°F)	0.037	1,690.0
Aqueous water vapor at 68°F and 50 per cent relative humidity	0.00054	115,500.0

Note the amazing difference in volumes between 1 cu ft of water (liquid) and the same weight of water vaporized (gas) at atmospheric conditions.

Heat. Heat is a form of physical energy existing within a substance because of molecular activity. Heat is one form of energy. There are many others, and all are mutually convertible under the laws of conservation of energy. The quantity of heat per pound of a material is generally expressed in Btu (British thermal unit). Water is taken as the standard unit of heat, and 1 Btu is defined as the heat required to raise 1 lb of water 1°F. The French thermal unit is the calorie, which is the heat required to raise 1 kg of water 1°C.

$$1 \text{ Btu} = 0.2520 \text{ calorie} \qquad 1 \text{ calorie} = 3.968 \text{ Btu}$$

Mechanical energy is measured by the foot-pound, which is the amount of work required to raise 1 lb weight 1 ft vertically against gravity. Expressed in heat equivalent, 1 Btu = 778 ft-lb. Power is the rate of doing work expressed in horsepower. One horsepower is the power to do work at the rate of 33,000 ft-lb per min. Expressed in heat equivalent, 1 hp = 42.42 Btu per min, or 2545 Btu per hr. Work and power are frequently expressed in electrical terms, the unit of work being the watthour or kilowatthour and the unit of power being the watt or kilowatt (1,000 watts).

$$1 \text{ hp} = 746 \text{ watts} \qquad 1 \text{ kw} = 1.341 \text{ hp}$$

Specific heat is defined as the number of Btu required to raise the temperature of 1 lb of a substance 1°F or the number of calories required to raise 1 kg 1°C. Water is taken as the unity of standard for specific heat, expressed 1.0. The specific heat of air at constant pressure and at ordinary temperature is 0.2375, usually figured as 0.24.

Sensible heat is the heat represented by a change in dry-bulb temperature either added to or subtracted from a substance.

Latent Heat of Fusion. To change the physical state of a substance such as ice, an appreciable amount of heat must be added. Suppose 1 lb of ice is heated under atmosphere conditions (29.92-in. barometer) from 0 to 32°F; 15.4 Btu will be required since the specific heat of ice is 0.48. Further addition of heat will cause the ice to melt, the temperature remaining constant. To melt the whole pound of ice will require 144 Btu at 32°F. This 144 Btu is called the "latent heat of fusion of ice."

Heat of the Liquid. When the pound of ice is all melted, further addition of heat will cause the temperature of the water to rise until the boiling point is reached at 212°F. The amount of heat required to raise the temperature of water from 32°F to any given temperature is called the "heat of the liquid." The heat of the liquid is determined by multiplying the temperature degree rise times the specific heat of the liquid. The heat of the liquid for water from 32 to 212°F is 180 Btu.

Latent Heat of Vaporization. Further addition of heat to the pound of water at 212°F will cause boiling, the temperature remaining constant, until all the water is evaporated. To vaporize the whole pound of water will require 970 Btu. This is defined as the "latent heat of vaporization." Water will vaporize at temperatures below the boiling point under various conditions, and the lower the temperature, the higher the latent heat of vaporization. Thus to evaporate 1 lb of water from a humidifier at 60°F requires 1058 Btu. Whenever evaporation takes place, heat must be supplied either from some source added to the air or by heating the water.

Total Heat of Steam. The total heat of steam is the amount of heat required to convert 1 lb of water at 32°F to 1 lb of steam at any given temperature and pressure. For example, total heat of dry saturated steam at 212°F is 180 + 970 = 1150 Btu per lb.

Saturated and Superheated Steam. If the process is continued and additional heat is added to the 1 lb of steam, completely evaporated, the temperature will rise, and the steam will now become "superheated," or "dry saturated." Should any particles of free moisture be carried into the space containing the steam vapor and the space thus contain all the steam vapor it can hold, the condition is called "wet saturated."

The preceding explanation of the physical change of state of water from ice (solid), water (liquid), steam, or vapor (gas) is generally understood; however, this same process takes place at atmospheric pressures at ordinary temperatures and thus explains the fundamental process of psychrometry. The air-washer type of spray humidifier is frequently referred to as a "boiler at atmospheric pressure." The process of evaporation under normal atmospheric temperatures and pressures is similar to the evaporation process of the ordinary boiler at higher temperatures and pressures within a closed space.

Total Heat. In atmospheric air, water vapor exists in varying amounts simultaneously with temperature variations. Total heat is the sum of the latent heat of the water vapor plus the sensible heat of the air mixture (air and water vapor). The wet-bulb temperature is the measure of the total heat. At a constant wet-bulb temperature the total heat remains constant irrespective of any change in temperature or relative humidity (see Table 21-3, page 1469).

Wet-bulb Temperature. Wet-bulb temperature or thermodynamic wet-bulb temperature is the temperature at which a liquid (water), by evaporating into the surrounding air, can bring the air to saturation. The wet-bulb thermometer is an ordinary dry-bulb thermometer with a clean piece of cloth or wick covering the bulb. This bulb, wet with water and with vigorous movement of air across the wick, will cause evaporation from the wetted surface of the wick, lowering the temperature below the dry-bulb temperature in direct proportion to the amount of moisture in the air. The difference in reading between the dry-bulb and wet-bulb temperatures is termed the "wet-bulb depression." This is a direct means of determining the wet-bulb temperature, a measure of the total heat of the air. The wet-bulb depression is the temperature at which the air would saturate without a change in total-heat content.

Sling Psychrometer. The sling psychrometer, the standard instrument adopted by the United States Weather Bureau, is the most reliable device for determining the dry-bulb and wet-bulb temperatures, from which readings determination of the dew-point temperature and relative humidity can readily be obtained by reference to psychrometric tables or charts. This instrument consists of a pair of standard thermometers, with one bulb covered with a silk or muslin cloth, mounted and attached to a handle. After wetting the wet-bulb wick, the instrument is swung rapidly causing the thermometers to respond quickly and accurately to the dry-bulb and wet-bulb temperatures. *Caution:* the wick must be clean. This is the most practical and accurate method in determining the psychrometric conditions of the air and is used to adjust or calibrate other atmospheric reading instruments and devices (see Figs. 21-6, 21-7, 21-8 and 21-9).

Saturated Air, Dew Point, Vapor Pressure, and Evaporation. *Saturated Air.* Water vapor has one most important physical characteristic different from all the other components of atmospheric air, *i.e.*, the *temperature* is the sole determining factor of the weight of water vapor which can exist in a given volume of space. When the given volume of space contains the limiting maximum of water vapor (expressed in grains per cubic foot), this condition is termed "saturated air." Temperature is the only factor affecting this maximum condition, and the amount of water vapor each cubic foot can hold will increase and decrease with the temperature. Thus the presence of the air in a space has no effect on the maximum amount of vapor in that space. For example, in a perfect vacuum the maximum amount of vapor per cubic foot within the given space will be determined solely by the temperature. The same condition would exist with air present at the same temperature (see Table 21-2, page 1468).

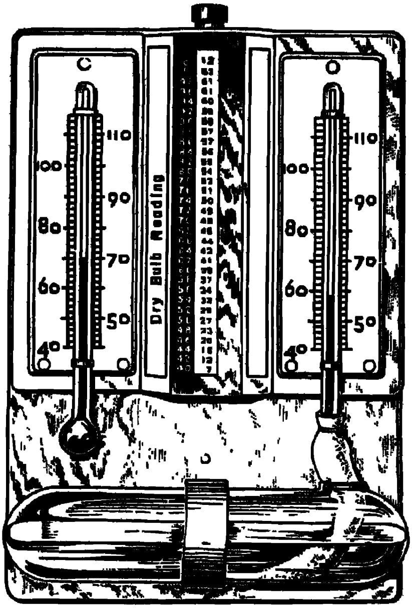


FIG. 21-6. Humidiguide—wet-bulb and dry-bulb temperatures with revolving humidity tables at the vertical center for computing the relative humidity.

Dew Point. The temperature corresponding to saturation for a given moisture content is termed the "dew-point temperature." When water vapor is present in the air to a lower extent than saturation, it is termed "superheated" since the temperature may be lowered without condensation. Should the temperature of the air be lowered to the point corresponding to the saturation point for the weight of vapor present per cubic foot, the temperature at which saturation occurs is termed the "dew point."

Vapor Pressure. The pressure exerted by the water-vapor component of the existing atmospheric pressure is termed vapor pressure and is measured in inches of mercury. Water vapor tends to saturate the surrounding space and exerts a pressure

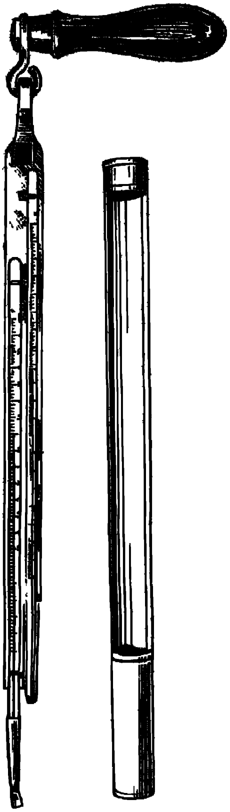


Fig. 21-7. Sling psychrometer standard type with carrying case. Adopted by the United States Weather Bureau.

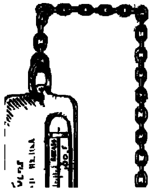


Fig. 21-8. Sling psychrometer pocket type. This sling is one-half the size of sling shown by Fig. 21-7.

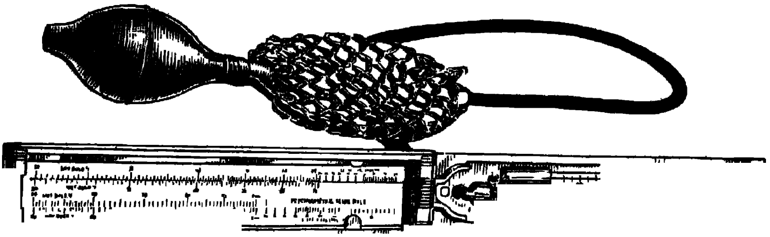


Fig. 21-9. Sling psychrometer—hand-aspirated type with psychrometric slide rule for computing the relative humidity.

which varies with the density of the vapor. At a given temperature, the vapor pressure may vary with the amount of vapor present but cannot exceed the maximum which exists when the space is saturated. Vapor pressure increases as the temperature increases.

Evaporation. The physical characteristic of evaporation of water or vaporization is due to opposing molecular forces tending to come to equilibrium. The rate of evaporation may be slow, as at ordinary temperatures, or may be rapid when water is actually boiling. The temperature is the controlling factor as to the maximum possible weight of vapor per cubic foot. However, the atmospheric pressure will affect the rate at which evaporation of water takes place. For example, in a perfect vacuum, water would instantly "flash" into vapor and diffuse within the contained space. As water vapor increases in partially saturated air, the vapor pressure increases until equilibrium between the vapor pressure and the evaporative force is established at saturation. To obtain complete saturation, the atmosphere must be confined in a closed space; otherwise vapor will diffuse rapidly to less humid surrounding areas. This is also true for areas where partial saturation is maintained within a confined space.

The factors affecting the rate of evaporation of water from a free surface, are a rise in temperature, an increase in surface area, and an increase in air velocity over the surface area. There are other factors that will increase the rate of evaporation under control of normal atmospheric conditions, such as a decrease in atmospheric pressure or a reduction in the amount of vapor present in the surrounding air.

The terms "atmospheric pressure," "barometric pressure," "vapor pressure," "water vapor," and "steam" should be clearly understood. The expressions "water vapor" and "steam" are synonymous, but customarily water vapor or aqueous vapor refers to atmospheric moisture, whereas steam refers to water vapor within a closed space, unmixed with air, more particularly at temperatures at or above the boiling point.

Humidity. Humidity refers to moisture or water vapor mixed with atmospheric air in the form of vapor.

Absolute Humidity. Absolute humidity refers to the weight of water vapor per unit volume (vapor density) expressed in grains of water vapor per cubic foot. This term, "absolute humidity," is precise as to the weight of water vapor per unit volume regardless of its temperature or whether or not the air is saturated.

Relative Humidity. Humidity as above defined refers to moisture or water vapor mixed with atmospheric air. It therefore follows that relative humidity implies the degree or amount of moisture vapor present within a given space. Then relative humidity may be defined as the ratio of the actual weight of water vapor per cubic foot (vapor density) to the maximum weight of water vapor per cubic foot at saturation at the same temperature expressed as a percentage. Thus saturated air has a relative humidity of 100 per cent. Since the laws of perfect gases apply to vapors (gases), the volume of 1 lb of aqueous vapor at a constant temperature varies inversely with the vapor pressure. The vapor pressure, therefore, is always proportional to the weight of water vapor per cubic foot (absolute humidity) at a given temperature. For every temperature at which saturation occurs (dew point), there is a corresponding value of vapor pressure. Therefore, the relative humidity may be expressed in terms of the ratio of the actual vapor pressure in the air to the vapor pressure at saturation at the same temperature, expressed as a percentage. Thus relative humidity may be expressed by a ratio of either *vapor density*, *absolute humidity*, or *vapor pressure.*

The amount of moisture present in the air is solely a function of the temperature. It therefore follows that the presence of air has nothing to do with the relative humidity in a given space. Relative humidity is frequently spoken of as "humidity," meaning per cent of saturation or degree of saturation. The tremendous effect temperature has on the capacity of air to hold moisture vapor may be illustrated by the following example: air at 35°F saturated contains 2.38 grains per lb; air at 110°F saturated contains 26.34 grains per lb, or more than eleven times as much.

Humidity Ratio. Humidity ratio is the weight of water vapor per pound of dry air, sometimes called specific humidity, expressed in grains per pound of dry air.

Humidity Percentage. Humidity percentage is the ratio of the actual weight of water vapor mixed with 1 lb of dry air to the weight of saturated water vapor mixed with 1 lb of dry air at the same temperature, expressed in percentage.

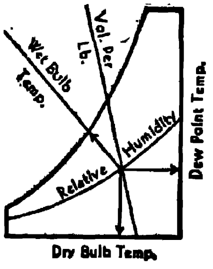
Relation of Dry-bulb, Wet-bulb, and Dew-point Temperature and Relative Humidity

The relation between the dry-bulb and wet-bulb temperature, dew point, and relative humidity are of concern to "psychrometry." Physical properties of atmospheric moisture should be thoroughly understood in order to have a clear conception of humidity. As previously explained, the wet-bulb temperature is lower than the dry-bulb temperature owing to cooling by evaporation, except at saturation, when the dew-point, dry-bulb, and wet-bulb temperatures read exactly the same. As previously stated, wet-bulb temperature is the function of the total heat of the air (air and moisture); thus when the dry-bulb and wet-bulb temperatures are known, the dew point and relative humidity corresponding to those conditions can be readily determined. It further follows that, when any two of the four factors—dry bulb, wet bulb, dew point, and relative humidity—are known, the other two are fixed and are readily determined from reference to psychrometric tables or charts. It is generally more convenient and practical to use the psychrometric chart because the graphical presentation substantially aids visualization, and degree of accuracy is ordinarily within practical application for most calculations. There are many types and arrangements of psychrometric charts in use, such as those prepared by W. H. Carrier, E. V. Hill, General Electric, Sturtevant, and Westinghouse, all of which have merit. For convenience, charts of the Sturtevant and the Westinghouse "Psychrograph" are given for reference applications (see Figs. 21-10 and 21-11).

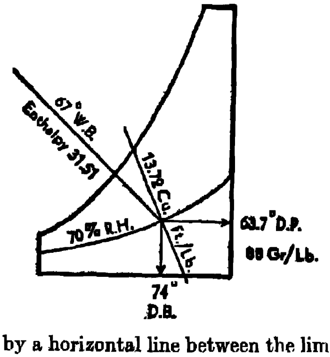
With careful study of these charts, proficiency and accuracy can readily be obtained in converting and interpreting these physical factors of psychrometry to the particular problems of process air conditioning.

Directions for Using the Sturtevant Chart (Fig. 21-10)

If any two properties of air are known, all others may be found.

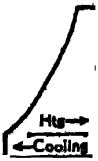


Numerical values of properties may be read directly.



Sensible heating and cooling is represented by a horizontal line between the limits of the process. All the properties of air change except the moisture content.

EXAMPLE: Air initially at 90° dry bulb is heated to 105° dry bulb.



	Initial	Final
Dry bulb	90°	105°
Wet bulb	71°	75°
Relative humidity	40 %	26.0 %
Grains/lb	84	84

	Initial	Final
Dew point	62.4	62.4
Vapor pressure	0.562	0.562
Enthalpy	34.83	38.46
Cu ft /lb	14.12	14.5

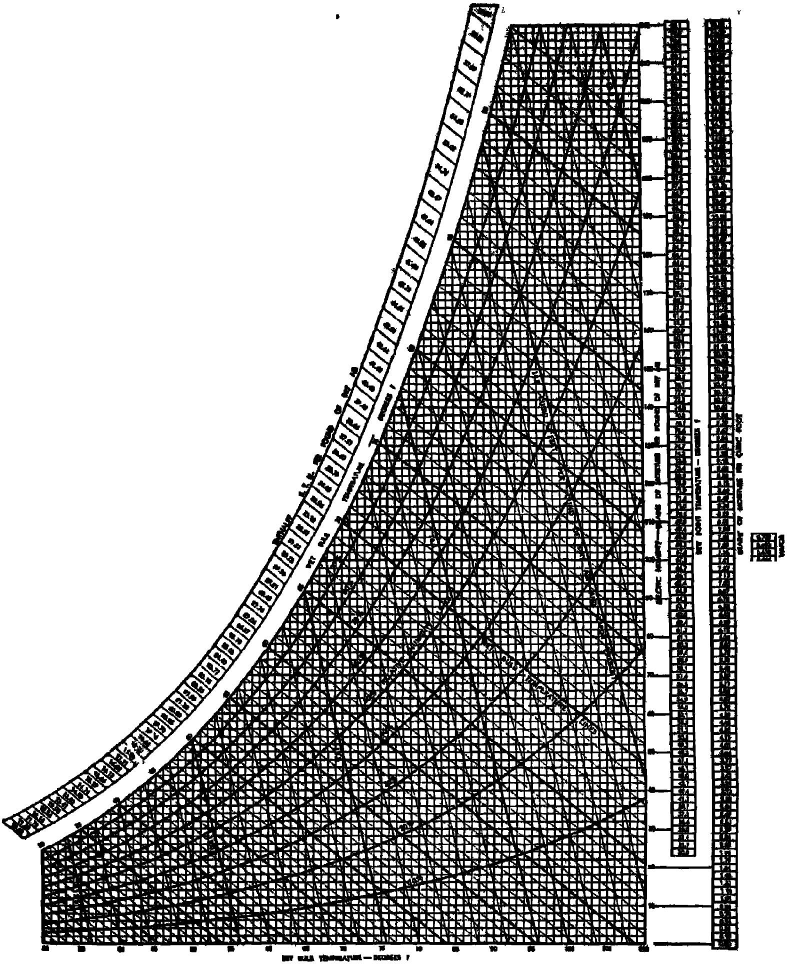
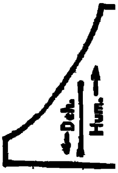


Fig. 21-10. **Sturtevant psychrometric chart.**

Humidifying and Dehumidifying with no change in dry bulb is represented by a vertical line between the limits of the process.

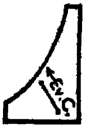
Example: Air at 90° dry bulb and 84 grains per lb is humidified without change of temperature to 128 grains per lb.



	Initial	Final		Initial	Final
Dry bulb	90°	90°	Dew point	62.4	74.3
Wet bulb	71°	78°	Vapor pressure .	0.562	0.854
Relative humidity.	40 %	60 %	Enthalpy .	34.83	41.42
Grains/lb	84	128	Cu ft/lb	14.12	14.26

Evaporating cooling is accomplished by passing air through a spray or finely divided curtain of recirculated water, and is represented by a line of constant wet-bulb temperature. Sensible heat is absorbed in evaporation, thereby increasing the moisture content while the total heat remains constant.

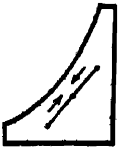
EXAMPLE: Air at 90° dry bulb and 40 per cent relative humidity is passed through a spray of recirculated water. The water temperature will approach the wet-bulb temperature of the air which remains constant.



	Initial	Final
Dry bulb	90°	72°
Wet bulb. . . .	71°	71°
Relative humidity.	40 %	95 %
Grains/lb	84	112

	Initial	Final
Dew point	62.4	70.5
Vapor pressure. . .	0.562	0.747
Enthalpy.	34.83	34.83
Cu ft/lb	14.12	13.74

Temperatures of a Mixture. Find the final wet- and dry-bulb temperatures of a mixture of 3,000 cfm at 60° dry bulb and 46° wet bulb and 5,000 cfm at 80° dry bulb and 63° wet bulb. Read from constant-volume lines on chart,



3,000 cfm	60° dry bulb	46° wet bulb	13.17 cu ft/lb
5,000 cfm	80° dry bulb	63° wet bulb	13.85 cu ft/lb

$$\frac{3{,}000}{13.17} = 228 \text{ lb/min} \qquad \frac{5{,}000}{13.85} = 361 \text{ lb/min}$$

$$228 + 361 = 589 \text{ total wt of air per min}$$

Dry-bulb temperature of the mixture

$$(228/589) \times 60 = 23.25^\circ \qquad (361/589) \times 80 = 49.10^\circ$$
$$23.25 + 49.10 = 72.35^\circ \text{ final dry bulb}$$

Wet-bulb temperature of the mixture

$$\text{Enthalpy at } 46^\circ \text{ wet bulb} = 18.12 \text{ at } 63^\circ \text{ wet bulb} = 28.48$$
$$(228/589) \times 18.12 = 7.01 \qquad (361/589) \times 28.48 = 17.45$$
$$7.01 + 17.45 = 24.46 \text{ enthalpy of mixture}$$
$$\text{Corresponding wet-bulb temperature} = 57^\circ \text{ final wet bulb}$$

Application of the Fundamental Law of Saturation. The fundamental law of saturation is expressed as "cooling by evaporation," which is cooling the air by the evaporation of moisture. This has practical application to humidifiers of all spray types when recirculating water at ordinary temperature. When passing air through a spray-type humidifier, in which no heat is added or subtracted from the air, the air will *saturate* at the *entering wet-bulb temperature*, the function of the total heat of the air. The process of saturating the air has been an exchange of heat by lowering the sensible heat by exactly the latent heat gained by the evaporation of moisture. This is an adiabatic condition of saturation at constant wet-bulb temperature. Since there is no change of total heat, the dry bulb, dew point, and relative humidity will vary until saturation occurs. The application of this fundamental law of saturation to spray-type humidifiers simplifies calculations for determining the leaving humidifier conditions (fixed dew point, moisture content of the air) for process air conditioning.

For calculations of spray-type humidifiers requiring the addition of heat or cooling by artificial source (well water or refrigerated chilled water), the above statement does not apply.

Velocity. Velocity expresses movement of air, generally flowing through a large duct, under low heads and provides the mechanical means of conveying the conditioned air, heat, and moisture, in sufficient volume, to or away from the materials in process being treated. The ultimate effectiveness and efficiency of industrial process air-conditioning operations depends upon this all-important factor—air circulation.

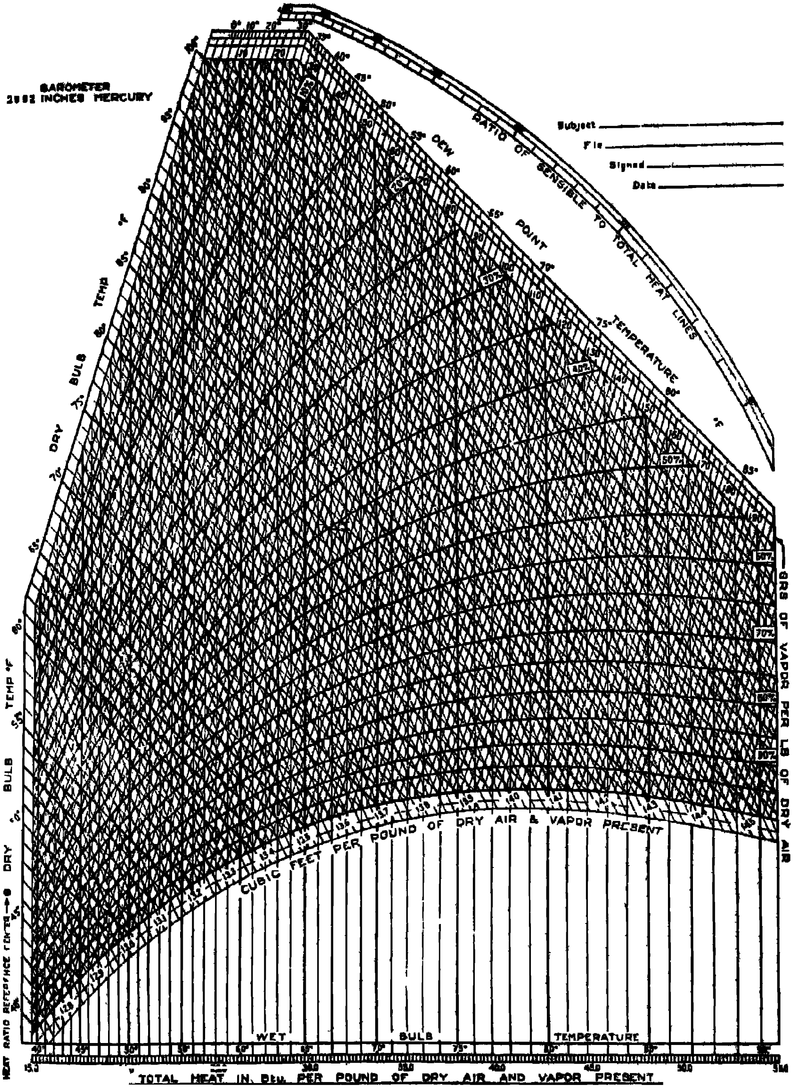


FIG 21-11. Westinghouse psychrograph

Measurements of velocity pressure and static pressures are generally made with a pitot tube to determine the velocity, expressed in feet per minute or feet per second (see Figs. 21-12 and 21-13)

Table 21-2. Properties of Air Saturated with Water Vapor

Temp, deg F	Pressure of water vapor		Weight of water vapor to saturate 1 lb dry air		Volume, cu ft		Enthalpy, Btu per lb dry air		
	In. Hg	Psi	Lb	Grains	1 lb dry air	1 lb dry with vapor to saturate	Dry air (sensible heat)	Vapor to saturate (latent heat)	Saturated mixture (total heat)
0	0.03773	0.01853	0.0007852	5.50	11.58	11.59	0.000	0.832	0.832
4	0.04645	0.02282	0.0009671	6 77	11.68	11.70	0.960	1.026	1.986
8	0.05692	0.02796	0.001186	8.30	11.78	11.80	1.920	1.260	3.180
12	0.06958	0.03418	0.001450	10.15	11.88	11.91	2.880	1.544	4.424
16	0.08469	0.04160	0.001766	12.36	11.98	12.01	3.839	1.883	5.722
20	0.1028	0.05050	0.002144	15.01	12.08	12.12	4.798	2.290	7.088
24	0.1244	0.06111	0.002596	18.17	12.18	12 23	5.758	2.778	8.536
28	0.1500	0.07368	0.003133	21.93	12.28	12.34	6.717	3.358	10.075
30	0.1645	0.08080	0.003439	24 07	12 33	12.40	7.197	3.689	10.886
32	0.1803	0.08856	0.003771	26 40	12.38	12 46	7.677	4.049	11.726
34	0.1957	0.09610	0.004094	28.66	12.43	12.51	8.157	4.399	12.556
36	0.21195	0.1041	0.004438	31.07	12.48	12.57	8.636	4.773	13.409
38	0.22925	0.1126	0.004803	33.62	12.53	12.63	9.116	5.169	14.285
40	0.24778	0.1217	0.005194	36 36	12.59	12.69	9 596	5 595	15.191
42	0.26773	0.1315	0.005616	39 31	12.64	12.75	10.08	6.05	16.13
44	0.28911	0.1420	0.006069	42.48	12.69	12.81	10.56	6.55	17.11
46	0.31191	0.1532	0.006553	45.87	12.74	12 87	11.04	7.08	18.12
48	0.33635	0.1652	0.007072	49 50	12 79	12 93	11.52	7 64	19.16
50	0 36241	0.1780	0.007626	53.38	12.84	12.99	12.00	8 25	20.25
52	0.39051	0.1918	0.008226	57.58	12.89	13.06	12.47	8.91	21 38
54	0.42003	0.2063	0.008856	61.90	12 94	13 12	12.95	9.60	22.55
56	0.45179	0.2219	0.009536	66 75	12.99	13 19	13.43	10.34	23.77
58	0.48538	0.2384	0 01026	71 82	13 04	13 25	13 91	11.14	25 05
60	0.52142	0.2561	0.01103	77.21	13.09	13 32	14 39	11.98	26.37
62	0.55970	0.2749	0.01186	83 02	13.14	13 39	14 87	12.89	27.76
64	0.60042	0.2949	0.01274	89.18	13 19	13 46	15.35	13 86	29.21
66	0.64378	0 3162	0.01368	95 76	13 24	13 53	15.83	14 90	30 73
68	0.68980	0.3388	0.01468	102 8	13.29	13.60	16.31	16 00	32 31
70	0.73866	0.3628	0.01574	110.2	13 34	13 68	16.79	17.17	33.96
72	0.79058	0.3883	0.01688	118 2	13.40	13 75	17 27	18.43	35.70
74	0.84555	0.4153	0.01809	126 6	13 44	13 83	17 75	19.76	37.51
76	0.90398	0.4440	0.01938	135 7	13.49	13 91	18.23	21.19	39.42
78	0.96588	0.4744	0.02075	145 3	13 54	13 99	18.71	22.71	41.42
80	1.0316	0.5067	0.02221	155.5	13.59	14.08	19 19	24 32	43.51
82	1.1013	0.5409	0.02377	166.4	13.64	14 16	19.67	26.05	45.72
84	1.1752	0.5772	0.02543	178 0	13.69	14 26	20.15	27.90	48.05
86	1.2527	0.6153	0.02718	190 3	13.74	14 34	20.63	29.84	50 47
88	1.3346	0.6555	0.02904	203 3	13 79	14 44	21.11	31.91	53.02
90	1.4211	0.6980	0.03102	217.1	13 84	14 53	21 59	34 11	55 70
92	1.5125	0 7429	0.03312	231.8	13 89	14.63	22.07	36.45	58.52
94	1.6088	0.7902	0.03535	247.5	13 94	14.73	22.56	38.94	61 50
96	1.7108	0.8403	0.03772	264.0	13 99	14.84	23.04	41.58	64 62
98	1.8181	0.8930	0.04024	281 7	14 04	14 95	23.52	44.40	67 92
100	1.9316	0.9487	0.04293	300.5	14.10	15.07	24.00	47.40	71.40
105	2.2414	1.1009	0 05037	352 6	14 22	15.37	25 20	55.73	80 93
110	2.5939	1.274	0.05904	413 3	14 35	15.71	26.40	65.46	91 86
115	2.9929	1.470	0.06913	483.9	14.47	16.08	27.60	76.80	104.40
120	3.4449	1.692	0.08093	566.5	14.60	16.50	28.80	90.09	118.89
125	3.9519	1.941	0.09466	662.6	14.72	16.96	30.00	105.59	135.59
130	4.5220	2.221	0 1107	774.9	14.85	17.49	31 20	123.73	154.93
135	5.1633	2 536	0.1293	907.9	14.98	18.10	32.41	145.26	177.67
140	5.8779	2.887	0.1521	1,064.7	15.10	18.79	33.61	170.69	204.40
145	6.6781	3.280	0.1787	1,250.9	15.23	19.60	34.81	200.95	235.76
150	7.5658	3.716	0.2105	1,473.5	15.35	20.55	36.02	237 17	273.19
160	9.6486	4.739	0.2961	2,072.7	15.61	23.03	38.43	334.95	373 38
170	12.196	5.990	0.4280	2,996.0	15.86	26.77	40.83	486.08	526.91
180	15.290	7.510	0.6501	4,550.7	16 11	32.94	43.24	741.24	784.48
190	19.008	9.336	1 083	7,581.0	16 36	44 85	45.66	1,239.71	1,285.37
200	23.465	11.525	2 261	15,827 0	16 61	76 99	48.07	2,598.34	2,646.41

Table 21-3. Total Heats
[Enthalpy (Btu per pound of dry air)]

Wet-bulb temp, deg F	Tenths of a degree									
	0	1	2	3	4	5	6	7	8	9
40	15.19	15.24	15.28	15.33	15.38	15.43	15.47	15.52	15.57	15.61
41	15.66	15.71	15.75	15.80	15.85	15.90	15.94	15.99	16.04	16.08
42	16.13	16.18	16.23	16.28	16.33	16.38	16.42	16.47	16.52	16.57
43	16.62	16.67	16.72	16.77	16.82	16.87	16.91	16.96	17.01	17.06
44	17.11	17.16	17.21	17.26	17.31	17.36	17.41	17.46	17.51	17.56
45	17.61	17.66	17.71	17.76	17.81	17.87	17.92	17.97	18.02	18.07
46	18.12	18.17	18.22	18.28	18.33	18.38	18.43	18.48	18.54	18.59
47	18.64	18.69	18.74	18.80	18.85	18.90	18.95	19.00	19.05	19.11
48	19.16	19.21	19.27	19.32	19.38	19.43	19.48	19.54	19.59	19.65
49	19.70	19.76	19.81	19.87	19.92	19.98	20.03	20.09	20.14	20.20
50	20.25	20.31	20.36	20.42	20.47	20.53	20.58	20.64	20.69	20.75
51	20.80	20.86	20.92	20.97	21.03	21.09	21.15	21.21	21.26	21.32
52	21.38	21.44	21.49	21.55	21.61	21.67	21.72	21.78	21.84	21.89
53	21.95	22.01	22.07	22.13	22.19	22.25	22.31	22.37	22.43	22.49
54	22.55	22.61	22.67	22.73	22.79	22.85	22.91	22.97	23.03	23.09
55	23.15	23.21	23.27	23.34	23.40	23.46	23.52	23.58	23.65	23.71
56	23.77	23.83	23.90	23.96	24.02	24.09	24.15	24.21	24.27	24.34
57	24.40	24.47	24.53	24.60	24.66	24.73	24.79	24.86	24.92	24.99
58	25.05	25.12	25.18	25.25	25.31	25.38	25.44	25.51	25.57	25.64
59	25.70	25.77	25.83	25.90	25.97	26.04	26.10	26.17	26.24	26.30
60	26.37	26.44	26.51	26.58	26.65	26.72	26.78	26.85	26.92	26.99
61	27.06	27.13	27.20	27.27	27.34	27.41	27.48	27.55	27.62	27.69
62	27.76	27.83	27.90	27.98	28.05	28.12	28.19	28.26	28.34	28.41
63	28.48	28.55	28.63	28.70	28.77	28.85	28.92	28.99	29.06	29.14
64	29.21	29.29	29.36	29.44	29.51	29.59	29.66	29.74	29.81	29.89
65	29.96	30.04	30.11	30.19	30.27	30.35	30.42	30.50	30.58	30.65
66	30.73	30.81	30.89	30.96	31.04	31.12	31.20	31.28	31.35	31.43
67	31.51	31.59	31.67	31.75	31.83	31.91	31.99	32.07	32.15	32.23
68	32.31	32.39	32.47	32.55	32.63	32.72	32.80	32.88	32.96	33.04
69	33.12	33.20	33.29	33.37	33.46	33.54	33.62	33.71	33.79	33.88
70	33.96	34.05	34.13	34.22	34.31	34.40	34.48	34.57	34.66	34.74
71	34.83	34.92	35.00	35.09	35.18	35.27	35.35	35.44	35.53	35.61
72	35.70	35.79	35.88	35.97	36.06	36.15	36.24	36.33	36.42	36.51
73	36.60	36.69	36.78	36.87	36.96	37.06	37.15	37.24	37.33	37.42
74	37.51	37.61	37.70	37.80	37.89	37.99	38.08	38.18	38.27	38.37
75	38.46	38.56	38.65	38.75	38.84	38.94	39.04	39.13	39.23	39.32
76	39.42	39.52	39.62	39.71	39.81	39.91	40.01	40.11	40.20	40.30
77	40.40	40.50	40.60	40.71	40.81	40.91	41.01	41.11	41.22	41.32
78	41.42	41.52	41.63	41.73	41.84	41.94	42.04	42.15	42.25	42.36
79	42.46	42.57	42.67	42.78	42.88	42.99	43.09	43.20	43.30	43.41
80	43.51	43.62	43.73	43.84	43.95	44.06	44.17	44.28	44.39	44.50
81	44.61	44.72	44.83	44.94	45.05	45.16	45.28	45.39	45.50	45.61
82	45.72	45.84	45.95	46.07	46.18	46.30	46.42	46.53	46.65	46.76
83	46.88	47.00	47.12	47.23	47.35	47.47	47.58	47.70	47.82	47.93
84	48.05	48.17	48.29	48.41	48.53	48.65	48.76	48.88	49.00	49.12
85	49.24	49.36	49.49	49.61	49.73	49.86	49.98	50.10	50.22	50.35
86	50.47	50.60	50.72	50.85	50.98	51.11	51.23	51.36	51.49	51.61
87	51.74	51.87	52.00	52.11	52.25	52.38	52.51	52.64	52.76	52.89
88	53.02	53.15	53.29	53.42	53.55	53.69	53.82	53.95	54.08	54.22
89	54.35	54.49	54.62	54.76	54.89	55.03	55.16	55.30	55.43	55.57
90	55.70	55.84	55.98	56.12	56.26	56.40	56.53	56.67	56.81	56.96
91	57.09	57.23	57.38	57.52	57.66	57.81	57.95	58.09	58.23	58.38
92	58.52	58.67	58.81	58.96	59.11	59.26	59.40	59.55	59.70	59.84
93	59.99	60.14	60.29	60.44	60.59	60.75	60.90	61.05	61.20	61.35
94	61.50	61.66	61.81	61.97	62.12	62.28	62.43	62.59	62.74	62.90
95	63.05	63.21	63.36	63.52	63.68	63.84	63.99	64.15	64.31	64.46
96	64.62	64.78	64.95	65.11	65.27	65.44	65.60	65.76	65.92	66.09
97	66.25	66.42	66.58	66.75	66.92	67.09	67.25	67.42	67.59	67.75
98	67.92	68.09	68.26	68.43	68.60	68.78	68.95	69.12	69.29	69.46
99	69.63	69.81	69.98	70.16	70.34	70.52	70.69	70.87	71.05	71.22
100	71.40	71.58	71.76	71.94	72.12	72.31	72.49	72.67	72.85	73.03
101	73.21	73.40	73.58	73.77	73.95	74.14	74.32	74.50	74.69	74.88
102	75.06	75.25	75.44	75.63	75.82	76.02	76.21	76.40	76.59	76.78
103	76.97	77.17	77.36	77.56	77.75	77.95	78.14	78.34	78.53	78.73
104	78.92	79.12	79.32	79.52	79.72	79.93	80.13	80.33	80.53	80.73
105	80.93	81.14	81.34	81.55	81.76	81.96	82.17	82.38	82.59	82.79

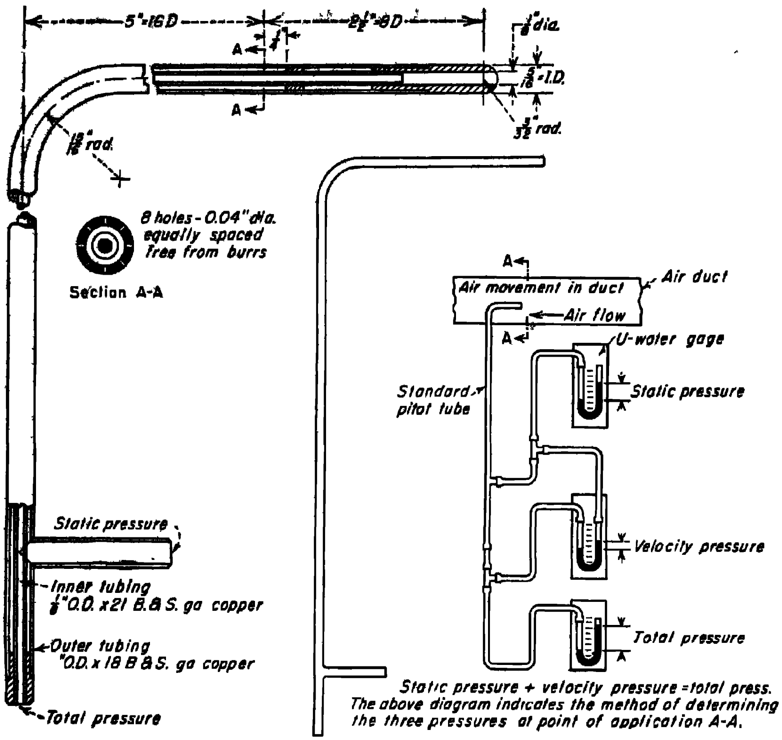

5"=16D
2½"=8D
¼"
A
A
⅛" dia.
5/16"=I.D.
3/32" rad.
13/16" rad.
8 holes - 0.04" dia. equally spaced free from burrs
Section A-A
A
Air movement in duct
Air duct
Air flow
A
U-water gage
Static pressure
Standard pitot tube
Velocity pressure
Total pressure
Static pressure
Inner tubing ⅛" O.D. x 21 B.&S. ga copper
Outer tubing 5/16" O.D. x 18 B&S. ga copper
Total pressure
Static pressure + velocity pressure = total press.
The above diagram indicates the method of determining the three pressures at point of application A-A.

FIG. 21-12. Standard pitot tube for measurement of air movement in confined spaces.

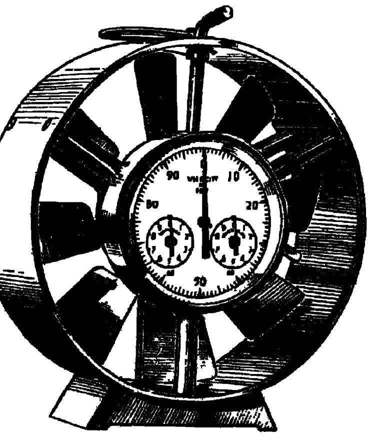


FIG. 21-13. Anemometer for measurement of air movement in open spaces.

PROPERTY CHARACTERISTICS OF PROCESS MATERIALS

Industrial process air conditioning covers a broad field and is frequently confused with the *process drying of materials*. The distinction must be clearly defined so that the manufacturer handling atmospheric-sensitive products will understand the fundamental principles involved between process air conditioning and process drying of materials, thus enabling him to classify the process procedure adequately. Briefly, the distinction between these process classifications may be stated as follows: process air conditioning means controlled humidity of the air *controlling the moisture content* within the product, whereas process drying means *removal of water content* by evaporation within the product. The discussion in this text will be limited to process air conditioning.

Physical Combination of Water with Solids. Water associates with solids in various combinations, *e.g.*, free moisture on the surface of a substance, water held within the pores or interstices of a solid, particles suspended in the water, and crystalline formation, or a substance which absorbs or retains moisture. The last is known as the hygroscopic characteristic of a substance. Water combinations with materials may be in one or more of these physical states. The last, hygroscopic material, is the most common combination met in process air conditioning. Hygroscopic materials are those which readily absorb and retain moisture and at a fixed temperature have the quality to absorb a fixed percentage of their own weight of moisture from the surrounding atmosphere containing a fixed amount of water vapor.

Process Classification. Products for manufacture may be classified as chemical, crystalline, metallic, fibrous, or hygroscopic materials. Process air conditioning may be broken down into control of rate of chemical reactions, control of rate of crystallization, control of corrosion of metals, moisture control of fibrous or hygroscopic material, and elimination of static electricity.

Control of Chemical Reactions. The control of chemical reactions is primarily the function of controlling the temperature, thus regulating the rate of the chemical reactions, although some chemical products require control of both the temperature and humidity. For example, rayon manufacture would be a typical illustration in the mercerizing process where temperature controls the rate of reaction and relative humidity controls the rate of evaporation. The dough room in the baking industry is a typical example of biochemical reaction. The temperature controls the rate of fermentation of yeast, and the relative humidity prevents crusting of the surface of the dough, allowing carbon dioxide gases formed by fermentation to pass freely through the surface of the dough.

Control of Crystallization. Crystallization is the process of forming crystals by controlling the rate of cooling of a saturated solution. The temperature controls the size, whereas the relative humidity controls the rate of evaporation, regulating the density of the solution. The coating of pills is a typical example. A heavy solution of sugar is added to the tumbler, and as the water evaporates, crystals of sugar form on the surface of the pills. Close control of temperature produces a fine, clear, smooth coating on the pills, and the relative humidity regulates the rate of evaporation. This process is frequently referred to as control of the dry-bulb and wet-bulb temperatures, since the product temperature approaches the surrounding wet-bulb temperature owing to evaporation.

Control of Corrosion of Metals. Manufacture of precision instruments requires close tolerance limitations, protection against corrosion, uniformity, elimination of dust and injurious gases, and improved employee efficiency. The manufacture of gauges is a typical example where accurate control of temperature assures uniformity of tolerance standards, tooling, assembly precision, calibration, reduction of scrap; clean air avoids spoilage or rejects from dirt particles; removal of gases harmful to health and low relative humidity reduce corrosion caused by perspiration from worker's hands; comfortable conditions increase employee accuracy of workmanship and personnel efficiency.

Control of Moisture Content of a Fibrous or Hygroscopic Material. Moisture content of a hygroscopic material is a general term and refers to either free-moisture content (wet cloth) or hygroscopic moisture (regain) which varies with atmospheric conditions. To clarify this distinction between moisture content and moisture regain, suppose, for example, a piece of cloth is soaking wet. The moisture content may be 75 per cent of the total weight. If the cloth is hung up to dry, the free moisture will run off or evaporate, and the cloth eventually will be dry with no apparent surface moisture. However, there is still hygroscopic moisture present in the cloth. The amount will be proportional to the prevailing conditions of the surrounding atmosphere. If this cloth is placed in a heated dry atmosphere, it will continue to lose weight until an equilibrium point is reached. No further loss in weight will occur. This final constant weight is the "dry weight" frequently referred to as "the bone-dry weight." Then the moisture content is the sum of free moisture and hygroscopic

Table 21-4. Regain of Hygroscopic Materials*

(Moisture content expressed in per cent of dry weight of the substance at various relative humidities—temperature, 75°F)

Classification	Material	Description	Relative humidity, per cent									Authority
			10	20	30	40	50	60	70	80	90	
Natural textile fibers	Cotton	Sea island, roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1	Hartshorne
	Cotton	American, cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Australian merino, skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4	Hartshorne
	Silk	Raw chevennes, skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.8	Schloesing
	Linen	Tablecloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Linen	Dry spun, yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8	Sommer
	Jute	Average of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2	Storch
	Hemp	Manila and sisal, rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa
Rayons.......	Viscose nitrocellulose cupramonium	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Cellulose acetate	Fiber	0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3	Robertson
	Nylon	Fiber	1.1	1.4	1.7	2.3	2.8	3.4	4.1	5.1	5.7	du Pont
Paper.........	M. F. newsprint	Wood pulp, 24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6	U. S. B. of S.
	H. M. F. writing	Wood pulp, 3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2	U. S. B. of S.
	White bond	Rag, 1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U. S. B. of S.
	Com. ledger	75% rag, 1% ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	U. S. B. of S.
	Kraft wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U. S. B. of S.
Miscellaneous organic materials	Leather	Sole oak, tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2	Phelps
	Catgut	Racquet strings	4.6	7.2	8.6	10.2	12.0	14.3	17.3	19.8	21.7	Fuwa
	Glue	Hide	3.4	4.8	5.8	6.6	7.6	9.0	10.7	11.8	12.5	Fuwa
	Rubber	Solid tire	0.11	0.21	0.32	0.44	0.54	0.66	0.76	0.88	0.99	Fuwa
	Wood	Timber (average)	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0	Forest P. Lab.
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8	Fuwa
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0	Ford
Foodstuffs....	White bread	....	0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0	Atkinson
	Crackers	....	2.1	2.8	3.3	3.9	5.0	6.5	8.3	10.9	14.9	Atkinson
	Macaroni	....	5.1	7.4	8.8	10.2	11.7	13.7	16.2	19.0	22.1	Atkinson
	Flour	....	2.6	4.1	5.3	6.5	8.0	9.9	12.4	15.4	19.1	Bailey
	Starch	....	2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7	Atkinson
	Gelatin	....	0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4	Atkinson
Miscellaneous inorganic materials	Asbestos fiber	Finely divided	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84	Fuwa
	Silica gel	....	5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6	Fuwa
	Domestic coke	....	0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89	Selvig
	Activated charcoal	Steam activated	7.1	14.3	22.8	26.2	28.3	29.2	30.0	31.1	32.7	Fuwa
	Sulfuric acid	H_2SO_4	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason

* Courtesy of the American Society of Heating and Ventilating Engineers.

Table 21-5. Temperatures and Relative Humidities Applicable to Industrial Process Air Conditioning

Industry and process	Dry-bulb temp, deg F	Relative humidity, per cent
Food products		
Baking		
Fermentation, dough room	80	76–80
Proofing, proof box	90–95	85–90
Loaf cooling, paraffin wrapping paper	70–90	60–70
Pie cooler, pie crust	70–90	60–70
Cake icing	70	50
Storage, flour	65–75	55–65
Storage, yeast	32–45	60–75
Breweries		
Fermentation, vat room	44–50	50–80
Storage of grains, prevent sprouts	60	30–45
Confectionery		
Hard candy	75–80	45–50
Chocolate candy		
Dipping	66	45–50
Molding	60	58
Cooling tunnel	35–45	80
Wrapping	65–68	46–50
Enrobing	61–67	40–50
Packing	65–68	46–50
Storage, candy	68–70	40–50
Storage, starch	75–85	45–50
Cocoa	45–65	45–50
Caramels	70	50
Chewing gum, chicle	66–70	45–50
Marshmallows	80	57
Lifesavers, cutting, wrapping	72	35
Cereals		
Preparation	60–70	38
Drugs		
Coffee substitutes, packing and storage	75	35
Headache powders	80	40
Pills, tablets	70	20–30
Packaging tablets	80	40
Storage of powders, tablets	70–80	30–40
Fruits		
Storage of citrus fruits		
Lemons	58	84–88
Grapefruit	60	86–88
Oranges	58	88–90
Bananas		
Storage	62–65	50
Ripening	70–76	90
Meats		
Ripening of sausage	40	80
Miscellaneous food products		
Flour mill	80–85	60–65
Gelatin	70–80	65
Nuts	30–32	75–80
Ceramic products		
Grinding wheels	75	40
Molding powders	80	30–60
Chemical products		
Explosives		
Powder mixing	80	60
Fuse loading	70	55
Cellophane	70	5
Matches		
Manufacturing	68–72	50–55
Storage	60	30–40

Table 21-5. Temperatures and Relative Humidities Applicable to Industrial Process Air Conditioning (*Cont*)

Industry and process	Dry bulb temp, deg F	Relative humidity, per cent
Chemical products (*Cont*)		
Rubber		
Manufacturing	75–80	25–30
Cementing	80	25–30
Varnish drying		
Drying of lacquers	70–90	25–50
Drying of paints	60–90	25–50
Hygienic tubes	45	80
Electrical products		
Toll-cable cooling and storage	80–100	5–10
Insulation winding	120	20
Transformer testing	80	60–80
Fur products		
Storage	28–40	50–65
Leather products		
Shoes, milling	90–110	65–90
Storage	50–60	40–60
Metal products		
Gold beating	86	70
Needles	80	20–30
Ball bearings	80	30–40
Precision instruments	80	20–40
Photographic products		
Film drying	70–75	60–70
Storage	65–70	55–65
Printing products		
Color printing	72–75	50–55
Gravure		
Sensitizing	68–75	45–50
Engraving	68–70	45
Etching	68–70	45
Layout and transfer	68–70	45
Color gravure	68–75	45
Drying room	75	50
Retouching	75	38
Pressroom	75–80	50–60
Lithographing		
Pressroom	75–80	50–60
Binding	70–80	45–60
Storage	75–80	45–60
Storage of rollers	70–90	35–50
Paper products		
Sandpaper	90–100	40
Sealed containers	70	60
Textile products		
Cotton		
Carding	80–95	50–55
Combing	80–95	50–60
Roving	80–95	50–60
Spinning	80–90	55–65
Weaving	80–85	85
Linen	80–90	65
Jute	80–90	65
Hemp	80–90	65
Rayon manufacture		
Viscose process		
Spinning	75–85	80–85
Textile departments	72–80	70–80
Acetate process		
Spinning	75–85	75–85
Textile departments	72–80	70–80
Cuprammonium process		
Textile departments	75–80	70–80

Table 21-5. Temperatures and Relative Humidities Applicable to Industrial Process Air Conditioning *(Cont.)*

Industry and process	Dry-bulb temp deg F	Relative humidity, per cent
Textile products *(Cont.)*		
Silk		
Spinning	80-90	55-60
Throwing	80-90	55-60
Weaving	80-90	60-65
Wool		
Carding	80-95	50-55
Preparatory	80-90	55-60
Spinning	80-90	55-60
Weaving	80-90	60-65
Miscellaneous textiles		
Rayon dress goods		
Preparation	80-90	55
Twisting	80-90	55-60
Weaving	80-90	60-65
Inspection, cloth room	80-95	50
Full-fashioned hosiery, women's		
Manufacture machine room		
Silk	80-90	60
Nylon	75-80	50-55
Tire fabrics	80-90	60-65
Draperies rayon	80-90	60-65
Embroidering spooling and weaving	80-90	60-65
Knitting	80-90	50-60
Aralac milk fiber	75-85	80
Tobacco products		
Cigars		
Sweat room	90	85
Bin and preparation room	70-80	70
Stemming and stripping	75-85	70
Binder and wrapping	70-85	70
Filler	70-85	70
Machine room	70-85	70
Banding and packing	75-85	65
Storage	68-75	60-80
Cigarettes		
Sweat room	90	85
Preparatory storage	75-85	65
Machine room	75-85	58-60
Packing	75-85	58-60
Storage	75-85	58-60
Miscellaneous products		
Hats		
Forming	75-90	65-75
Blowing	75-90	65-70
Sizing	75-90	70-75
Safety glass	90	30
Optical lenses	80-90	50-60
Lenses fusing	75	45
Feathers	70-75	46-55
Linoleum	80	40

moisture contained in a material, expressed as a percentage of the *gross* weight of the material The regain is the hygroscopic moisture only contained in a material, expressed as a percentage of the bone-dry weight of the material The regain of a hygroscopic material will depend upon the physical characteristics of that material, the temperature, and especially upon the relative humidity of the surrounding atmosphere The temperature and relative humidity control the hygroscopic moisture content of materials and correspondingly affect the physical properties of a material—weight, strength, pliability, flexibility, softness, stretch, quality, appearance, and facility of process of manufacture Different materials contain various percentages of moisture after continued exposure to atmosphere Likewise, the rate of absorption or loss of moisture varies with the thickness and density of the material. The fact that hygroscopic materials come into equilibrium with the surrounding air is the fundamental basis for the control of the physical properties during the process of manufacture (see Table 21-4).

Classification of Industry. Industrial process air conditioning concerns the atmospheric conditions required for the process and preservation of materials for manufacture. Requirements within the range of human comfort may be maintained within the manufacturing areas, but when atmospheric conditions are not within the range of human comfort, special consideration must be made, within a closed space, for maintenance of required conditions. The most favorable temperature and relative-humidity requirements may range as low as 5 per cent relative humidity in certain applications to a condition approximating saturation in other applications. Variations of maintained atmospheric conditions will often occur within the same industry, depending upon the process of manufacture, and frequently a compromise is required where the employee's health and comfort are concerned. The ideal atmospheric conditions have been established for many industrial applications, but when the requirements are unknown, special laboratory tests will be required to determine the optimum atmospheric conditions (see Table 21-5).

LOAD SURVEY AND CALCULATION OF REQUIREMENTS

The standard of industrial process air conditioning, as previously defined, is simultaneous control of temperature, relative humidity, ventilation, air purity, and air movement. The design of an air-conditioning system for any treated space must conform to these standards. Before designing a system for process air conditioning, it is necessary to organize and assemble all available information and data. This information will include geographical location, building plans, outside atmospheric conditions, inside temperatures and relative humidity desired for summer and winter, layout of machinery, internal heat loads, number of occupants, lighting, machinery horsepower, and all other heat-producing factors, as well as a study of the removal of obnoxious fumes or gases for proper ventilation, elimination of dust, moisture gain or loss within the treated area, air distribution, an analysis of the economic situation, and any other factors affecting the design. For convenience refer to Figs. 21-14, 21-15, and 21-16, which give a summary of information desired before preparing calculations. This format was primarily designed for an existing building but is equally applicable to assembling data for a new building project. Since the entire subsequent estimate is based on the data shown on the survey sheet, it follows that a careful study and analysis must be made before an accurate engineering determination can be calculated to meet load requirements.

CALCULATION OF LOAD REQUIREMENTS

The design objective is to determine the capacity of the air-distributing equipment to balance the performance of the heat- and moisture-producing factors. The summer load requirement is most frequently the controlling load-determining factor, since winter requirements are invariably smaller. Determination of the summer load calculations are somewhat involved owing to the variable contributing load factors, the maximum effect of which may not occur simultaneously. Considerable judgment must be used to determine the various load relationships, *e.g.*, the heat-transmission difference between day and night, fluctuations in solar-heat load, lighting loads within the building, monitor construction or saw-tooth type, intermittent machinery operation for design heat-load requirements, and load location within the conditioned space. The air-conditioning system must be so designed that variable load factors will not affect maintenance of desired atmospheric conditions.

The summer internal heat load is determined by calculating the heat load produced from five major factors:

1. Heat transmission through the building structure
2. Solar-heat transmission
3. Heat produced by occupants
4. Heat produced from the lights
5. Heat produced from motors, steam, and all other heat-producing sources

Table 21-6. Design Dry- and Wet-bulb Temperatures

(Wind velocities and wind direction for June, July, August, and September; also average number of days per year that outside temperatures exceed design temperatures; temperatures, deg F)

State	City	Design temp, deg F		Max temp, deg F		No. days per year exceeds design		Summer wind velocity, mph	Prevailing summer wind direction	Altitude, ft
		Dry bulb	Wet bulb	Dry bulb, normal year	Wet bulb, normal year	Dry bulb	Wet bulb			
Ala.	Birmingham	95	78	101	81	1	8	5.2	S	594
	Mobile	95	80	106	81	1	4	8.6	SW	10
Ariz.	Phoenix	105	76	110	78	2	2	6.0	W	1,108
Ark.	Little Rock	95	78	103	81	1	10	7.0	NE	324
Calif.	Los Angeles	90	70	109	75	6	8	6.0	SW	261
	San Francisco	90	65	101	66	0	2	11.0	SW	50
Colo.	Denver	95	64	99	67	1	5	6.8	S	5,221
Conn.	New Haven	95	75	101	81	2	6	7.3	S	23
Del.	Wilmington	95	78					9.7	SW	25
D.C.	Washington	95	78	102	81	3	6	6.2	S	72
Fla.	Jacksonville	95	78	99	81	2	7	8.7	SW	18
	Tampa	94	79	98	81	2	5	7.0	E	25
Ga.	Atlanta	95	76	101	81	2	7	7.3	NW	975
	Savannah	95	78	105				7.8	SW	42
Idaho	Boise	95	65	109	71	4	6	5.8	NW	2,705
Ill.	Chicago	95	75	104	80	3	5	10.2	NE	594
	Peoria	95	76	103	81	4	4	8.2	S	602
Ind	Indianapolis	95	76	101	81	2	10	9.0	SW	715
Iowa	Des Moines	95	77					6.6	SW	800
Kans.	Wichita	100	75	110	79	7	4	11.0	S	1,300
Ky	Louisville	95	76					8.0	SW	459
La.	New Orleans	95	79	95	81	1	9	7.0	SW	9
	Shreveport	100	78	102	81	1	8	6.2	S	197
Me	Portland	90	73					7.3	S	47
Md.	Baltimore	95	78	97	81	2	3	6.9	SW	97
Mass.	Boston	92	75	96	78	3	2	9.2	SW	48
Mich.	Detroit	95	75	101	79	2	2	10.3	SW	619
Minn	Minneapolis	95	75	103	76	2	2	8.4	SE	839
Miss.	Vicksburg	95	78	104	81	2	8	6.2	SW	226
Mo.	Kansas City	100	76	109	79	4	3	9.5	S	741
	St. Louis	95	78	108	80	9	1	9.4	SW	465
Mont.	Helena	95	67	98	70	1	1	7.3	SW	4,090
Nebr	Lincoln	95	75	108	76	6	1	9.3	S	1,180
Nev.	Reno	95	65	102	66	3	1	7.3	W	4,493
N.H.	Manchester	90	73					5.6	NW	
N.J.	Trenton	95	78	103	81	2	2	10.0	SW	56
N.Y.	Albany	92	75	97	78	2	1	7.1	S	19
	Buffalo	93	75	93	75	0	1	12.2	SW	604
	New York	95	75	100	81	2	5	12.9	SW	29
N. Mex.	Santa Fe	90	65	97	68	8	6	6.5	SE	6,994
N.C.	Asheville	90	75	99	79	7	5	5.6	SE	2,192
	Wilmington	95	79	94	81	0	2	7.8	SW	25
N.D.	Bismarck	95	73					8.8	NW	1,670
Ohio	Cincinnati	95	78	106	81	3	2	6.6	SW	553
	Cleveland	95	75	101	79	2	2	9.9	S	651
Okla.	Oklahoma City	101	76	106	79	4	4	10.1	S	1,254
Ore.	Portland	90	65	99	70	1	7	6.6	NW	66
Pa.	Philadelphia	95	78	103	81	2	2	9.7	SW	149
	Pittsburgh	95	75	98	79	2	3	9.0	NW	742
R.I.	Providence	93	75					10.0	NW	11
S.C.	Charleston	95	80	100	81	2	2	9.9	SW	13
	Greenville	95	76					6.8	NE	982
S.D.	Sioux Falls	95	75	108	76	5	1	7.6	S	

Table 21-6. Design Dry- and Wet-bulb Temperatures (*Cont.*)

State	City	Design temp, deg F		Max temp, deg F		No. days per year exceeds design		Summer wind velocity, mph	Prevailing summer wind direction	Altitude, ft
		Dry bulb	Wet bulb	Dry bulb, normal year	Wet bulb, normal year	Dry bulb	Wet bulb			
Tenn	Chattanooga	95	77					6 5	SW	689
	Memphis	95	78	103	81	4	5	7 5	SW	271
Texas	Dallas	100	78	105	80	4	1	9 4	S	460
	El Paso	100	69	101	72	1	2	6 9	E	3,720
	Galveston	95	80					9 7	S	6
	Houston	95	78	100	81	2	5	7 7	S	52
	San Antonio	100	78	102	81	2	2	7 4	SE	646
Utah	Salt Lake City	92	63	102	70	8	5	8 2	SE	4,222
Vt	Burlington	90	73					8 9	S	308
Va	Norfolk	95	78					10 9	S	11
	Richmond	95	78					6 2	SW	162
Wash	Seattle	85	65	86	68	1	4	7 9	S	14
	Spokane	90	65	106	68	5	1	6 5	SW	1,870
W. Va	Parkersburg	95	75					5 3	SE	615
Wis	Madison	95	75					8 1	SW	938
	Milwaukee	95	75					10 4	S	619
Wy	Cheyenne	95	65					9 2	S	6 139

The classification of heat loads (page 1476) all affect the sensible-heat gain within the treated space. There are other factors to be considered, such as the latent-heat gain caused by increasing the moisture content within the conditioned space.

Outside Design Conditions

The outside design atmospheric conditions vary for different geographical locations. The Weather Bureau records show that maximum summer dry-bulb and wet-bulb temperatures occur for a relatively short duration, usually in midafternoon. The peak transmission heat load on a building has a delayed effect, time lag, owing to the residual heat stored within the massive weight of the building structure. The practical design dry-bulb- and wet-bulb-temperature conditions, taking into consideration peaks and prevailing-wind velocities, are given in Table 21-6.

Table 21-6 is applicable for industrial process air-conditioning systems when refrigeration or chilled water is required to maintain specified conditions. For evaporative cooling and humidifying systems not requiring refrigeration, the design is based on the maximum outside wet-bulb depression for that particular geographical location; therefore, the dry-bulb temperatures given in this table should be used. To obtain the maximum wet-bulb depression, refer to the local United States Weather Bureau for that geographical location.

Heat Transmission. The heat transmission through the building structure is computed by determining the area of each type of room surface—walls, windows, partitions, floors, ceilings, or roof. Having determined the area on which transmission heat gain is to be figured, multiply each type of construction surface by its proper coefficient of transmission to obtain the Btu per hour per degree difference. The sum of all the transmission factors, when multiplied by the temperature difference between outside and inside design dry-bulb temperatures, will give the total Btu per hour.

Heat-transmission coefficients common to mill- or plant-type construction are given in Table 21-7.

Table 21-7. Heat-transmission Coefficients of Building Materials*

Exterior Walls

Masonry construction

Masonry side walls	Thickness in	Uninsulated			Insulated		
		Plain walls no finish	½ in plaster on walls	Plaster on gypsum lath furred	½-in plaster, ½-in. rigid insulation furred	½-in plaster, 1-in. rigid insulation furred	½-in. plaster gypsum lath, 1-in. flexible insulation furred
Brick 4-in hard remainder common	8	0 50	0 46	0 30	0 22	0 16	0 14
	12	0 35	0 34	0 24	0 19	0 14	0 13
	16	0 28	0 27	0 20	0 16	0 13	0 12
Hollow tile stucco exterior	8	0 40	0 37	0 26	0 20	0 15	0 13
	12	0 30	0 29	0 21	0 17	0 13	0.12
	16	0 25	0 24	0 19	0 15	0 12	0 11
Limestone or sandstone	8	0 70	0 64	0 36	0 25	0 18	0 16
	12	0 57	0 53	0 33	0 23	0 17	0 15
	16	0 49	0 45	0 30	0 22	0 16	0 14
Concrete monolithic	6	0 79	0 71	0 39	0 26	0 19	0 16
	10	0 63	0 58	0 34	0 24	0 18	0 15
	12	0 58	0 53	0 33	0 23	0 17	0 15
Hollow concrete blocks gravel aggregate	8	0 56	0 52	0 32	0 22	0 17	0 15
	12	0 50	0 46	0 30	0 19	0 16	0 14
4 in brick veneer hollow tile backing	6†	0 35	0 34	0 24	0 18	0 14	0 13
	8†	0 34	0 32	0 23	0 18	0 14	0 13
4 in brick veneer concrete backing	6†	0 59	0 54	0 33	0 23	0 17	0 15
	8†	0 54	0 50	0 31	0 23	0 17	0 15
4 in cut-stone veneer hollow tile backing	6†	0 37	0 35	0 25	0 19	0 15	0 13
	8†	0 36	0 34	0 24	0 19	0 15	0 13
4 in cut-stone veneer concrete backing	6†	0 63	0 58	0 34	0 24	0 18	0 15
	8†	0 57	0 53	0 33	0 23	0 17	0 15

† Thickness of backing

NOTE: Coefficients are expressed in Btu per hour per square foot per deg F difference in temperature between the air on the two sides and with a wind velocity of 15 mph

Frame construction

Exterior walls		Type of sheathing			
Exterior finish	Interior finish	Gypsum ½ in thick	Plywood $\frac{5}{16}$ in	Wood $\frac{25}{32}$ in., paper	Insulation $\frac{25}{32}$ in thick
Wood siding (clapboard)	Metal lath and plaster	0 33	0 32	0 26	0 20
	Gypsum board	0 32	0 32	0 25	0 20
	Wood or gypsum lath and plaster	0 31	0 31	0 25	0 19
	Plywood	0 30	0 30	0 24	0 19
	Insulating board lath ½ in and plaster	0 22	0 22	0 19	0 15
Wood shingles	Metal lath and plaster	0 25	0 25	0 26	0 17
	Gypsum board	0 25	0 25	0 25	0 17
	Wood or gypsum lath and plaster	0 24	0 24	0.25	0 16
	Plywood	0 24	0 24	0 24	0.16
	Insulating board lath ½ in and plaster	0 19	0 18	0 19	0 13
Brick veneer	Metal lath and plaster	0 37	0 36	0 28	0 21
	Gypsum board	0 36	0 36	0 28	0 21
	Wood or gypsum lath and plaster	0 35	0 34	0 27	0 20
	Plywood	0 34	0 33	0 27	0 20
	Insulating board lath ½ in and plaster	0 24	0 24	0 20	0.16

NOTE Coefficients are expressed in Btu per hour per square foot per deg F difference in temperature between the air on the two sides and with a wind velocity of 15 mph

Table 21-7. Heat-transmission Coefficients of Building Materials* (*Cont.*)

Windows and skylights

Description	*Coefficient*
Single	1.13
Double	0.45
Triple	0 281

Hollow-glass-block walls

Description	*Coefficient*
Smooth-surface blocks (7¾ × 7¾ × 3⅞ in.)	0.49
Ribbed-surface blocks (7¾ × 7¾ × 3⅞ in.)	0.46

Solid wood doors

Nominal thickness, in.	Coefficient	
	Exposed door	Glass storm door
1	0.69	0.42
1¼	0.59	0.38
1½	0.52	0.35
1¾	0.51	0.35
2	0.46	0.32
2½	0.38	0.28
3	0.33	0.25

Note: Coefficients are expressed in Btu per hour per square foot per deg F difference in temperatur between the air on the two sides and with an outside wind velocity of 15 mph.

Interior Walls

Masonry partitions

Type of partition	Thick-	Type of finish		
		No finish (plain walls)	Plaster one side	Plaster both sides
Hollow tile, clay		0.50	0.47	0.43
		0.45	0.42	0.40
Hollow gypsum tile		0.35	0.33	0.32
		0.29	0.28	0.27
Hollow concrete tile or blocks, cinder aggregate		0.50	0.47	0 43
		0.45	0.42	0.40
Lightweight aggregate		0.41	0.39	0.37
		0.35	0.34	0.32
Common brick		0.50	0.46	0.43

Note: Coefficients are expressed in Btu per hour per square foot per deg F difference in temperature between still-air conditions on both sides.

Frame partitions (wood and studs)

Interior finish	Single partition (finish on only one side of studs)	Double partition	
		No insulation between studs	1-in. blanket between studs
Metal lath and plaster (¾ in.)	0.69	0.39	0.16
Gypsum board (⅜ in.)	0.67	0.37	0.16
Wood lath and plaster (¾ in.)	0.62	0.34	0.15
Gypsum lath (⅜ in.) plastered	0.61	0.34	0.15
Plywood (⅜ in.) plain	0.59	0.33	0.15
Insulating board (½ in.) plain	0.36	0.19	0.11
Insulating board (½ in.) plastered	0.35	0.18	0.11
Insulating board (1 in.) plastered	0.23	0.12	0.082

Note: Coefficients are expressed in Btu per hour per square foot per deg F difference in temperature between still-air conditions on both sides.

Table 21-7. Heat-transmission Coefficients of Building Materials* (*Cont.*)

Floors and Ceilings

Concrete construction

Type of ceiling	Thickness of concrete, in.	Type of flooring: No flooring (bare concrete)	Tile or terrazzo on concrete	¼ in. battleship linoleum on concrete	Parquet flooring in mastic on concrete	Double wood floor in sleepers
No ceiling	3	0.69	0.65	0.45	0.45	0.25
	6	0.59	0.56	0.41	0.41	0.23
	10	0.50	0.48	0.36	0.36	0.22
½ in. plaster applied directly to underside of concrete	3	0.62	0.59	0.43	0.43	0.24
	6	0.54	0.52	0.39	0.39	0.22
	10	0.46	0.44	0.34	0.34	0.21
Suspended or furred metal lath and plaster	3	0.38	0.37	0.30	0.30	0.19
	6	0.35	0.34	0.28	0.28	0.18
	10	0.32	0.31	0.26	0.26	0.17
Suspended or furred gypsum board and plaster	3	0.36	0.35	0.28	0.28	0.19
	6	0.33	0.32	0.27	0.27	0.18
	10	0.30	0.29	0.24	0.24	0.17
Suspended or furred insulating-board lath and plaster	3	0.25	0.24	0.21	0.21	0.15
	6	0.23	0.23	0.20	0.20	0.15
	10	0.22	0.21	0.19	0.19	0.14

NOTE: Coefficients are expressed in Btu per square foot per deg F difference in temperature between the air on the two sides, and with still air on both sides (no wind).

Frame construction

Type of ceiling	Insulation between or on joists—no flooring above												With flooring	
	None	Insulating board on joists		Blanket or bat insulation between joists			Vermiculite insulation between joists			Mineral-wool insulation between joists			Single wood floor	Double wood floor
		½ in.	1 in.	1 in.	2 in.	3 in.	2 in.	3 in.	4 in.	2 in.	3 in.	4 in.		
No ceiling	...	0.37	0.24	....	.....	.....	....	.....	.....	.....	.....	.....	0.45	0.34
Metal lath and plaster	0.69	0.26	0.19	0.19	0.12	0.093	0.18	0.14	0.11	0.12	0.092	0.066	0.30	0.25
Gypsum board (⅜ in.)	0.67	0.26	0.18	0.19	0.12	0.092	0.18	0.14	0.10	0.12	0.092	0.066	0.30	0.24
Wood lath and plaster	0.62	0.25	0.18	0.19	0.12	0.092	0.17	0.14	0.10	0.12	0.091	0.065	0.29	0.24
Gypsum lath and plaster	0.61	0.25	0.18	0.19	0.12	0.092	0.17	0.14	0.10	0.12	0.091	0.065	0.28	0.24
Plywood (⅝ in.)	0.59	0.24	0.18	0.19	0.12	0.09	0.17	0.14	0.10	0.12	0.09	0.065	0.28	0.23
Insulating board (½ in.)	0.37	0.19	0.15	0.16	0.11	0.082	0.15	0.11	0.092	0.11	0.082	0.060	0.22	0.19
½-in. insulating-board lath and plaster	0.35	0.19	0.15	0.15	0.10	0.081	0.14	0.11	0.091	0.10	0.082	0.060	0.21	0.18
1-in. insulating-board lath and plaster	0.23	0.15	0.12	0.12	0.089	0.072	0.12	0.097	0.080	0.089	0.072	0.055	0.16	0.14

NOTE: Coefficients are expressed in Btu per hour per square foot per deg F difference in temperature between the air on the two sides and with still air on both sides (no wind).

Ground Floors. Until more complete data are available, it is recommended that a coefficient of 0.10 be used for all types of concrete floors on the ground, with or without insulation. For basement walls below grade, use the same average coefficient (0.10).

For basement walls below grade, a ground temperature of 32°F is recommended. For basement floors below grade, a ground temperature of from 40 to 55°F, depending on geographic location, should be used.

Table 21-7. Heat-transmission Coefficients of Building Materials* (Cont.)

Roofs

Flat roofs, concrete and frame construction

Flat roofs covered with built-up roofing		Without ceiling, underside roof exposed			With metal lath and plaster ceilings		
Type of roof deck	*Thickness of roof deck, in.*	No insulation	½-in. rigid insulation	1-in. rigid insulation	No insulation	½-in. rigid insulation	1-in. rigid insulation
Precast cement tile	1⅝	0 84	0.37	0.24	0.43	0.26	0.19
Concrete	2	0 82	0.37	0.24	0.42	0.26	0.19
	4	0.72	0.34	0.23	0.40	0.25	0.18
	6	0.65	0.33	0.22	0.37	0 24	0.18
Wood	1	0.49	0.28	0.20	0.32	0.21	0 16
	1½	0.37	0.24	0.18	0.26	0.19	0.15
	2	0.32	0.22	0.16	0.24	0 17	0.14
	4	0.23	0.17	0.14	0.18	0 14	0.12
Flat metal roofs	...	0.94	0.39	0.24	0.46	0.27	0.19

NOTE: Bare corrugated iron without roofing has a coefficient of 1.50 based on the projected area.

Coefficients are expressed in Btu per hour per square foot per deg F difference in temperature between the air on the two sides and with a wind velocity of 15 mph.

Pitched roofs, frame construction

Type of ceiling (applied to roof rafters)	Type of roofing and sheathing											
	Wood shingles on 1- × 4-in. wood strips spaced 2 in.				Asphalt shingles or roll roofing on solid wood sheathing				Slate or tile on solid wood sheathing			
	Insulation between rafters				Insulation between rafters				Insulation between rafters			
	None	Blanket or bat			None	Blanket or bat			None	Blanket or bat		
		1 in.	2 in.	3 in.		1 in.	2 in.	3 in.		1 in.	2 in.	3 in.
No ceiling	0.48	0.15	0.11	0 083	0.53	0.15	0.11	0 085	0.55	0.16	0.11	0.085
Metal lath and plaster	0.31	0.14	0.10	0.082	0.33	0.15	0.10	0.083	0.34	0.15	0.11	0 083
Gypsum board (⅜ in.)	0.31	0.14	0 10	0.082	0.32	0.15	0.10	0.082	0.34	0.15	0.11	0 083
Wood lath and plaster	0.30	0.14	0.10	0.080	0.31	0.14	0.10	0.082	0.32	0.15	0.10	0.082
Gypsum lath and plaster	0.29	0.14	0.10	0.080	0.31	0.14	0.10	0.082	0 32	0.15	0.10	0 082
Plywood (⅜ in.)	0.29	0.14	0.10	0.080	0.30	0.14	0 10	0.080	0 31	0.14	0 10	0 082
Insulating board (½ in.)	0.22	0.12	0.09	0.073	0.23	0.12	0.093	0.074	0.24	0.13	0.094	0.076
Insulating-board lath (½ in.) plastered	0.22	0.12	0.09	0 073	0.22	0.12	0.090	0.073	0.23	0.12	0.093	0.074
Insulating-board lath (1 in.) plastered	0 16	0.10	0.077	0.065	0.17	0.10	0.077	0.065	0.17	0.10	0.080	0.066

NOTE: Coefficients are expressed in Btu per hour per square foot per deg F difference in temperature between the air on the two sides and with a wind velocity of 15 mph.

* Reprinted by permission of the American Society of Heating and Ventilating Engineers, "ASH & VE Guide."

As a word of caution, when selecting an insulating building material, in surface contact with *humidified air*, the material must be impervious to the penetration of moisture.

Solar-heat Transmission. The solar-heat transmission, or sun effect, through building structures is difficult to determine owing to the periodic character of heat flow, variations in radiation, intensity on differently oriented surfaces, geographical location, and further effective time lag due to the mass of the building materials. The more generally accepted thesis is that the sun's rays will heat the outer surface of the walls and roofs and thereby give the same effect as an increase in temperature difference between inside and outside conditions. The sun's rays strike the roof nearly perpendicularly and strike the walls at various angles. Hence the roof surface is raised to a higher temperature than the walls. The effect of the sun's rays on glass differs in another respect. A small amount of heat energy is reflected, some is absorbed with slight rise in glass temperature, but the greater portion of the radiant heat energy passes through the glass and is absorbed on interior surfaces. This heat is quickly given up to the surrounding air, raising the temperature in the treated space. It is quite apparent that a barrier to solar heat's direct rays of the sun must be employed by means of awnings, white shades, paint, or blinds. Common mill practice is to paint the glass white. This presents an opaque nontransparent surface to reflect or absorb the radiant heat from the sun's rays.

There is some difference of opinion among recognized authorities as to how much of a heat load is imposed by the sun. However, one practical method, considered good engineering practice, is to calculate the sun effect separately by figuring 25°F increased temperature difference on roofs and 15°F temperature difference on walls for northern hemispheres and increasing these factors to 35°F on roofs and 20°F on walls in southern latitudes. In tropical climates these temperature differences would be further increased, and proper factors would be determined by the judgment and experience of the design engineer.

Heat Load from Occupants. The heat and moisture given off by persons vary with the activity of the individual and in a proportional degree are affected by the temperature and relative humidity. The total heat given off by a person is the sum of the sensible and latent heats and is practically a constant depending upon the activity of that person.

Table 21-8. Heat of Persons, Btu per Hour

Design room temperature	Activity *A* per person moving slowly			Activity *B* per person working rapidly		
	Sensible heat	Latent heat	Total heat	Sensible heat	Latent heat	Total heat
76	256	144	400	270	390	660
78	240	160	400	240	420	660
80	220	180	400	210	450	660
82	200	200	400	180	480	660
84	180	220	400	150	510	660

A practical basis of design in process air conditioning is to figure a heat-load factor of 500 Btu per hr total heat per person.

Heat Load from Lighting. Lighting is a usual source of heat generation in design calculations for process air-conditioning systems. The total lighting load requires analysis and good judgment to determine the effective lighting load factor. When considering a building with a saw-tooth or monitor roof, it is not necessary to figure both sun load and light load simultaneously. The greater of these two factors should be figured. Each watt of lighting consumed represents a heat load of 3.41 Btu per hr heat gain in the conditioned space.

Heat Load from Motors. The power generated by all machinery motors within the treated space is dissipated in the form of heat within that space. The engineering design must evaluate the motor-horsepower factor and connected horsepower within the treated space, *i.e.*, the total horsepower may be effective continuous operation, or the motors may operate intermittently. It is frequently permissible to figure less than the connected horsepower in operation at one time; on the other hand, it is advisable at times to increase the internal motor-horsepower load owing to overload operations. Electric motors are generally rated in units of horsepower output: 1 hp is equivalent to 2,545 Btu per hr. The following is a basis for calculating horsepower:

1. When motor and driving load are located within the treated space,

$$\text{Heat generated, Btu per hr} = 3.41 \times \text{watts input} = \frac{2545 \times (\text{actual brake horsepower})}{\text{motor efficiency}}$$

2. When motor is outside room and driving load is within the treated space,

$$\text{Heat generated, Btu per hr} = 2545 \times (\text{actual brake horsepower})$$

3. When motor is driving fan supplying air into the treated space through duct work, the heat equivalent in energy, applied to the air, is transmitted to the room.

$$\text{Heat generated, Btu per hr} = 2545 \times (\text{actual brake horsepower})$$

Customary practice is to add 10 per cent to the previously calculated sensible-heat load to provide for the fan-horsepower heat load. Then after the fan has been selected and actual brake horsepower determined, a back check is generally advisable.

Table 21-9. Motor Efficiency Ratings

Rating, hp	Efficiency average, per cent	Multiplying factor	Equivalent Btu/hp-hr
⅛–1	80	1.25	3170 (input)
1–10	85	1.18	3000 (input)
10 and above	88	1.14	2900 (input)

Other Heat-load-producing Factors. Heat-producing appliances, giving off heat, either sensible or latent heat, or both, must be determined and included in calculations. The generating sources may be uncovered steam pipes, hot surfaces, gas-fired heaters, electric generators, steam-heated cylinders, or open vats giving off moisture to the treated areas. All must be carefully analyzed and the heat produced taken into design calculations, or else these heat sources must be hooded and heat or moisture exhausted to atmosphere.

It is obvious from the foregoing discussion that the determination of the heat-load calculation is rather complicated by reason of the variable character of the contributing load components. A careful survey of the legitimate sources of heat loads will reveal in practice that it is not necessary to lump together "ignorance" and add safety factors sufficient to increase the apparatus and equipment capacity in exaggerated proportion to the economic actual requirements.

Ventilation. Ventilation requirements are determined as follows:

1. Sufficient fresh outdoor air is introduced into the room so that a slight static air pressure will be maintained within the conditioned space; losses are outward; there is no infiltration.
2. Sufficient fresh outdoor air to maintain adequate air for the occupants, keeping the dilution of obnoxious gases, when present, to the requirements of safety standards.
3. Sufficient fresh outdoor air must be provided to make up for air exhausted from manufacturing processes, in addition to requirement 1, to maintain a balanced air pressure within the treated space.

The estimation of fresh-outdoor-air requirements to maintain an air pressure within the treated area is generally calculated by the "air-change method," which is the determination of air leakage outward by roughly assuming a certain number of air changes per hour for each treated area, this assumption being dependent upon the type, use, and location of the building and whether it is an old wooden structure or a new modern brick or windowless type of building. This method, predicated upon actual experience records tabulated from many successful applications, has proved practical and reliable, thus eliminating the exhausting method of determining air leakage through doors and window cracks, with variable prevailing-wind pressures. This latter method may be used when more accurate and exact calculations are required.

Air-change Method. Air changes per hour under average conditions to maintain an air pressure within the treated room may be taken as shown in Table 21-10.

Table 21-10

Location and type (one or all four sides exposed)	Assumed air changes	
	Minutes per air change	Changes per hour
Old wooden-mill-type construction	30	2
Average mill-type construction	60–90	1–⅔
New brick modern or windowless building	120	½

To determine ventilation requirements, divide the cubical contents of the treated area by the assumed minutes per air change. This gives air-volume requirements directly in cubic feet per minute. Except in extreme cases, this air volume is generally sufficient and well above the cubic feet per minute per person required to meet the ventilation health standards for the occupants.

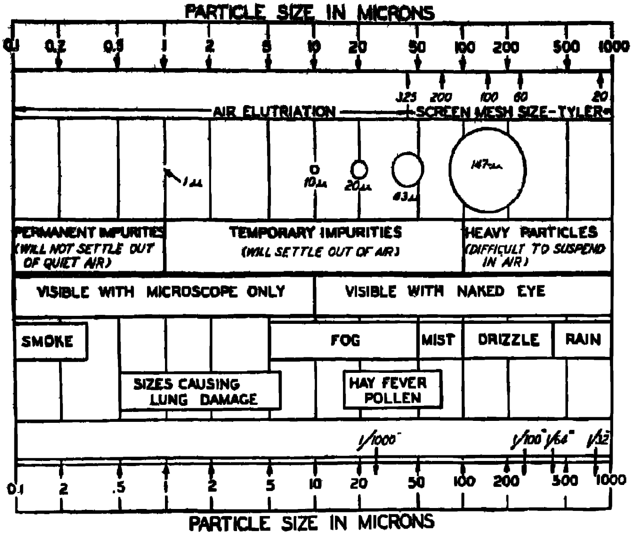
To determine ventilation requirements when obnoxious gases are present, refer to Table 21-11. The amounts and rate of production of the obnoxious gas are first determined, and then sufficient fresh outdoor air is introduced to keep the dilution concentration well within the standards of safety.

Table 21-11. Maximum Allowable Concentration of Toxic Fumes*

Gases	*Cu ft per million cu ft of air*	*Solvents*	*Cu ft per million cu ft of air*
Arsine	1	Amyl acetate	400
Ammonia	100	Aniline	5
Carbon monoxide	100	Benzene	75
Chlorine	1	Butyl acetate	400
Formaldehyde	20	Carbon bisulfide	15
Hydrogen cyanide	20	Carbon tetrachloride	30
Hydrochloric acid	10	Dichlorbenzene	75
Hydrogen fluoride	3	Dichlorethyl ether	15
Hydrogen sulfide	20	Ether	400
Nitrogen dioxide	10	Ethylene dichloride	100
Ozone	1	Methanol	200
Phosphine	2	Monochlorbenzene	75
Sulfur dioxide	10	Naphtha, gasoline	1000
Metallic fumes	*Mg per cu m of air*	Nitrobenzene	5
		Tetrachlorethane	10
Cadmium	0.1	Tetrachlorethylene	200
Chromic acid	0.1	Toluene	200
Lead	0.15	Trichlorethylene	200
Manganese	30.0	Turpentine	200
Mercury	0.1	Xylene, coal-tar naphtha	200
Zinc oxide fumes	15.0		

* From "Toxic Fumes in Massachusetts Industry," published by Massachusetts Division of Occupational Hygiene.

Table 21-12. Classification of Dust Particles and Standard Sieve Sizes



U. S. Bureau of Standards			W. S. Tyler Co. screen		
Mesh No.	Microns	Opening, in.	Mesh No.	Microns	Opening, in.
10	2000	0.0787	9	1981	0.0780
12	1680	0.0661	10	1651	0.0650
16	1190	0.0469	14	1168	0.0460
20	840	0.0331	20	833	0.0328
30	590	0.0232	28	589	0.0232
40	420	0.0165	35	417	0.0164
50	297	0.0117	48	295	0.0116
60*	250	0.0098	60	246	0.0097
70	210	0.0083	65	208	0.0082
80	177	0.0070	80	175	0.0069
100*	149	0.0059	100	147	0.0058
140	105	0.0041	150	104	0.0041
170	88	0.0035	170	88	0.0035
200*	74	0.0029	200	74	0.0029
230	62	0.0024	250	61	0.0024
270	53	0.0021	270	53	0.0021
325*	44	0.0017	325	43	0.0017

* Screen sizes used in ASME Power Test Code for Dust Separating Apparatus.

Air Purity. Dust suspended in air consists of solid particles of any material, all considered contaminative and generally most objectionable in process manufacturing. The control of air purity is one of the functions of process air conditioning. The control of dust consists of collection of the dust at the point of entry to the air-conditioning apparatus and the separation and elimination of the dust entrained in the air stream. The industrial dust problem is generally limited to solid particles larger than 10 microns. Thus this discussion will exclude fumes, smoke, and odors which are controlled by electric precipitation. Relative sizes of various dusts are shown in Table 21-12.

Industrial dust control has always been a problem in process manufacturing, and only in recent years has extensive research been conducted.

Dust problems may be classified as follows:

1. *Dust nuisance*—Causing damage to machinery, contamination, injury, or spoilage to product, and unclean working area.

2. *Fire and explosive hazard*—From any combustible dust which will oxidize rapidly or burn under certain concentrations; a potential hazard from explosion and fire.

3. *Occupational diseases*—Health of workers is impaired by various types of dusts and in most states the elimination of the hazard is required.

Dust varies in size, specific gravity, and shape. Size is the most important variable and is usually given as a mean diameter measured in microns or in screen sizes. A micron is one millionth of a meter, or 1/25,400 in. in size. Screen sizes are given in mesh or the number of openings per lineal inch of standard screen. Table 21-12 shows the U.S. Bureau of Standards standard test screens and their opening size in both microns and decimal parts of an inch.

Dust is rarely all of one size, and dust samples will be found to have a wide range of sizes, with the percentage of different sizes varying. It is thus impractical to specify a dust size by one number. The usual description of a dust includes the percentages of the dust in various size groups (either micron or mesh sizes). The determination of dust concentration or measurement of dust count is involved and many times rather complicated. Some of the methods of determining dust count are impingement on a prepared viscous surface, collection on filter paper, electric precipitation, and by direct analysis of samples by laboratory methods. The filter-paper method, by which air passes through filter paper for a known time period, is a convenient method of sampling, counting, and determining for analysis of the dust problem in process manufacturing.

Fire and Explosion Hazard from Dust. Dust explosions are caused by very rapid burning of air-borne dust, causing a rapid pressure rise. The explosion is generally the result of a limited concentration of dust in suspension in contact with some source of ignition. The resulting increase of pressure frequently dislodges large accumulations of dust within the building, resulting in secondary and more violent explosions. Thus the process air-conditioning problem is to prevent accumulations of inflammable dust in explosive concentration, to know rate of dispersion and the moisture content, and to determine the explosive pressure in known air concentrations. The minimum explosive concentration of air-borne dust generally ranges from 0.01 to 0.5 oz per cu ft, or 10 to 500 g per cu m. The industrial process air-conditioning engineer's problem primarily is dealing with hygroscopic dusts, moisture content, and elimination of static spark-ignition hazard by maintaining proper relative humidity within the treated space, thus prevent fire and resulting explosion.

Air Movement. The design of an industrial process air-conditioning system for ultimate effectiveness and efficiency depends upon an ample amount of air being circulated within the treated area to balance the internal heat-load gains. Since the treated or conditioned air is the mechanical carrier of heat and moisture to, or away from, process materials, sufficient air must be circulated and distributed uniformly in proportion to the load requirements encountered and varied with the variations of the heat load to maintain constant atmospheric conditions. The method of diffusing the air from supply outlets within the treated space is primarily a consideration of the material in process. Some products are very sensitive to direct air contact, suffering breakage or hardening of the surface, whereas other products require a direct air blast, *e.g.*, crystallization process. Types of supply outlets may be classified on a basis of outlet velocities, *e.g.*, low velocity (grille outlets), medium velocity (slotted outlets), and high velocity (nozzle outlets). The terms low, medium, and high are relative and vary with practical considerations of the problem involved. The higher the velocity, the greater the induction of room air.

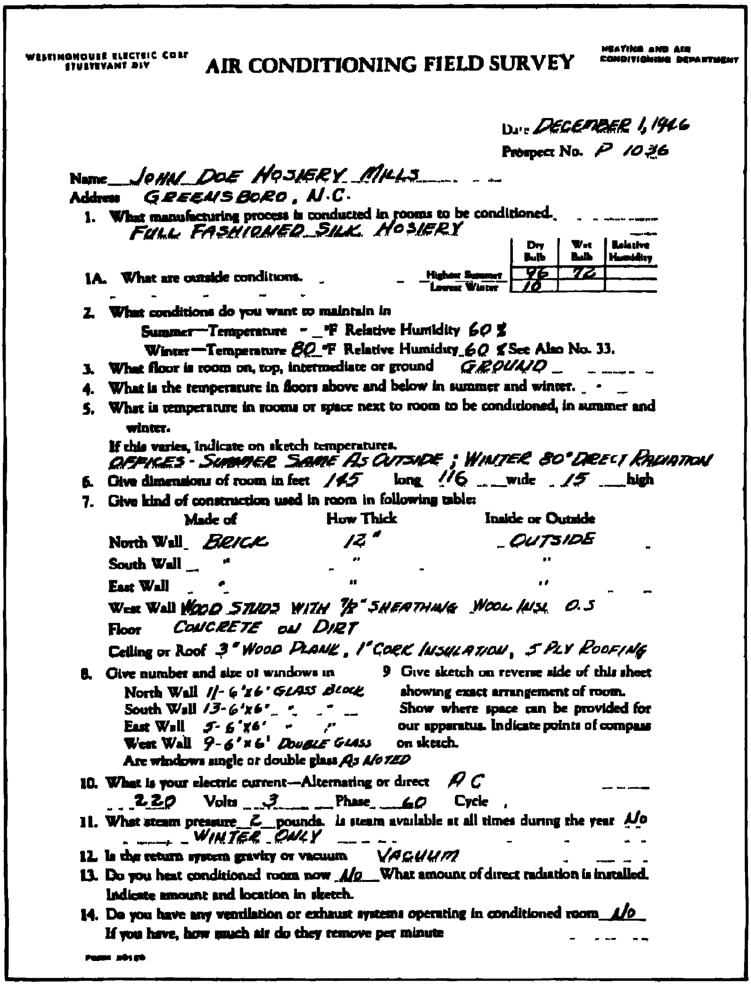
Table 21-13. Relative Comparison of Supply-outlet Velocities

Velocity	*Fpm*
Low	Up to 600
Medium	600-1,200
High	Above 1,200

There are many variations of supply outlets—slotted grilles, fishtail diffusers, horizontal and vertical slots, and rectangular or circular nozzles. Their selection all depends upon practical experience and judgment in application to the particular problem under consideration. The ultimate decision and results to be obtained are to maintain uniform air conditions throughout the treated area without damage to the process materials, spoilage, delays, and resultant loss of labor. Conditioned air supply and diffusion are very important factors in process air conditioning. *No system is better than its air-distribution system.*

Calculations of Typical Process Air-conditioning Systems

EXAMPLE 1: Evaporative cooling—humidifying, heating, and ventilating system—no refrigeration required.



WESTINGHOUSE ELECTRIC CORP
STURTEVANT DIV

AIR CONDITIONING FIELD SURVEY

HEATING AND AIR CONDITIONING DEPARTMENT

Date DECEMBER 1, 1946
Prospect No. P 1036

Name JOHN DOE HOSIERY MILLS
Address GREENSBORO, N.C.

1. What manufacturing process is conducted in rooms to be conditioned. FULL FASHIONED SILK HOSIERY

1A. What are outside conditions.

	Dry Bulb	Wet Bulb	Relative Humidity
Highest Summer	96	72	
Lowest Winter	10		

2. What conditions do you want to maintain in
Summer—Temperature - °F Relative Humidity 60 %
Winter—Temperature 80 °F Relative Humidity 60 % See Also No. 33.

3. What floor is room on, top, intermediate or ground GROUND

4. What is the temperature in floors above and below in summer and winter.

5. What is temperature in rooms or space next to room to be conditioned, in summer and winter.
If this varies, indicate on sketch temperatures.
OFFICES - SUMMER SAME AS OUTSIDE; WINTER 80° DIRECT RADIATION

6. Give dimensions of room in feet 145 long 116 wide 15 high

7. Give kind of construction used in room in following table:

	Made of	How Thick	Inside or Outside
North Wall	BRICK	12"	OUTSIDE
South Wall	"	"	"
East Wall	"	"	"
West Wall	WOOD STUDS WITH 7/8" SHEATHING WOOL INSL 0.5		
Floor	CONCRETE ON DIRT		
Ceiling or Roof	3" WOOD PLANK, 1" CORK INSULATION, 5 PLY ROOFING		

8. Give number and size of windows in
North Wall 11- 6'x6' GLASS BLOCK
South Wall 13-6'x6' " "
East Wall 5- 6'x6' " "
West Wall 9-6'x6' DOUBLE GLASS
Are windows single or double glass AS NOTED

9 Give sketch on reverse side of this sheet showing exact arrangement of room. Show where space can be provided for our apparatus. Indicate points of compass on sketch.

10. What is your electric current—Alternating or direct AC
220 Volts 3 Phase 60 Cycle

11. What steam pressure 2 pounds. Is steam available at all times during the year NO
WINTER ONLY

12. Is the return system gravity or vacuum VACUUM

13. Do you heat conditioned room now NO What amount of direct radiation is installed. Indicate amount and location in sketch.

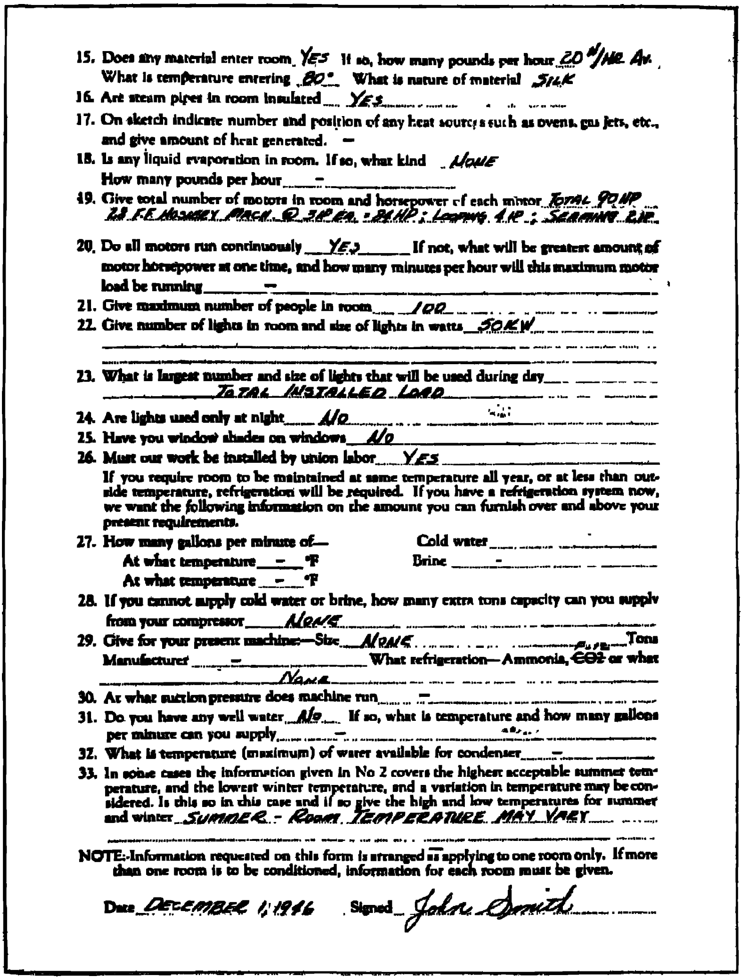
14. Do you have any ventilation or exhaust systems operating in conditioned room NO
If you have, how much air do they remove per minute

FIG. 21-14. Field survey, page 1.

The atmospheric conditions required in the treated area would be as follows:

1. *Summer*—Constant relative humidity, varying dry-bulb temperature
2. *Winter*—Constant dry-bulb temperature and relative humidity

Basis of Design. Refer to Figs. 21-14, 21-15, 21-16, and use these given factors as a basis of design. Then calculate the heat transmission through the walls, windows, and roof on a per-degree temperature difference (see Figs. 21-17, 21-18). Note that the outside summer atmospheric conditions are figured at the maximum wet-bulb depression because the transmission through the building will be the greatest under these conditions, thereby requiring the maximum air volume to balance the sensible-heat load. Remember the fact, as previously explained, that the summer requirements are the controlling factor in establishing air volume. Likewise, in the summer the fundamental law of saturation to spray-type humidifiers applies, *i.e.*, outside fresh air passing through a humidifier will *saturate* leaving the humidifier at the *entering wet-bulb temperature.* At saturation the dry-bulb and wet-bulb temperatures and dew point come together. Thus the leaving dew point (which is the same as the entering wet bulb) now becomes the dew-point temperature of the treated area. The air volume required to maintain the desired relative humidity, corresponding to that dew point, will be based on the sensible-heat rise from the dew-point temperature (also dry-bulb temperature) leaving the humidifier to the treated-area dry-



15. Does any material enter room YES If so, how many pounds per hour 20 #/HR. AV.
What is temperature entering 80° What is nature of material SILK

16. Are steam pipes in room insulated YES

17. On sketch indicate number and position of any heat sources such as ovens, gas jets, etc., and give amount of heat generated. —

18. Is any liquid evaporation in room. If so, what kind NONE
How many pounds per hour —

19. Give total number of motors in room and horsepower of each motor TOTAL 90 HP
28 F.F. HOSIERY MACH. @ 3 HP EA. = 84 HP; LOOPING 4 HP; SEAMING 2 HP.

20. Do all motors run continuously YES If not, what will be greatest amount of motor horsepower at one time, and how many minutes per hour will this maximum motor load be running —

21. Give maximum number of people in room 100

22. Give number of lights in room and size of lights in watts 50 KW

23. What is largest number and size of lights that will be used during day
TOTAL INSTALLED LOAD

24. Are lights used only at night NO

25. Have you window shades on windows NO

26. Must our work be installed by union labor YES

If you require room to be maintained at same temperature all year, or at less than outside temperature, refrigeration will be required. If you have a refrigeration system now, we want the following information on the amount you can furnish over and above your present requirements.

27. How many gallons per minute of— Cold water
At what temperature — °F Brine —
At what temperature — °F

28. If you cannot supply cold water or brine, how many extra tons capacity can you supply from your compressor NONE

29. Give for your present machine:—Size NONE Tons
Manufacturer — What refrigeration—Ammonia, ~~CO2~~ or what
NONE

30. At what suction pressure does machine run —

31. Do you have any well water NO If so, what is temperature and how many gallons per minute can you supply —

32. What is temperature (maximum) of water available for condenser —

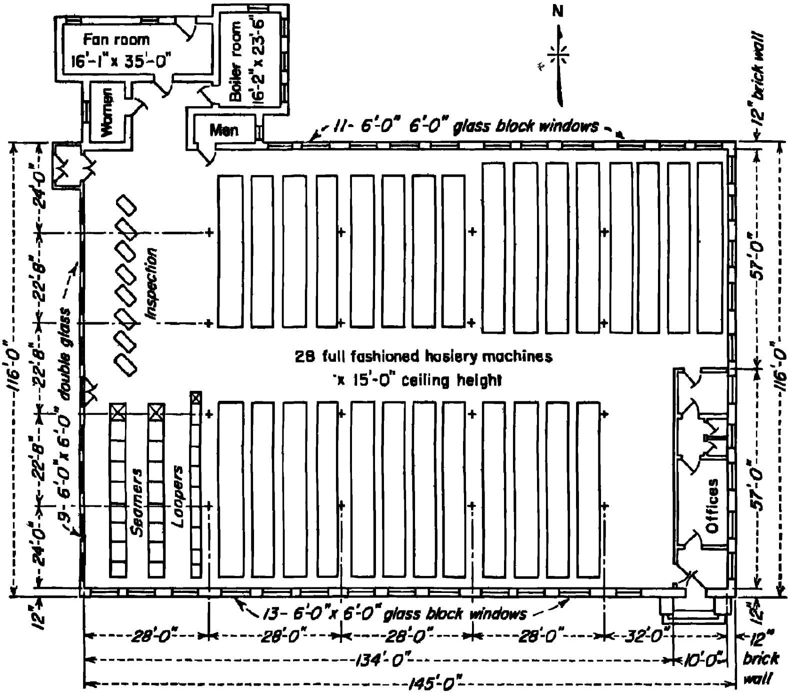
33. In some cases the information given in No 2 covers the highest acceptable summer temperature, and the lowest winter temperature, and a variation in temperature may be considered. Is this so in this case and if so give the high and low temperatures for summer and winter SUMMER - ROOM TEMPERATURE MAY VARY

NOTE:-Information requested on this form is arranged as applying to one room only. If more than one room is to be conditioned, information for each room must be given.

Date DECEMBER 1, 1946 Signed John Smith

FIG. 21-15. Field survey, page 2.

bulb temperature. The design for this load factor in the summer requires all outside fresh air through the humidifier for cooling by evaporation, thereby providing ample ventilation; in fact, pressure-relief dampers in the outside walls must be furnished to release this air to atmosphere. The relative humidity in the treated area will remain a constant, but the inside dry-bulb temperature will vary with the outside wet bulb. The reason that the relative humidity remains a constant is based on the *psychrometric phenomenon*, that, for a given relative humidity, there is practically a constant wet-bulb depression within the operating range of atmospheric conditions generally encountered in industry. The winter operation functions the same as for summer, except as the outside wet-bulb temperature drops below the winter dew-point temperature, air is automatically recirculated from the treated area, with a higher wet-bulb temperature, to the inlet side of the humidifier, so that the wet bulb of the mixture will maintain a constant winter dew-point temperature leaving the humidifier. Winter ventilation is adjusted to a minimum based on the air-change method. Generally the heat in the return air from the treated area is sufficient, so that no fresh-air heaters are required. In the summer, the atmospheric conditions in the treated area are maintained by varying the volume of supply air by means of a hygrostat, thereby controlling constant relative humidity. The dry-bulb temperature varies with the outside wet bulb. In the winter, the atmospheric conditions in the treated area are maintained by controlling the dew-point temperature leaving the humidifier and the dry-bulb temperature of the room. Thus the corresponding relative humidity and wet bulb are maintained.



Plan 1st floor

FIG. 21-16. Field survey, page 3.

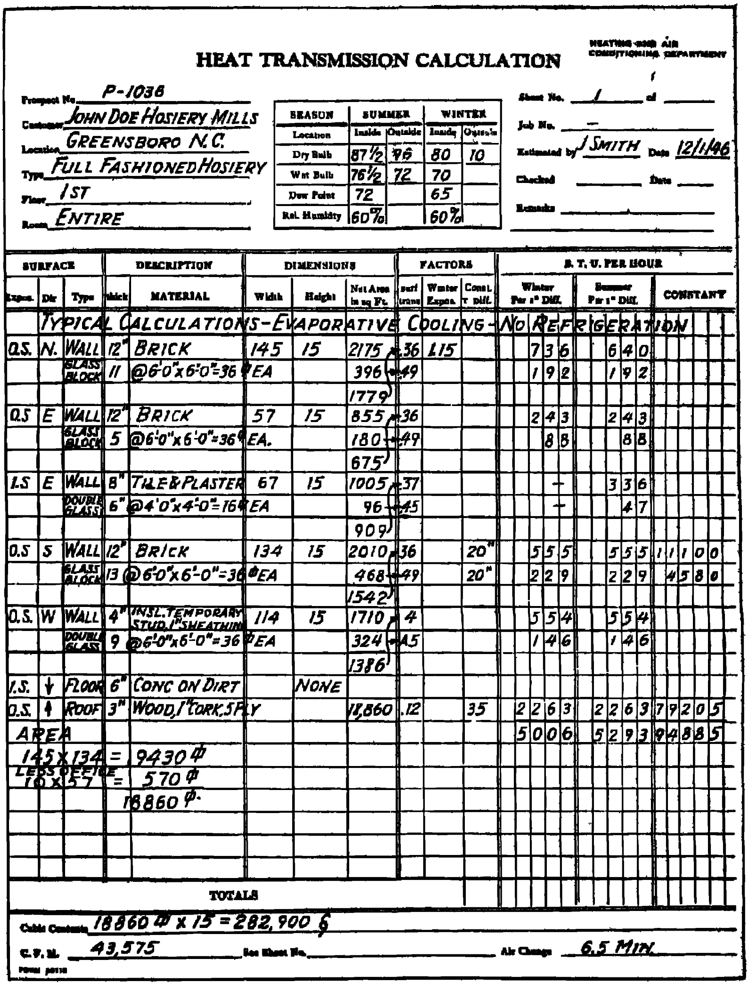
The evaporative cooling (cooling by evaporation), humidifying, heating, and ventilating system is applicable to many types of industrial products in process of manufacture where relative humidity is the controlling factor.

EXAMPLE 2: Cooling, dehumidifying, humidifying, heating, and ventilating system, requiring refrigeration.

The atmospheric conditions required in the treated area would be as follows:

1. *Summer*—Constant dry-bulb temperature and relative humidity
2. *Winter*—Constant dry-bulb temperature and relative humidity

Basis of Design. Assume the same given factors as in Example 1, except that the summer inside dry-bulb temperature must not exceed 80°F, in other words, constant year-round atmospheric conditions, in the treated area. Since the heat transmission was calculated on a per-degree difference, the same calculations will apply, corrected for the temperature difference between outside and inside conditions (see Example 2 and Fig. 21-19).



HEAT TRANSMISSION CALCULATION

HEATING AND AIR CONDITIONING DEPARTMENT

Prospect No. P-1036
Customer JOHN DOE HOSIERY MILLS
Location GREENSBORO N.C.
Type FULL FASHIONED HOSIERY
Floor 1ST
Room ENTIRE

SEASON	SUMMER		WINTER	
Location	Inside	Outside	Inside	Outside
Dry Bulb	87½	96	80	10
Wet Bulb	76½	72	70	
Dew Point	72		65	
Rel. Humidity	60%		60%	

Sheet No. 1 of
Job No. –
Estimated by J SMITH Date 12/1/46
Checked Date
Remarks

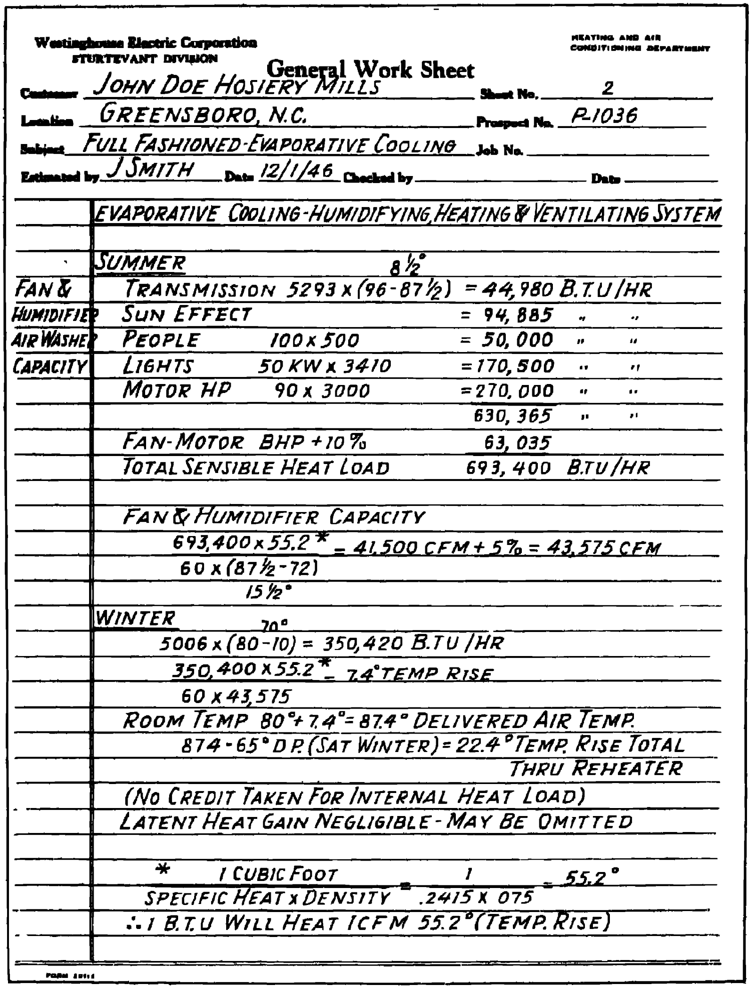
SURFACE			DESCRIPTION		DIMENSIONS			FACTORS			B. T. U. PER HOUR		
Expos.	Dir	Type	Thick	MATERIAL	Width	Height	Net Area in sq Ft.	Surf trans	Winter Expos.	Const. T Diff.	Winter Per 1° Diff.	Summer Per 1° Diff.	CONSTANT
	TYPICAL CALCULATIONS-EVAPORATIVE COOLING-NO REFRIGERATION												
O.S.	N.	WALL	12"	BRICK	145	15	2175	.36	1.15		736	640	
		GLASS BLOCK	11	@6'0"x6'0"=36 Φ	EA		396	.49			192	192	
							1779						
O.S	E	WALL	12"	BRICK	57	15	855	.36			243	243	
		GLASS BLOCK	5	@6'0"x6'0"=36 Φ	EA.		180	.49			88	88	
							675						
I.S	E	WALL	8"	TILE & PLASTER	67	15	1005	.37			–	336	
		DOUBLE GLASS	6"	@4'0"x4'0"=16 Φ	EA		96	.45			–	47	
							909						
O.S	S	WALL	12"	BRICK	134	15	2010	.36		20"	555	555	11100
		GLASS BLOCK	13	@6'0"x6'-0"=36 Φ	EA		468	.49		20"	229	229	4580
							1542						
O.S.	W	WALL	4"	INSL. TEMPORARY STUD. 1" SHEATHING	114	15	1710	.4			554	554	
		DOUBLE GLASS	9	@6'0"x6'-0"=36 Φ	EA		324	.45			146	146	
							1386						
I.S.	↓	FLOOR	6"	CONC ON DIRT		NONE							
O.S.	↑	ROOF	3"	WOOD, 1" CORK, 5 PLY			18,860	.12		35	2263	2263	79205
AREA											5006	5293	94885
145 X 134 =				19430 Φ									
LESS OFFICE 10 X 57 =				570 Φ									
				18860 Φ.									
TOTALS													

Cubic Contents 18860 Φ x 15 = 282,900 §
C.F.M. 43,575 See Sheet No. Air Change 6.5 MIN.

Form 6019

Fig. 21-17. Typical calculations of an evaporative cooling and humidifying system—no refrigeration. Example 1, page 1.

Mechanical refrigeration cools chilled water which is sprayed in the air washer, termed the dehumidifier. Air is drawn through, thereby cooling the leaving air to saturation at the designed dew-point temperature required for the treated space or slightly below to allow for latent-heat gain within the conditioned space. The air volume is determined as before, *i.e.*, figured on the sensible-heat rise between the dry-bulb temperature leaving the dehumidifier and the conditioned-room dry-bulb temperature. To economize on refrigeration, the outside air is calculated on the minimum requirements sufficient to maintain an air pressure within the conditioned space and consistent with proper ventilation standards. The refrigeration load calculations are the sum of the total sensible heat plus the latent-heat



Westinghouse Electric Corporation
STURTEVANT DIVISION

HEATING AND AIR CONDITIONING DEPARTMENT

General Work Sheet

Customer JOHN DOE HOSIERY MILLS **Sheet No.** 2
Location GREENSBORO, N.C. **Prospect No.** P-1036
Subject FULL FASHIONED-EVAPORATIVE COOLING **Job No.**
Estimated by J SMITH **Date** 12/1/46 **Checked by** **Date**

EVAPORATIVE COOLING-HUMIDIFYING, HEATING & VENTILATING SYSTEM

SUMMER

FAN &	TRANSMISSION	5293 x (96 - 87½) 8½°	= 44,980	B.T.U/HR
HUMIDIFIER	SUN EFFECT		= 94,885	" "
AIR WASHER	PEOPLE	100 x 500	= 50,000	" "
CAPACITY	LIGHTS	50 KW x 3410	= 170,500	" "
	MOTOR HP	90 x 3000	= 270,000	" "
			630,365	" "
	FAN-MOTOR BHP + 10%		63,035	
	TOTAL SENSIBLE HEAT LOAD		693,400	B.TU/HR

FAN & HUMIDIFIER CAPACITY

$$\frac{693{,}400 \times 55.2^{*}}{60 \times (87\tfrac{1}{2} - 72)} = 41{,}500 \text{ CFM} + 5\% = 43{,}575 \text{ CFM}$$

15½°

WINTER

70°

5006 x (80 - 10) = 350,420 B.TU/HR

$$\frac{350{,}400 \times 55.2^{*}}{60 \times 43{,}575} = 7.4^{\circ} \text{ TEMP RISE}$$

ROOM TEMP 80° + 7.4° = 87.4° DELIVERED AIR TEMP.

874 - 65° D.P. (SAT WINTER) = 22.4° TEMP. RISE TOTAL THRU REHEATER

(NO CREDIT TAKEN FOR INTERNAL HEAT LOAD)

LATENT HEAT GAIN NEGLIGIBLE - MAY BE OMITTED

$$^{*}\ \frac{1 \text{ CUBIC FOOT}}{\text{SPECIFIC HEAT} \times \text{DENSITY}} = \frac{1}{.2415 \times 075} = 55.2^{\circ}$$

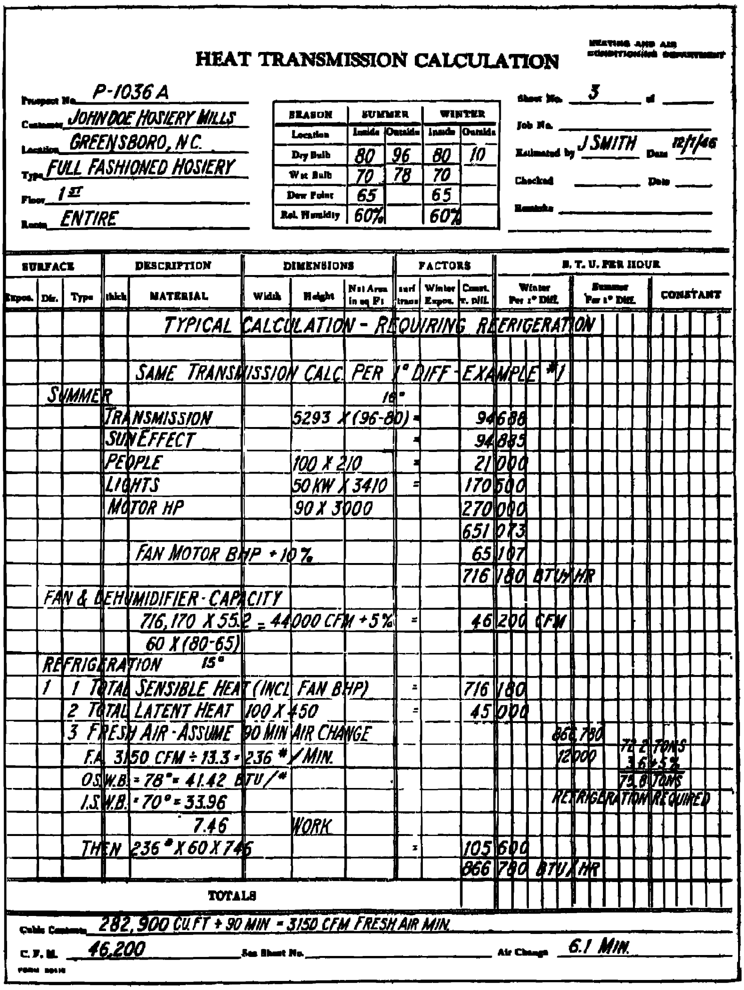
∴ 1 B.T.U WILL HEAT 1 CFM 55.2° (TEMP. RISE)

FORM [illegible]

FIG. 21-18. Typical calculations of an evaporative cooling and humidifying system—no refrigeration. Example 1, page 2.

load and the heat of the outside air from the outside wet-bulb temperature to the inside wet-bulb temperature. When the latent-heat load within the treated areas is high in proportion to the sensible-heat load, correction in the dew-point temperature leaving the dehumidifier must be calculated to compensate for the moisture gain within the conditioned space.

The above methods of calculation are applicable to many types of industrial-process products of manufacture, where both the dry-bulb temperature and relative humidity are the controlling factors. When the characteristic properties of the process materials require control of dry-bulb temperature and relative humidity within limiting variations, direct-expansion or chilled-water cooling coils may be used in place of the spray-type dehumidifier.



HEAT TRANSMISSION CALCULATION

HEATING AND AIR CONDITIONING DEPARTMENT

Prospect No. P-1036 A
Customer JOHNDOE HOSIERY MILLS
Location GREENSBORO, N.C.
Type FULL FASHIONED HOSIERY
Floor 1ST
Room ENTIRE

SEASON	SUMMER		WINTER	
Location	Inside	Outside	Inside	Outside
Dry Bulb	80	96	80	10
Wet Bulb	70	78	70	
Dew Point	65		65	
Rel. Humidity	60%		60%	

Sheet No. 3 of
Job No.
Estimated by J SMITH Date 12/1/46
Checked Date
Remarks

SURFACE			DESCRIPTION		DIMENSIONS			FACTORS			B. T. U. PER HOUR		
Expos.	Dir.	Type	thick	MATERIAL	Width	Height	Net Area in sq Ft	surf trans	Winter Expos.	Const. r. diff.	Winter Per 1° Diff.	Summer Per 1° Diff.	CONSTANT

TYPICAL CALCULATION – REQUIRING REFRIGERATION

SAME TRANSMISSION CALC. PER 1° DIFF – EXAMPLE #1
SUMMER 16°
TRANSMISSION 5293 X (96-80) = 94 688
SUN EFFECT = 94 885
PEOPLE 100 X 210 = 21 000
LIGHTS 50 KW X 3410 = 170 500
MOTOR HP 90 X 3000 270 000
651 073
FAN MOTOR BHP + 10% 65 107
716 180 BTU/HR
FAN & DEHUMIDIFIER - CAPACITY
716,170 X 55.2 = 44000 CFM + 5% = 46 200 CFM
60 X (80-65)
REFRIGERATION 15°
1 1 TOTAL SENSIBLE HEAT (INCL FAN BHP) = 716 180
2 TOTAL LATENT HEAT 100 X 450 = 45 000
3 FRESH AIR - ASSUME 90 MIN AIR CHANGE
F.A. 3150 CFM ÷ 13.3 = 236 #/MIN.
O.S.W.B. = 78° = 41.42 BTU/#
I.S.W.B. = 70° = 33.96
7.46 WORK
THEN 236 # X 60 X 7.46 = 105 600
866 780 BTU/HR

866,780 / 12000 72.2 TONS
3.6 + 5%
75.8 TONS
REFRIGERATION REQUIRED

TOTALS

Cubic Contents 282,900 CU.FT ÷ 90 MIN = 3150 CFM FRESH AIR MIN.
C.F.M. 46,200 See Sheet No. Air Change 6.1 MIN.
FORM [illegible]

Fig. 21-19. Typical calculations of a cooling, humidifying, and dehumidifying system with refrigeration. Example 2.

Design of Supply Ducts. To apply fans correctly and economically to process air-conditioning systems, it is necessary to understand what governs system losses, since the total system resistance solely determines the point at which a fan will operate on its characteristic curve. The system losses in general are caused by two distinct factors—frictional losses from viscosity action of the air and losses from accelerations and decelerations of the air stream. The trade commonly uses various terms to specify system resistance, total-pressure loss, and static-pressure loss. Fan performances are selected from performance curves or tables on fan total pressure or fan static pressure, the latter being the most common. The procedure to determine the static pressure of a system is as follows:

1. Sum up all component total-pressure losses of the parts of the system, including the fresh-air inlet, apparatus, and equipment, losses in ductwork, and velocity pressure at the discharge outlets of the system.
2. Deduct the fan-outlet velocity pressure (determine from a preliminary fan-size selection).
3. Pick a fan for the resulting static pressure.

The system design in practical application is to keep the total static pressure to a minimum (approximately 1 to 1½ in. static pressure), thereby economizing on operating-horsepower requirements. The frictional-resistance losses through apparatus and equipment are generally given in catalogues and published literature of equipment manufacturers; however, the pressure losses in supply ducts must be calculated.

Pressure Losses in Supply Ducts. The conventional engineering procedure in calculating duct friction loss is based on "equivalent round-duct pipe sizes" and a loss, equal to one velocity pressure, in a duct length equal to a stated number of pipe diameters. In practice, the number of pipe diameters for the loss of one velocity pressure varies from 40 to 70, depending upon the caution and experience of the individual making the calculations. There is little doubt that the figure 40 is exceedingly safe and that the allowance of one velocity pressure loss in 70 diameters is sufficient for a duct of good commercial workmanship. This explanation means that the losses in ducts are calculated in terms of velocity pressure of flow (in inches of water). Thus a friction loss in pressure in 70 diameters of length of duct is equivalent to the velocity pressure in inches of water.

This procedure assumes that the friction loss varies as the square of the velocity, and the standard table values for equivalent round ducts are based on this assumption. Experiments have proved that this assumption is too high and may safely be reduced. Having determined the pressure loss, tables may be prepared on a basis of "equalization for constant friction," which means the friction pressure loss in inches of water (gauge) per 100 ft of duct remains a constant. The layout and design of a duct system requires that the over-all duct resistance be kept as low as practicable within reasonable limits. In practical application, this method enables the engineer to select his total duct resistance and design the system to operate within those limits.

The formulas for round ducts have been thoroughly tested and may be used with safety. The whole concept of transferring these results to "rectangular-duct-size equivalent" is a fallacy unless used with reasoned caution. There are no engineering data that give the length (one velocity pressure taken for round ducts as 70 diameters) and the equivalent length for rectangular ducts since the cross section and perimeter may vary. Therefore, while the physicist labors for greater exactness, the engineer must design with the material at hand and use sound judgment.

Good engineering practice suggests procedure as follows:

1. Table 21-14 may be used, assuming the finest sheet-metal workmanship.
2. For rectangular ducts, the aspect ratio (*i.e.*, the long side to the short side) should not be greater than 1½ times the equivalent round-duct size.

EXAMPLE: In the use of Table 21-14, assume that the supply-duct design was limited to 0.4 in. static pressure and that the longest run of duct was 200 ft. Then the equalization for constant friction for 100 ft would be one-half 0.4 in. static pressure, or 0.2 in. static pressure. Allowing caution for workmanship, refer to the column headed 0.18 in. static-pressure loss. This column gives the carrying capacity in cubic feet per minute of air for round ducts for diameter shown on the right. The corresponding "rectangular equivalent" ducts, having the same pressure loss per unit length when carrying the same volume, cubic feet per minute, and having the same density, are shown in the columns on the right.

The above conventional procedure, therefore, has very great virtues of consistency, simplicity, and practicability in sizing of duct systems, and experience has proved the accuracy of this method.

TYPES OF PROCESS AIR-CONDITIONING SYSTEMS

The fundamental factors governing the type or selection of apparatus depend upon the characteristic performance of the apparatus and equipment best suited to meet the psychrometric characteristics of the process material. Therefore, a knowledge of the assembled arrangement of various types of equipment is essential. The general classifications of types of air-conditioning systems are "unitary" and "central station." Since the functions of unitary and central-station systems are the same, the dividing line is somewhat arbitrary. In general, however, a unitary system means factory-built and -assembled equipment, shipped complete as a unit to perform some specified function, *e.g.*, "humidifying unit." The central-station type of system may be defined as factory-built functional apparatus, assembled, erected, and fabricated in the field in a large plant centrally located to perform some specified functions.

Unitary System

Unitary apparatus is designed for horizontal- or vertical-assembly arrangement, for ceiling or floor mounting, the capacity being governed by the practical size, within reasonable limits of dimensions, weight, and convenience of handling and shipping. Generally, in process air conditioning, the unitary system is applicable to the smaller installations, as the name "unit" implies.

Central-station Systems

Central-station systems are classified in accordance with the function performed, *i.e.*, evaporative cooling, cooling, humidifying, dehumidifying, heating, and ventilating. The comparative effectiveness of the type of system selected depends upon the sensitivity of the process material to atmospheric conditions. Some process materials require close control of atmospheric conditions throughout the year, whereas other materials may permit variations of either dry-bulb temperature or relative humidity within stipulated limits. The general types of equipment for central-station systems may be modified to effect greater flexibility, improve performance, provide accurate control of conditions, or effect greater economy of operation without affecting the fundamental basic functions. Some types of equipment are applicable to all types of systems, whereas some equipment has a limited application.

The most commonly encountered types of central-station systems for process air conditioning are outlined in the following paragraphs.

Systems Not Requiring Refrigeration. For systems not requiring refrigeration, use an evaporative cooling system, the functions of which are cooling by evaporation, humidifying, heating, and ventilating. The atmospheric conditions in the treated area would be as follows:

1. *Summer*—Constant relative humidity, varying dry-bulb temperature, cooling by evaporation, ventilation
2. *Winter*—Constant dry-bulb temperature and constant relative humidity, heating, and ventilating

Systems Requiring Refrigeration with Spray-type Dehumidifier. For systems requiring refrigeration with a spray-type dehumidifier, use a dehumidifying cooling system, the functions of which are cooling, dehumidifying, humidifying, heating, and ventilating. The atmospheric conditions in the treated area would be as follows:

1. *Summer*—Constant dry-bulb temperature and constant relative humidity, cooling by chilled water, refrigeration, and ventilation
2. *Winter*—Constant dry-bulb temperature and constant relative humidity, heating, and ventilating

System Requiring Refrigeration with Direct-expansion or Chilled-water Cooling Coils. For a system requiring refrigeration with direct-expansion or chilled-water cooling coils, use a surface dehumidifier, the functions of which are cooling, dehumidifying, heating, and ventilating. The atmospheric conditions in the treated area would be as follows:

1. *Summer*—Slight variations of dry-bulb temperature and relative humidity within reasonable limits; cooling by direct-expansion or chilled-water cooling coils, refrigeration, and ventilation
2. *Winter*—Constant dry-bulb temperature, with special provision for humidifying when required, heating, and ventilating

Table 21-14. Air Ducts Equalized for Constant Pressure

Friction loss per 100 ft pipe, in. of water										Diam., in.	Equivalent rectangular duct, in.																														
.40	.35	.30	.25	.20	.18	16	14	12	.10		4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34
Cu ft of air per min																																									
85	78	72	65	57	54	50	47	43	39	4 0																															
120	109	102	93	82	77	72	67	61	56	4 5																															
155	141	132	119	106	98	92	86	79	71	5 0																															
205	185	175	159	140	131	124	116	105	95	5 5	6	5	4																												
250	230	210	192	170	160	150	139	128	115	6 0	8	6	5	4																											
320	295	274	245	218	205	193	179	162	148	6 5	9	7	6	5																											
375	350	325	290	255	240	225	210	195	175	7 0	11	8	7	6	5																										
465	430	400	365	320	300	280	260	239	215	7 5	13	10	8	7	6																										
540	500	460	420	370	348	325	300	278	250	8 0	14	11	9	8	7	6																									
655	600	560	500	450	420	390	365	330	300	8 5	17	13	10	9	8	7																									
750	685	630	575	500	475	450	420	375	345	9 0	19	14	12	10	8		7																								
875	810	750	675	590	560	525	485	450	400	9 5	21	16	13	11	9	8																									
970	900	840	750	660	625	585	545	500	450	10 0	24	18	14	12	11	9	8	8																							
1,125	1,050	960	870	765	720	675	625	575	525	10 5	27	20	16	13	12	10	9																								
1,250	1,150	1,070	960	850	800	750	700	640	575	11 0	30	22	18	15	13	11	10	9																							
1,450	1,325	1,240	1,100	970	920	860	800	740	660	11 5	33	25	19	16	14	12	11	10	9																						
1,600	1,460	1,350	1,240	1,070	1,000	950	880	810	730	12 0	37	27	21	18	15	13	12	11	10	9																					
1,800	1,660	1,530	1,390	1,240	1,150	1,080	990	915	830	12 5	40	30	24	20	17	15	13	12	11	10																					
1,950	1,810	1,680	1,500	1 340	1,280	1,190	1,100	995	900	13 0	44	33	26	21	18	16	14	13	12	11	10																				
2,200	2,000	1,880	1,700	1,500	1,400	1,330	1,220	1,110	1,000	13 5	48	36	28	23	20	17	15	14	13	12	11																				
2,400	2,200	2,000	1,830	1,600	1,530	1,420	1,320	1,220	1 100	14 0		39	31	25	21	19	17	15				11																			
2,650	2,450	2,260	2,050	1,800	1,700	1,600	1,490	1,360	1,220	14 5		42	33	27	23	20	18	16	15	13	12	12																			
2,850	2,620	2,475	2,220	1,980	1,800	1,720	1,600	1,470	1,320	15 0		46	36	29	25	21	19	17	16	14	13		12																		
3,200	2,900	2,780	2,510	2,220	2,030	1,930	1,800	1,650	1,440	15 5		49	39	32	27	23	20	18	17	15	14	13																			
3,400	3,050	2,900	2,600	2,350	2,150	2,050	1,910	1,760	1,580	16 0			42	34	29	25	22	20	18	16	15	14	13																		
3,750	3,500	3,290	2,940	2,600	2,400	2,200	2,050	1,880	1,690	16 5			45	36	31	26	23	21	19	17	16	15	14	13																	
4,000	3,700	3,480	3,140	2,790	2,530	2,400	2,230	2,050	1,840	17 0			48	39	33	28	25	22	20	19	17	16	15	14	13																
4,400	4,100	3,820	3 450	3,050	2,800	2,650	2,460	2,260	2,040	17 5				42	35	30	27	24	21	20	18	17	16	15	14																
4,650	4,300	3,980	3,640	3,250	3,080	2,800	2,600	2,400	2,150	18 0				44	37	32	28	25	23	21	19	18	17	16	15	14															
5,100	4,700	4,440	4,000	3,540	3,250	3,070	2,850	2 600	2,350	18 5				47	40	34	30	27	24	22	20	19	18	17	16	15															
5,400	5,000	4,700	4,240	3,740	3,420	3,250	3,030	2,780	2,500	19 0					42	36	32	28	26	23	22	20	19	18	17	16	15														
5,850	5,400	5,080	4,550	4,000	3,750	3,530	3,290	3,000	2,700	19 5					45	39	34	30	27	25	23	21	20				16														
6,200	5,700	5,300	4,860	4,300	3,920	3,740	3,480	3,180	2,860	20 0					48	41	36	32	29	26	24	22	21	20	18	17	17	16													
6,650	6,150	5,780	5,220	4,500	4,250	4,000	3,720	3,400	3,070	20 5						43	38	33	30	27	25	23	22		19	18															
7,000	6,500	6,080	5,490	4,850	4,500	4,220	3,930	3,600	3,250	21 0						46	40	35	32	29	27	25	23	22	20	19	18	17	17												
7,500	7,000	6,600	5,960	5,280	4,810	4,520	4,200	3,850	3,500	21 5						48	42	37	34	30	28	26	24		21	20	19	18													
8,000	7,400	6,950	6,280	5,650	5,060	4,750	4,420	4,040	3,650	22 0						51	44	39	35	32	29	27	25	24	22	21	20	19	18	17											
8,600	8,000	7,470	6,750	5,950	5,450	5,100	4,750	4,340	3,950	22 5						54	47	41	37	34	31	28	26	25	23	22	21	20	19	18											

Table 21-14. Air Ducts Equalized for Constant Pressure (*Cont.*)

Friction loss per 100 ft pipe, in of water										Diam, in	Equivalent rectangular duct, in.																														
.40	.35	30	25	.20	18	16	14	12	10		4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34
Cu ft of air per min																																									
9,100	8,500	7,880	7,190	6,430	5,750	5,480	5,000	4,650	4,200	23.0							49	44	39	35	32	30	28	26	24	23	22	21	20	19	18										
9,650	9,000	8,390	7,500	6,700	6,100	5,800	5,400	4 940	4,450	23.5							52	46	41	37	34	31	29	27	25	24	23	22	21	20	19										
10,200	9,500	8,750	8,000	7,080	6,500	6,130	5,700	5,200	4,700	24.0							54	48	43	39	35	33	30	28	27	25	24	23	22	21	20	19									
10,800	9,800	9,380	8,460	7,500	6,850	6,500	6,050	5,500	5,000	24.5							57	50	45	41	37	34	32	30	28	26	25	24			21	20									
11,200	10,300	9,730	8,790	7,760	7,090	6,750	6,280	5,750	5,190	25.0							60	52	47	42	39	36	33	31	29	27	26	25	23	22		21	20								
11,800	11,000	10,250	9,250	8,200	7,470	7,100	6,600	6,040	5,450	25.5								55	49	44	40	37	35	32	30	28	27	26	24	23	22		21								
12,500	11,500	10,850	9,800	8,600	8,000	7,500	7,000	6,380	5,750	26.0								57	51	46	42	39	36	34	32	30	28		25	24	23	22									
13,000	12,000	11,300	10,200	9,075	8,230	7,850	7,300	6,680	6,000	26.5								60	54	48	44	41	37	35	33	31	29	28	26	25	24	23	22	21							
13,500	12,500	11,700	10,600	9,420	8,550	8,150	7,580	7,000	6,250	27.0								62	56	50	46	42	39	37	34	32	30	29	27	26	25	24	23	22							
											14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44
14,500	13,400	12,600	11,400	10,100	9,170	8,720	8,100	7,400	6,700	27.5	48	44	41	38	36	33	32	30	28	27	26	25	24	23	22		21														
15,000	14,000	13,050	11,555	10,500	9,600	9,010	8,400	7,680	7,000	28.0	50	46	42	39	37	35	33	31	30	28	27	26	25	24	23	22	22														
16,000	14,600	13,900	12,550	11,180	10,250	9,620	8,950	8,190	7,380	28.5	52	48	44	41	38	36	34	32	31	29	28	27	26	25	24	23															
16,900	15,500	14,680	13,200	11,800	10,700	10,200	9,500	8,550	7,800	29.0	54	50	46	43	40	37	35	34	32	31	29	28	27	26	25	24	23														
17,500	16,000	15,200	13,700	12,400	11,100	10,550	9,800	8,950	8,100	29.5	56	52	48	44	41	39	37	35	33		30	29	29	27	26	25	24														
18,100	16,800	15,720	14,200	12,620	11,450	10,900	10,100	9,250	8,350	30.0	58	54	49	46	43	40	38	36	34	33	31	30	29			26	25														
19,000	17,700	16,500	14,900	13,250	12,000	11,450	10,650	9,710	8,780	30.5	60	56	51	47	45	42	39	37	35	34	32	31		28	27																
20,000	18,500	17,000	15,700	13,960	12,650	12,000	11,200	10,200	9,200	31.0	63	58	53	49	46	43	41	39	37	35	33	32	31	29	28	27	26	25	25	24		23	22	22							
21,000	19,300	18,250	16,500	14,650	13,300	12,600	11,800	10,700	9,650	31.5	65	60	55	51	48	45	42	40	38	36	34	33		30	29	28	27			25	24		23								
21,900	20,100	19,000	17,200	15,280	13,850	13,200	12,300	11,000	10,100	32.0	68	62	57	53	49	46	44	41	39	37	36	34	33	31	30	29	28	27	26	26	25	24	24	23		22					
22,500	21,000	19,500	17,650	15,700	14,550	13,600	12,700	11,600	10,400	32.5	70	64	59	55	51	48	45	43	40	38	37	35	34	32	31	30	29	28	27		26	25		24	23						
23,000	22,000	20,000	18,100	16,000	15,000	14,000	13,000	12,000	10,760	33.0		66	61	57	53	49	47	44	42	40	38	36	35	33	32	31	30	29	28	27		26	25		24	23	23				
25,000	23,000	21,700	19,600	17,205	15,800	15,000	14,000	12,800	11,500	33.5		69	63	59	55	51	48	45	43	41	39	37	36	34	33	32	31	30	29	28	27		26	25		24					
25,600	24,000	22,000	20,200	18,000	16,300	15,600	14,500	13,200	12,000	34.0		71	65	61	56	53	50	47	45	42	40	39	37	36	34	33	32	31	30	29	28	27	27	26	25	26	24				
26,900	25,000	23,350	21,100	18,780	17,000	16,200	15,100	13,800	12,400	34.5		74	67	63	58	54	51	49	46	44	42	40	38		35	34	33	32	31	30	29	28			26		25	24			
27,900	25,900	24,200	21,900	19,000	17,650	16,900	15,750	14,300	13,000	35.0			70	65	60	56	53	50	47	45	43	41	39	38	36	35	34	33	32	31	30	29	28	27	27	26		25	25		
28,700	26,500	24,900	22,500	20 000	18,150	17,300	16,150	14,700	13,300	35.5			72	67	62	58	55	52	49	46	44	42	40	39	37	36	35					30	29	28		27	26				
29,500	27,100	25,300	23,100	20,600	19,000	17,800	16,600	15,100	13,650	36.0			74	69	64	60	56	53	50	48	46	44	42	40	38	37	36	34	33	32	31		30	29	28		27	26	26		
31,000	28,100	26,900	24,300	21,600	19,600	18,650	17,400	15,850	14,300	36.5			77	71	66	62	58	55	52	49	47	45	43	41	40	38	37	35	34	33	32	31		30	29	28	28	27		26	
32,000	29,500	27,000	25,100	22,580	20,550	19,300	18,000	16,200	14,800	37.0			79	73	68	64	60	57	54	50	48	46	44	42	41	39	38	37	35	34	33	32	31	31	30	29		28	27		26
34,000	31,000	29,500	26,700	23,700	21,500	20,500	19,200	17,400	15,700	37.5			82	76	70	66	61	58	55	52	50	47	45	44	42	40	39		36	35	34	33	32			30	29		28	27	27
35,500	32,100	30,800	27,800	24,800	22,000	21,400	20,000	18,200	16,400	38.0				78	72	68	63	60	57	54	51	49	47	45	43	41	40	39	37	36	35	34	33	32	31	31	30	29		28	
36,200	33,000	31,400	28,400	25,000	22,900	21,800	20,400	18,600	16,800	38.5				80	75	70	65	61	58	56	53	50	48	46	44	43	41	40	38	37	36	35	34	33	32		31	30	29	29	28
37,000	34,000	32,200	29,200	25,800	23,400	22,300	20,800	19,000	17,000	39.0				83	77	72	67	63	60	57	54	52	49	47	45	44	42	41	39	38	37	36	35	34	33	32		31	30		
38,000	35,500	33,000	29,800	26,600	24,000	22,900	21,400	19,400	17,600	39.5				85	79	74	69	65	62	58	56	53	50	49	47	45	43	42	40	39	38	37	36	35	34	33	32		31	30	

Table 21-14. Air Ducts Equalized for Constant Pressure (*Cont.*)

Friction loss per 100 ft pipe, in. of water										Diam, in.	Equivalent rectangular duct, in.																														
.40	.35	.30	.25	20	18	16	.14	.12	10		14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44
Cu ft of air per min																																									
39,000	37,000	33,900	30,600	27,200	24,650	23,500	22,000	20,000	18,000	40.0					81	76	71	67	64	60	57	55	52	50	48	46	44	43	42	40	39	38	37	36	35	34	33	32	32	31	30
40,100	38,000	34,800	31,500	27,900	25,400	24,200	22,600	20,500	18,550	40.5					81	78	73	69	65	62	59	56	53	51	49	47	46	44		41	40	39				35	34	33			31
41,500	39,000	36,000	32,500	28,400	26,200	25,000	23,400	21,300	19,200	41.0					86	80	75	71	67	63	60	57	55	52	50	49	47	45	44	42	41	40	38	37	36		35	34	33	32	32
43,500	40,000	37,800	34,100	30,100	27,500	26,200	24,500	22,300	20,000	41.5					89	83	77	73	69	65	62	59	56	54	52	50	48	46	45	43	42	41	39	38	37	36		35	34	33	
45,000	42,000	39,100	35,000	31,200	28,400	27,000	25,100	23,000	20,750	42.0					91	85	79	75	70	67	64	61	58	55	53	51	49	47	46	44	43	42	40	39	38	37	36		35	34	33
46,500	43,000	40,400	36,400	32,200	29,400	28,000	26,250	23,800	21,500	42.5						87	82	77	72	69	65	62	59	56	54	52	50	49	47	45	44	43	41	40	39	38	37	36		35	34
47,500	44,000	41,300	37,200	32,700	30,000	28,600	26,900	24,400	22,000	43.0						89	84	79	74	71	67	64	61	59	56	53	52	50	48	47	46	44	42	41	40	39	38	37	36		35
49,000	45,500	42,500	38,400	33,900	31,000	29,500	27,600	25,100	22,600	43.5						92	86	81	76	72	69	65	62	59	57	55	53	51	49	48	46	45	43	42	41	40	39	38	37	36	
50,500	47,000	43,000	39,700	35,000	32,000	30,500	28,500	26,000	23,400	44.0						94	88	83	78	74	70	67	63	61	58	56	54	52	50	49	47	46	44	43	42	41	40	39	38	37	36
52,500	48,500	45,500	41,200	36,300	33,200	31,500	29,500	26,800	24,200	44.5						97	91	85	80	76	72	69	65	63	60	58	56	53	52	50	48	47	45	44	43	42	41	40	39	38	37
54,000	50,000	47,000	42,400	37,400	34,200	32,500	30,000	27,600	25,000	45.0							93	87	82	78	74	70	67	64	61	59	57	55	53	51	50	48	47	45	44	43	42	41	40	39	38
55,100	52,000	47,900	43,200	38,100	34,900	33,300	31,000	28,400	25,600	45.5							95	90	84	80	76	72	69	66	63	61	58	56	54	52	51	50	48	46	45	44	43	42	41	40	39
57,000	53,000	49,500	44,600	39,400	36,000	34,300	31,900	29,000	26,300	46.0							97	92	87	82	78	74	70	67	64	62	59	57	55	54	52	51	49	47	46	45	44	43		41	40
58,000	54,500	50,400	45,500	40,100	36,700	35,000	32,500	29,800	26,800	46.5							100	94	89	84	80	76	72	69	66	63	61	59	57	55	53	52	50	48	47	46	45		42		
60,100	56,000	52,200	47,100	41,500	38,000	36,300	33,750	30,900	27,800	47.0								96	91	87	82	78	74	70	67	65	62	60	58	56	54	53	51	49	48	47	46	44	43	42	41
62,000	57,500	53,900	48,500	42,900	39,200	37,400	34,750	31,800	28,600	47.5								99	93	88	84	79	76	72	69	66	64	61	59	57	55	54	52	51	49	48	47	45	44	43	42
64,000	59,000	55,500	50,200	44,300	40,500	38,500	35,750	32,500	29,500	48.0								101	95	90	86	81	78	74	71	68	65	63	61	58	56	55	53	52	50	49	48	46	45	44	43
65,500	60,100	56,900	51,000	45,400	41,500	39,500	36,500	33,500	30,400	48.5								104	97	92	88	83	79	75	72	69	67	64	62	60	58	56	54	53	51	50	49	47	46	45	44
67,500	63,000	58,600	53,000	46,700	42,600	40,600	37,800	34,500	31,200	49.0								106	100	94	90	85	81	77	74	71	68	66	63	61	59	57	55	54	52	51	50	48	47	46	45
69,000	64,500	60,000	54,100	47,700	43,600	41,500	38,500	35,300	31,850	49.5									102	96	92	87	83	79	76	72	70	67	65	63	60	59	56	55	54	52	51	49	48	47	46
70,500	65,500	61,300	55,300	48,700	44,300	42,400	39,400	36,000	32,500	50.0									105	98	94	89	85	81	77	74	72	68	66	64	61	60	58	56	55	53	52	50		48	47
72,500	67,500	63,000	56,800	50,200	45,800	43,500	40,500	37,000	33,400	50.5									107	102	96	91	87	83	79	76	73	70	68	66	63	61	59	57	56	54	53	51	50	49	48
											26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56
75,000	69,000	65,100	57,500	51,000	47,500	45,000	41,900	38,250	34,500	51.0	89	85	81	77	74	71	69	67	64	62	61	59	57	55	54	52	51	50	49	48	47	46	45	44	43	42	41		40	39	
77,000	70,100	66,900	60,000	53,300	48,700	46,500	43,300	39,500	35,600	51.5	90	86	82	79	76	73	70	68	66	64	62	60	58	57	55	53	52	51	50	49	48		46	45	44	43	42	41		40	39
79,000	72,500	68,600	61,500	54,600	50,000	47,500	44,100	40,400	36,500	52.0	92	88	84	81	78	75	72	69	67	65	63	61	59	58	56	55	53	52	51	50		47			45	44	43	42	41	41	40
80,000	75,000	69,530	62,400	55,400	50,500	48,100	45,000	41,000	37,000	52.5	95	90	86	82	79	76	74	70	68	66	64	62	60	59	57	56	54	53	52	51	49	48	47	46		45	44	43	42		41
82,500	76,000	71,500	64,400	57,000	52,200	49,600	46,250	42,200	38,000	53.0	97	92	88	84	81	78	75	72	70	68	66	64	62	60	58	57	55	54	53		50	49	48	47	46			44	43	42	
85,000	78,500	73,800	66,300	58,800	53,800	51,000	47,500	43,400	39,000	53.5	99	94	90	86	82	80	76	74	72	69	67	65	63	61	59	58	56	55	54	53	51	50	49	48	47	46	45	45	44	43	42

Table 21-14. Air Ducts Equalized for Constant Pressure (*Cont*)

Friction loss per 100 ft pipe, in. of water										Diam, in	Equivalent rectangular duct, in.																														
40	.35	.30	.25	20	18	16	14	.12	.10		26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56
Cu ft of air per min																																									
87,000	80,000	75,000	67,800	60,200	55,000	52,500	48,900	45,000	40,250	54.0	101	96	92	88	84	81	78	75	73	70	68	66	64	62	61	59	58	56	55		52	51	50	49	48	47	46			44	43
90,000	82,500	78,100	70,100	62,200	57,000	54,000	50,100	46,000	41,500	54.5	103	98	94	90	86	83	80	77	74	72	70	67	65	64	62	60	59	57	56	55	53	52	51	50	49	48	47	46	45	45	44
92,000	85,000	80,000	71,700	63,500	57,500	55,200	51,500	47,000	42,500	55.0	105	100	96	92	88	84	82	78	76	73	71	69	67	65	63	61	60	58	57	56	54	53	52	51	50	49	48	47	46		45
95,000	87,500	82,500	74,000	65,700	60,000	57,000	53,000	48,500	43,750	55.5	107	102	98	93	90	86	83	80	77	75	72	70	68	66	64	63	61	59	58	57	55	54	53	52	51	50	49	48	47	46	
97,000	89,000	84,300	75,500	67,100	61,400	58,500	54,400	49,600	45,000	56.0	110	104	100	95	91	88	85	82	79	76	74	71	69	67	66	64	62	60	59	58	56	55	54	53	52	51	50	49	48	47	46
98,000	91,000	85,000	76,400	67,800	62,500	59,000	55,000	50,200	45,400	56.5	112	107	102	97	93	90	86	84	80	78	75	73	71	69	67	65	63	62	60	59	57	56	55	54	53	52	51	50	49	48	47
100,000	92,000	86,500	78,000	69,200	65,000	60,000	56,000	51,000	46,000	57.0	114	109	104	99	95	91	88	85	82	79	77	74	73	70	68	66	64	63	61	60	58	57	56	55	54				50	49	48
103,000	95,000	89,500	80,300	71,300	65,200	62,000	57,900	52,800	47,500	57.5	116	111	106	101	97	93	90	86	83	81	78	76	74	71	69	67	66	64	62	61	59	58	57	56	55	53	52	51		50	49
105,000	96,500	91,200	81,800	72,600	66,500	63,000	58,800	53,600	48,500	58.0	119	114	108	103	99	95	91	88	85	82	79	77	75	72	71	69	67	65	64	62	60	59	58	57		54	53	52	51		
107,000	98,000	93,000	83,500	74,000	67,800	64,500	60,000	55,000	49,500	58.5	121	115	110	105	101	97	93	90	86	84	81	78	76	74	72	70	68	66	65	63	62	60	59	58	56	55	54	53	52	51	50
110,000	100,000	95,500	85,700	76,000	70,000	66,000	61,200	56,000	50,600	59.0		118	112	107	103	99	95	91	88	85	83	80	77	75	73	71	69	68	66	64	63	61	60	59	57	56	55	54	53	52	51
112,000	104,000	97,300	87,300	77,500	70,900	67,500	62,500	57,500	51,800	59.5		120	114	109	105	101	97	93	90	87	84	81	79	77	74	72	70	69	67	65	64	62	61	60	58	57	56	55	54	53	52
116,000	107,000	101,000	90,500	80,300	73,500	70,000	65,000	59,500	53,800	60.0			117	111	107	103	99	95	92	88	86	83	80	78	76	74	72	70	68	67	65	64	62	61	59	58	57	56	55	54	53
118,000	108,000	102,500	92,000	81,600	74,600	71,000	66,000	60,500	54,500	60.5			119	114	109	104	100	97	93	90	87	84	82	79	77	75	73	71	69	68	66	65	63	62	60	59	58	57	56	55	54
120,000	110,000	104,100	93,500	83,000	76,500	72,500	67,500	61,800	55,500	61.0			121	116	111	106	102	98	95	92	89	86	83	81	79	76	74	72	71	69	67	66	64	63	62	60	59	58	57	56	55
123,000	112,500	107,000	96,000	85,000	77,800	74,000	69,000	63,000	56,900	61.5				118	113	108	104	100	97	93	90	87	85	82	80	78	76	74	72	70	68	67	65	64	63	61	60	59	58	57	56
126,000	116,000	110,000	98,000	87,000	80,000	76,000	70,800	64,800	58,100	62.0				120	115	110	106	102	98	95	92	89	86	84	81	79	77	75	73	71	70	68	66	65	64	62	61	60	59	58	57
128,500	118,000	111,600	100,000	88,500	81,300	77,500	72,000	66,000	59,300	62.5					117	112	108	104	100	97	93	90	88	85	83	80	78	76	74	72	71	69	67	66	65	63	62	61	60		
130,000	120,000	113,000	101,000	90,000	82,400	78,200	72,600	66,700	60,000	63.0					119	114	110	106	102	98	95	92	89	87	84	82	80	77	76	74	72	70	69	67	66	64	63	62	61	60	58
134,500	122,000	116,200	104,800	92,500	84,800	81,000	75,200	69,000	62,000	63.5					121	116	112	108	104	100	97	94	91	88	86	83	81	79	77	75	73	71	70	68	67	66	64	63	62		59
137,000	126,000	120,000	106,800	94,500	87,400	82,500	76,800	70,000	63,100	64.0						119	114	110	106	102	98	95	92	89	87	85	82	80	78	76	74	73	71	69	68	66	65	64	63	61	60
139,500	129,000	121,000	108,600	96,100	88,350	84,000	78,000	71,300	64,300	64.5						121	116	112	108	104	101	97	94	91	89	86	84	81	79	77	76	74	72	70	69	67	66	65	64	62	61
142,000	131,000	123,000	110,800	98,200	90,000	85,100	79,200	72,500	65,500	65.0							118	113	109	105	102	99	96	93	90	87	85	83	81	79	77	75	73	72	70	69	67	66	65	63	62
145,000	135,000	126,000	113,000	100,000	91,000	87,400	81,100	74,000	67,000	65.5							120	115	111	107	104	101	97	94	92	89	86	84	82	80	78	76	74	73	71	70	68	67	66	64	63
148,000	138,000	128,600	115,300	101,000	95,000	89,000	82,900	75,600	68,100	66.0								117	113	109	105	102	99	96	93	90	88	85	83	81	79	77	76	74	72	71	69	68	67	65	64
152,000	140,000	132,000	118,500	105,000	96,100	91,500	85,000	77,800	70,000	66.5								119	115	111	107	104	100	97	94	92	90	87	85	83	80	79	77	75	74	72	70	69	68	66	65
155,000	143,000	134,800	121,000	107,100	98,000	93,500	87,000	79,200	71,600	67.0								121	117	113	109	105	102	99	96	93	91	88	86	84	82	80	78	76	75	73	72	70	69	67	66
158,500	145,000	137,000	123,500	109,600	100,000	95,100	88,800	81,000	73,000	67.5									119	115	111	107	104	100	98	95	92	90	88	85	83	81	79	78	76	74	73	71	70	68	67
160,000	148,000	139,000	124,800	110,800	101,000	96,200	89,500	81,800	74,000	68.0									120	117	113	109	105	102	99	96	94	91	89	87	84	83	81	79	77	75	74	73	71	70	68
163,000	151,000	141,500	127,000	113,000	103,000	98,000	91,000	83,200	75,000	68.5										119	115	111	107	104	101	98	95	93	90	88	86	84	82	80	78	76	75	73	72	71	69
166,000	154,000	144,000	130,000	115,000	105,000	100,000	93,000	85,000	76,800	69.0										121	116	113	109	106	102	99	97	94	92	89	87	85	83	81	79	78	76	74	73	72	70
168,500	158,000	146,000	131,500	116,200	106,000	101,800	94,000	86,200	77,800	69.5											118	114	111	107	104	101	98	95	93	91	88	86	84	82	81	79	77	76	74	73	71
172,000	160,000	149,500	134,000	119,000	109,000	104,000	96,800	88,500	80,000	70.0											120	116	112	109	106	103	100	97	95	92	90	88	86	84	82	80	78	77	75	74	72
175,000	163,000	152,000	136,200	122,000	110,500	105,500	98,000	89,600	80,900	70.5												118	114	111	107	104	101	98	96	93	91	89	87	85	83	81	80	78	76	75	73
180,000	166,000	156,500	140,000	124,500	114,000	108,500	100,000	92,500	83,000	71.0												120	116	112	109	106	103	100	98	95	93	90	88	86	84	82	81	79	78	76	74
183,500	169,000	159,200	143,000	126,000	116,000	110,000	101,000	93,500	84,300	71.1													118	114	111	107	104	102	99	96	94	92	90	88	86	84	82	80	79	77	76
186,000	171,000	160,000	145,000	129,000	120,000	112,000	103,000	95,000	86,000	72.0													120	116	112	109	106	103	100	98	95	93	91	89	87	85	83	82	80	78	77

Proper relationship between equipment selected and the internal heat load must be obtained because the cooling-load maximum capacity may operate at a small percentage of the year, and the winter heating load may operate under design conditions only a few days of the year or during shutdown periods. It, therefore, is essential that good partial-load performance be obtained for low operation costs.

Functional Equipment for Central-station System (without Refrigeration). *Evaporative Cooling System.* The evaporative cooling system cools by evaporation, humidifies, heats, and ventilates. This type of system is used with hygroscopic materials requiring close control of relative humidity and varying dry-bulb temperature within specified limits.

The functional equipment for an evaporative cooling system includes

Supply fan, motor, and drive
Humidifier, air-washer spray type
Humidifier pump and motor
Preheating, reheating, booster heating coils
Hot-water heater or steam ejector
Air-cleaning equipment, filters, or precipitators
Automatic controls

This equipment is fabricated with

Casing connections and ducts, normally of sheet metal
Piping—water, steam, air
Electric wiring

In Fig. 21-20 are shown fans of the backward-curve blade type, self-limiting horsepower characteristic. These fans are particularly suitable for industrial applications where high efficiency and reduced power are essential.

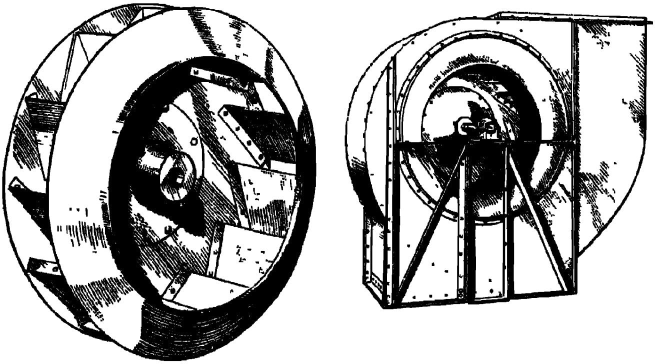


FIG. 21-20. Silentvane fan of the backward-curve blade type.

The humidifier shown in Fig. 21-21 is of the air-washer spray type. It cleanses, purifies, humidifies, and saturates the air that passes through. It is also suitable as a dehumidifier. It is suitable where precise control of humidity and cooling by evaporation is required. Velocities of 500 fpm are recommended.

The filter-washer type of humidifier (Fig. 21-22) is especially adaptable for humidifying and evaporative cooling and employs a bank of mineral-wool or glass-fiber filter pads in conjunction with water sprays to clean, humidify, and saturate the air with an extremely high degree of efficiency. It will remove dust, dirt, and solids down to micronomic fineness. The high efficiency is accomplished by spraying water against a bank of filters so positioned as to offer minimum resistance to air flow with a minimum amount of water recirculated. Thus a substantial saving in fan and pump operating horsepower is effected. Velocities of 700 fpm are recommended.

A sketch of an evaporative cooling system, general arrangement, is shown in Fig. 21-23, Arrangement 1.

Functional Equipment for Central-station System (with Refrigeration). *Dehumidifying Cooling System.* The dehumidifying cooling system cools, dehumidifies, humidifies, heats, and ventilates. This type of system is used with process materials requiring precise control of dry-bulb temperature and relative humidity at all times.

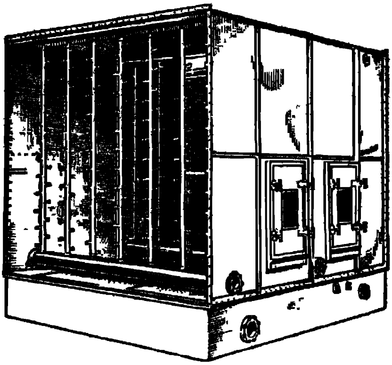


Fig. 21-21. Air-washer spray type—humidifier or dehumidifier with entering baffles removed.

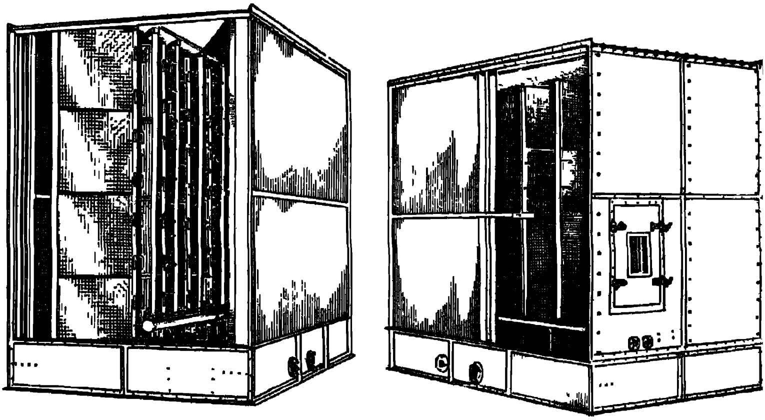


Fig. 21-22. Filter-washer—humidifier with side casing and entering eliminators removed.

The functional equipment for a dehumidifying cooling system includes

- Supply fan, motor, and drive
- Dehumidifier, air-washer spray type, spraying chilled water, well water, or with direct-expansion cooling coils or chilled-water cooling coils located in the spray chamber
- Dehumidifier pump and motor
- Preheating, reheating, booster heating coils
- Hot-water heater
- Air-cleaning equipment, filter, or precipitators
- Automatic control

Refrigeration system—compressor, with motor, condenser, water chiller, and all auxiliary equipment

This equipment is fabricated with

Casing connections and ducts, normally of sheet metal
Piping—water, steam, air
Electric wiring
Supply fans (see page 1500)
Dehumidifier (see page 1501)

The dehumidifier air washer is similar in detail in all respects to the humidifying air washer, except that a greater volume of recirculated water is required for cooling.

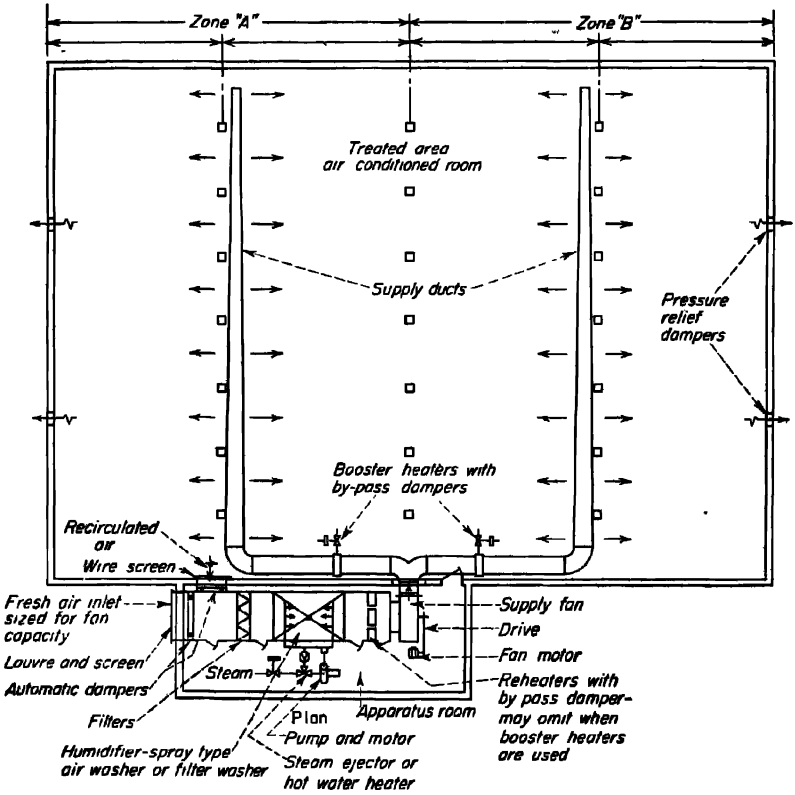

Zone "A"
Zone "B"
Treated area
air conditioned room
Supply ducts
Pressure relief dampers
Booster heaters with by-pass dampers
Recirculated air
Wire screen
Fresh air inlet sized for fan capacity
Louvre and screen
Automatic dampers
Filters
Humidifier-spray type air washer or filter washer
Steam
Plan
Apparatus room
Pump and motor
Steam ejector or hot water heater
Supply fan
Drive
Fan motor
Reheaters with by pass damper-may omit when booster heaters are used

Fig. 21-23. Diagrammatic arrangement of an evaporative cooling, humidifying, heating, and ventilating system. Arrangement 1.
Application: Hygroscopic product requiring precise control of the relative humidity with varying dry-bulb temperature permissible within limits.

In some applications a third bank of sprays is desirable, and the length is increased accordingly.

A sketch of a cooling humidifying and dehumidifying system with refrigeration general arrangement, is shown by Fig. 21-24, Arrangement 2, and Fig. 21-25, Arrangement 3.

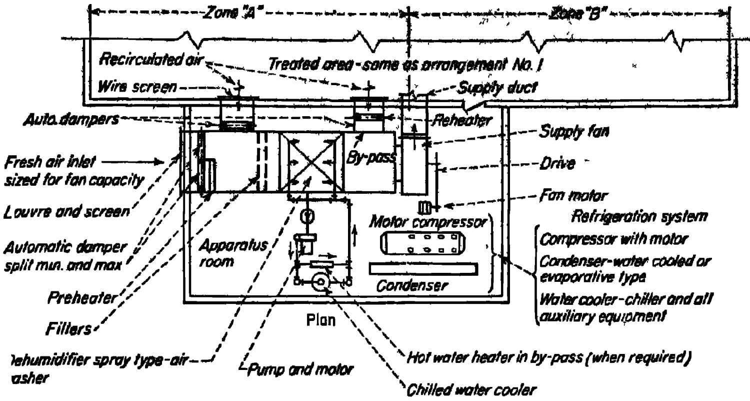

Zone "A"
Zone "B"
Recirculated air
Treated area - same as arrangement No. 1
Wire screen
Supply duct
Auto. dampers
Reheater
Supply fan
Fresh air inlet sized for fan capacity
By-pass
Drive
Fan motor
Louvre and screen
Refrigeration system
Motor compressor
Compressor with motor
Automatic damper split min. and max.
Apparatus room
Condenser-water cooled or evaporative type
Condenser
Water cooler-chiller and all auxiliary equipment
Preheater
Plan
Filters
ehumidifier spray type-air-asher
Pump and motor
Hot water heater in by-pass (when required)
Chilled water cooler

Fig. 21-24. Diagrammatic arrangement of a cooling, humidifying, dehumidifying, heating, and ventilating system with refrigeration. Arrangement 2.
Application: Process material requiring precise control of the dry-bulb temperature and elative humidity within very close limits.

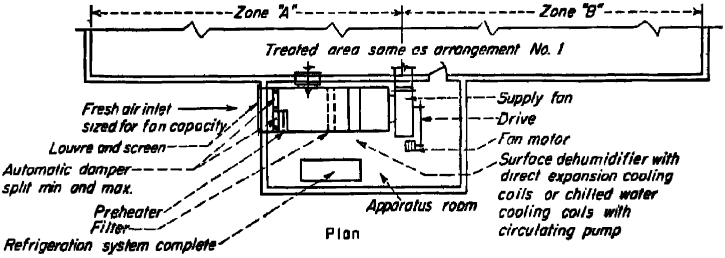

Zone "A"
Zone "B"
Treated area same as arrangement No. 1
Fresh air inlet sized for fan capacity
Supply fan
Drive
Louvre and screen
Fan motor
Automatic damper split min and max.
Surface dehumidifier with direct expansion cooling coils or chilled water cooling coils with circulating pump
Preheater
Filter
Apparatus room
Plan
Refrigeration system complete

Fig. 21-25. Diagrammatic arrangement of a cooling, humidifying, dehumidifying, heating, and ventilating system with refrigeration using direct-expansion or chilled-water cooling coils. Arrangement 3.
Application: Process material permitting slight variations in dry-bulb temperature and relative humidity within limits.

Refrigeration System

The function of the refrigeration system is to cool and dehumidify the air supplied to the treated area. The methods, arrangement, and design have so many ramifications that the subject is covered thoroughly in Sec. 9.

Other types of functional equipment applicable to dehumidifying and cooling systems for process materials include

Refrigeration compressors (Fig. 21-26)—The compressor illustrated is hermetically sealed with a refrigerant-cooled motor. It is designed for Freon-12 refrigerant for use in air-conditioning and industrial refrigeration applications requiring suction evaporating temperatures ranging from 10 to 50°F.

Water-cooled condenser (Fig. 21-27)—The water-cooled condenser is of the shell-and-tube type with the water tubes constructed with integral copper fins for high capacity. Water headers are removable for inspection and cleaning of the tubes.

Aquamiser (evaporative condenser) (Fig. 21-28)—The Aquamiser is a Freon-12 condenser which, in a single, compact, economical unit, combines the desirable features of a water-cooled condenser, an air-cooled condenser, and a cooling tower.

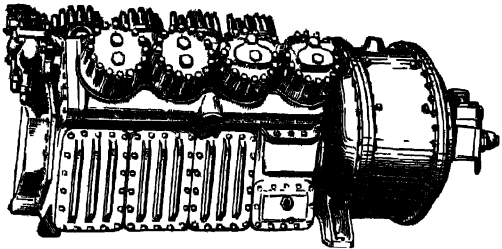


FIG. 21-26. Westinghouse hermetically sealed Freon compressor with refrigerant-cooled motor.

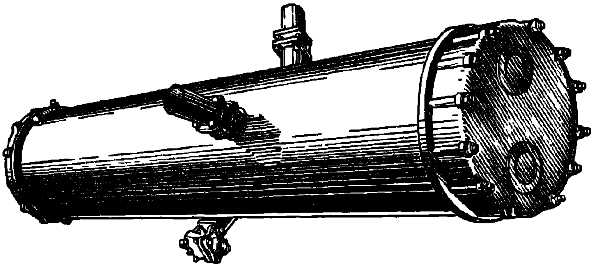


FIG. 21-27. Water-cooled condenser—shell- and tube-type.

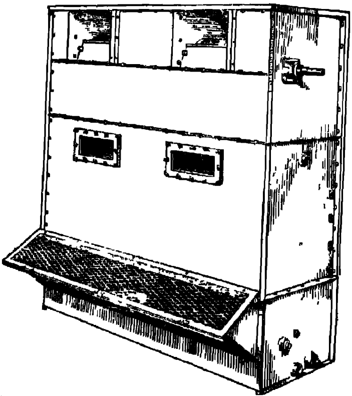


FIG. 21-28. Aquamiser—evaporative-condenser type.

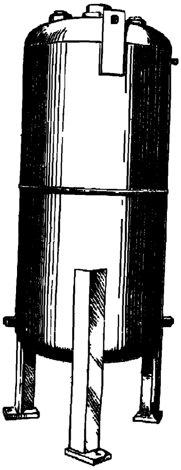


FIG. 21-29. Water-chilling unit—vertical type

It is used instead of a water-cooled condenser where condenser water in sufficient quantity is not available, or its use is restricted. It is more economical in first cost and operating cost than a cooling tower used in conjunction with a water-cooled condenser.

Water-chilling units (Fig. 21-29)—The water-chilling unit illustrated in Fig. 21-29 is a semiflooded type of Freon-12 vertical design. Water is circulated through the coil tubes which are cooled by refrigerant gas evaporating around the outside of the tubes within the steel shell.

Central-plant-type unitaire (Fig. 21-30)—Central-plant-type air conditioners (unitaires) are completely self-contained, with all the component parts of an air-conditioning system, and are designed especially for installation with supply and return air ducts. The compressor is hermetically sealed, direct connected, equipped with 50 per cent unloader, with water-cooled condenser, direct-expansion Freon-12 cooling coils, supply fan, motors, and drive equipped with complete automatic control.

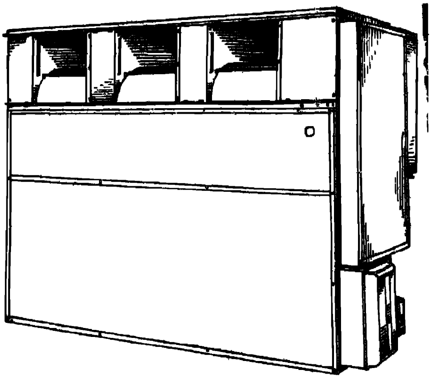


Fig. 21-30. Unitaire air conditioner—central-plant type.

Surface dehumidifier (Fig. 21-31)—The surface dehumidifier is designed with either direct-expansion or chilled-water coils placed directly in the air stream with spray water directed against the coils on the air-entering side. The spray water is carried through the coils so as to wet the entire surface. This arrangement keeps the coils clean and results in practically saturated air leaving the coils. The surface dehumidifier is applicable for process materials where the finesse and accuracy of the dry-bulb temperature and relative humidity are not so essential as in arrangement 2 (Fig. 21-24), slight variations being permissible.

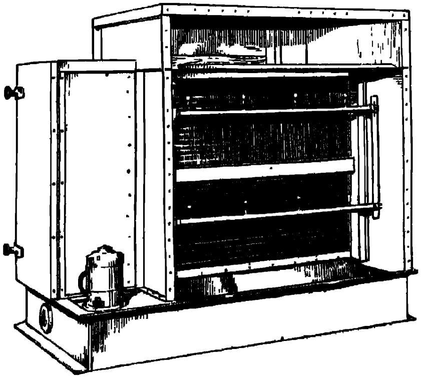
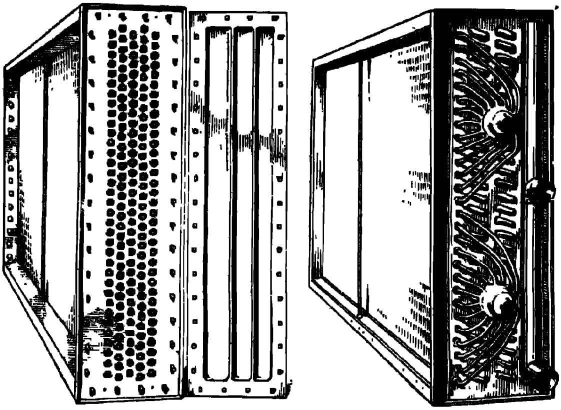


Fig. 21-31. Surface dehumidifier, entering end, with cooling coils and recirculating water pump.

Direct-expansion and Chilled-water Cooling Coils. Diagrammatic sketch of apparatus for arrangement 3 (Fig. 21-25) will be the same as for arrangement 2, except that direct-expansion or chilled-water coils are substituted in place of dehumidifier air washer. Direct-expansion cooling coils have distributing headers giving equal refrigerant distribution to all circuits, whereas the chilled-water cooling coils employ two types: (1) with tubes individually assembled into the headers to permit cleaning and (2) with continuous tubing (see Fig. 21-32).

Arrangement 3 is applicable for process materials requiring control of the dry-bulb temperature within limiting variations and low relative humidities varying but not exceeding prescribed limits.

The preceding arrangements of central-station systems with refrigeration may be varied in design to suit different manufacturers' equipment with many ramifications. The essential fact, however, is that the degree of accuracy of maintaining the atmospheric conditions desired will be governed by the process material and the selection of functional equipment. The degree of accuracy in maintaining precise atmospheric control will be in the order of: (1) spray-type air washer; (2) surface dehumidifier; (3) direct-expansion or chilled-water cooling coils. Each type of functional equipment has a logical application solely judged by the characteristic properties of the process material in sensitiveness to atmospheric conditions.



Chilled-water cooling coils—continuous-tube type with header removed.

Direct-expansion cooling coils with distributing headers.

Fig. 21-32.

CONCLUSION

Industrial process air-conditioning engineering today is one of the most specialized fields of engineering and is indispensable to modern industry. Thousands of manufacturers in hundreds of industries in ever-broadening fields have benefited from the practical application of the fundamental laws and underlying principles of this new and exacting science. Industries whose products in process of manufacture are affected by changing atmospheric conditions demand climate made to order, which means increased production, better quality, ample ventilation, clean, improved, healthful, and comfortable conditions for workers. It improves personal efficiency and results in less loss of time, less labor turnover, and greater profits to the manufacturer. Industrial process air conditioning has progressed to be a large and well-established industry, accepted by the leading and progressive manufacturers as a basic necessity.

Partial List of Industries Employing Industrial Process Air Conditioning

Food products
- 1. Baking
    - Mixer cooling
    - Fermentation—dough room
    - Proofing—proof boxes
    - Loaf cooler

    - Pie cooler—pie crust
    - Cake icings
    - Storage—sugar, flour
  2. Breweries
     - Fermentation room
     - Storage of grains—prevents sprouts
  3. Confectionary
     - Hard candy
     - Chocolate candy
       - Dipping
       - Molding
       - Wrapping
       - Enrobing
       - Packing
       - Storage
     - Cocoa
     - Caramels
     - Chewing gum—chicle
     - Coated gums
     - Marshmallows
     - Salt-water taffy
     - Lifesavers
  4. Cereals
     - Oatmeals
     - Preparation
  5. Drugs
     - Coffee substitutes
     - Headache powders
     - Pills, tablets
     - Antitoxine
     - Storage of powders and tablets
     - Tablet compressing
     - Vaccines
     - Packaging
  6. Fruits
     - Storage of apples
     - Storage of citrus fruits
  7. Meats
     - Ripening of sausage
  8. Miscellaneous food products
     - Almond nuts
     - Yeast
     - Bananas
     - Macaroni
     - Butterine
     - Dry milk
     - Dextrine
     - Milk sugar
     - Gelatin
     - Flour mill
     - Popcorn
- Ceramic products
  1. Grinding wheels
  2. Molding powders
  3. Storage
- Chemical products
  1. Artificial pearls
  2. Artificial jewelry
  3. Artificial leather
  4. Explosives
     - Dynamite
     - Powder mixing
     - Fuse loading
  5. Celluloid
  6. Cellophane
  7. Fibreloid
  8. Dyes
  9. Crystals
  10. Matches
      - Manufacturing
      - Storage
  11. Rubber
      - Manufacturing
      - Cementing
      - Storage
  12. Russian cement
  13. Paint pigments
  14. Powders
  15. Pharmaceuticals
  16. Salts
  17. Mica insulation
  18. Varnish drying
      - Drying of lacquers
      - Drying oil paints
      - Spray drying
  19. Hygienic tubes
- Electrical products
  1. Toll-cable cooling and storage
  2. Insulation winding
  3. Manufacture of cotton-covered wire
  4. Filaments—lamps
  5. Storage of electrical products
  6. Transformer testing
- Fur products
  1. Storage
- Leather products
  1. Shoes
     - Taps
     - Lasts
     - Soles
     - Shoes
- Metal products
  1. Cartridges
  2. Gold beating
  3. Needles
     - Inspection
     - Packing
     - Storage
  4. Enameling
  5. Curtain rods—spray painting
  6. Scales (weighing)—spray painting
  7. Ball bearings—inspection
  8. Typewriters
  9. Precision instruments

Photographic products
1. Developing of film
2. Drying
3. Printing
4. Cutting

Printing products
1. Color printing
2. Decalcomania
3. Gravure
    - Sensitizing
    - Etching
    - Layout
    - Transfer
    - Printing
    - Color gravure
4. Rotoprint
5. Press room
6. Lithographing
    - Press room
    - Bindery
    - Card room
    - Boxing
    - Storage

Paper products
1. Papier-mâché
2. Sandpaper
3. Gum paper
4. Sealed containers
5. Envelopes
6. Carton paper

Textile products
1. Cotton
    - Carding
    - Combing
    - Roving
    - Spinning
    - Weaving
2. Linen
3. Jute
4. Hemp
5. Rayon—manufacture
    - Viscose process
    - Acetate process
    - Cupramonium process
    - Staple rayons
    - Each process
        - Chemical storage
        - Spinning
        - Twisting
        - Reeling
        - Spooling
        - Inspection
        - Storage
6. Silk
    - Spinning
    - Throwing
    - Weaving
7. Wool
    - Carding
    - Preparatory
    - Spinning
    - Weaving
8. Miscellaneous textiles
    - Rayon dress goods
        - Preparation
        - Twisting
        - Weaving
        - Cloth room—inspection
    - Stockings—women's
        - Full-fashioned hoisery
            - Manufacture—machine room
            - Extracting and storage
            - Dyeing and finishing
        - Circular knitting
            - Print cloth and finishing
            - Mercerized cotton
            - Tire fabrics
            - Embroidery
            - Knitting goods
            - Draperies
            - Clothing
            - Testing laboratories

Tobacco products
1. Cigars
    - Sweat room
    - Bin-storage
    - Machine room
    - Packing
    - Storage
2. Cigarettes
    - Sweat room
    - Pulling-up room
    - Storage
    - Machine room
    - Packing
    - Storage

Miscellaneous products
1. Hats
    - Forming
    - Blowing
    - Sizing
    - Blocking
    - Starting
2. Fiber silk
3. Aralac—milk fiber
4. Cork seal
5. Wood finishing
6. Safety glass
7. Optical lenses
8. Glue
9. Catgut
10. Cements
11. Feathers
12. Linoleum

SECTION 22

PNEUMATIC CONVEYING, DUST CONTROL, AND FUME EXHAUST

BY HOWARD M. NICHOLS

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Pneumatic conveying
- *A*. Low-velocity air conveyors; air requirements
- *B*. High-velocity conveying
- *C*. Pneumatic-tube conveyors

Dust control
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III. Fume exhaust
- *A*. Average air velocities
- *B*. Fan requirements
    - 1. Air volume
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    - 3. Pipe friction
- *C*. Pressure drop due to changes in velocities; other pressure drops

IV. Reference

PNEUMATIC CONVEYING, DUST CONTROL, AND FUME EXHAUST

BY HOWARD M. NICHOLS

Assistant Manager, Industrial Department, Westinghouse Electric Corp., Sturtevant Division

PNEUMATIC CONVEYING

There are three general types of pneumatic conveyors: low-velocity air conveyors, high-velocity air conveyors, and pneumatic-tube conveyors.

Air conveyors of these types are adapted to the conveying of bulk materials and find their widest field of usefulness in those situations where mechanical conveyors are restricted because of structural difficulties, lack of room, or excessive first cost compared with the quantity of material to be conveyed. The characteristic advantages of air-conveyor systems are usually lower initial cost than mechanical conveyors (where moderate quantities of material are to be carried a considerable distance), low maintenance expense, and a flexibility of plant whereby all obstacles such as roads, rivers, railroads, buildings, machines, etc., can be overcome without difficulty. The air conveyor will shoot the material around corners, up or down, anywhere the pipe is laid. Other advantages are automatic features, economy in labor, elimination of spillage, elimination of dust and its harmful effect on employees and machinery alike, protection of the conveyed material from exposure and loss while in transit, reduction of accident and fire hazards, reduction of the number of moving machine parts, and the concentration of these parts at some convenient place where they can be most easily taken care of. Balanced against these many advantages are the facts that air conveyors usually consume more power than mechanical conveyors and are restricted in the types and sizes of materials that can be conveyed. As an illustration, sticky materials, friable materials, and materials containing large and heavy lumps are types best handled by other methods. However, air conveyors are successfully handling a wide variety of materials ranging in size from sawdust to wood chips and blocks and in weight from feathers to metal chips. However, air conveyors are limited to the handling of bulk materials, and they are also restricted to such materials as will pass through the piping system without clogging and to such materials as can be readily separated from the conveying air at the discharge end.

Air-conveying systems of both the low- and high-velocity types include the following elements: an air mover which may be a centrifugal fan; centrifugal compressor, or positive-pressure blower; a conveying pipe; a feeding device or suction nozzle; and finally a discharge device where the material is separated from the conveying air. The discharge may be an open end of pipe, a screened bin, a cyclone separator, or a rotary valve or air locks.

Low-velocity Air Conveyors. Low-velocity air conveyors are usually of the fan type and are particularly adapted to handling light bulky materials such as raw cotton, wool, wood shavings, sawdust, shoddy, cut rags, waste paper, rubber scrap, jute, hemp, granulated cork, asbestos, leather buffings, cast-iron chips and borings, crushed-steel chips, aluminum chips, and brass chips. Materials which have a tendency to stick in the conveyor pipes or fans and materials which are easily damaged require special care in the application of pneumatic conveying. Also, hygroscopic materials such as salt and raw sugar and such materials as silk, rayon, unscoured grease wool, and wool heavily loaded with oil emulsions all have a tendency to adhere to the pipe and fan. Fan wheels which have cone backs of smooth construction and which are free from crevices and fins are required to convey cotton, wool, paper trimmings, and similar materials through the fan. Special care is required to have close fits between the arms and blades of the fan as materials of this type will catch in any crevice and build up until the air passages are blocked. In the trade this type of fan wheel is known as a "wool wheel" and is furnished of cast iron, cast bronze, or structural steel. Naturally, the cast wheels are preferable from a smoothness standpoint, but fan speeds are often such that built-up steel wheels are required.

Materials can be conveyed equally well by a suction or a pressure system since the conveying effect is due to the air velocity or the carrying power of air in motion.

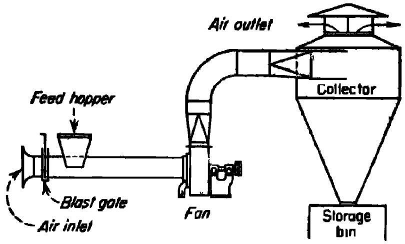

Fig. 22-1. Blow-through conveying system.

The most common and simplest type of air conveyor is the blow-through system (Fig. 22-1) where the material is fed into a hopper in the suction pipe and is then carried by the air stream through the fan and delivered, together with the conveying air, to the collector or separator. At this point, the material is separated from the air, the conveying air being discharged to the room or atmosphere, while the conveyed material is delivered to storage or process by means of a gravity spout. Materials such as wool and wastepaper are usually taken through the fan and blown directly into storage bins, which have the upper portion covered with coarse screen (¼- to 1-in. mesh) to allow the escape of the conveying air.

In draw-through conveying systems (Fig. 22-2), the fan comes after the collector, which consequently is under suction or vacuum and hence requires an airtight seal in the material discharge spout, usually provided by a rotary valve. If intermittent discharge is satisfactory, seal can be maintained by attaching to the discharge spout an airtight storage hopper with blast gates at top and bottom of hopper. Normal operation is with upper gate open and lower gate closed. Collected material is discharged without breaking the vacuum by closing the upper gate and opening the lower gate.

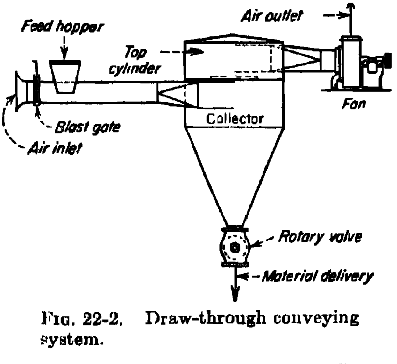

Fig. 22-2. Draw-through conveying system.

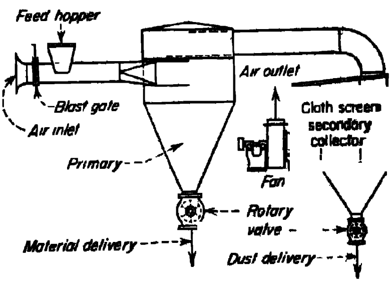

Fig. 22-3. Draw-through conveying system with secondary filter collector.

In the conveying of materials containing fine light dust particles which are to be removed from the conveyed material or which would constitute a dust hazard if discharged at the delivery end of a cyclone collector, the use of a secondary filter collector, as shown by Fig. 22-3; is required. This secondary collector may be a cloth-screen arrestor, electric precipitator, air washer, scrubber, or other comparable device.

In blow-through pressure-feeder conveying systems (Fig. 22-4) the material is fed into the air stream by means of an airtight rotary valve located in the discharge line near the fan outlet. As in the regular blow-through system, the material is discharged from the collector by means of a gravity spout.

Where materials cannot be passed through the fan and blow-through feeder or draw-through arrangements are unsuited to the material, an injector feeder (Fig. 22-5) may be used. Paper strips and wool are types of materials that are sometimes conveyed by the injector method. While the injector method is well adapted for conveying materials which cannot be successfully fed into the system by mechanical means and which cannot be handled through the fan, the mechanical efficiency is much lower than with any of the other types of air conveying. The power requirements are usually five times the power required by the conventional air-conveyor methods, and thus this system is limited to special cases where the advantages outweigh the drawback of greater power consumption.

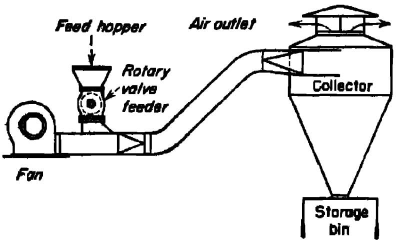

Fig. 22-4. Blow-through pressure-feeder conveying system.

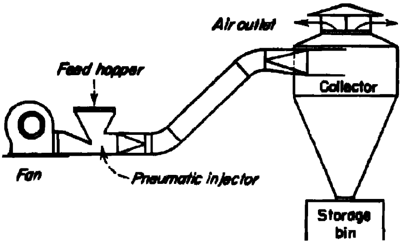

Fig. 22-5. Injector conveying system.

Air Requirements. Two factors are involved in the design of low-velocity air conveyors, *viz.*, necessary air velocity to keep the conveyed material in suspension and in motion in the air stream and quantity of air (cfm) necessary to move the given amount of material. Both the air velocity and the quantity of air per pound of conveyed material depend on the density and physical conditions of the materials If either the air velocity or quantity of air per pound of material falls below the minimum value, the material will start to settle out in the bottom of the horizontal pipe. and at the foot of the risers, and the system will soon plug and become inoperative.

Conveying velocities are dependent on a number of factors, the more important being the size and shape of the particles to be conveyed, the density of the material, and its surface condition.

The theoretical air velocity required to lift a smooth spherical particle is

$$V = 15{,}000 \left(\frac{G}{G+1}\right) \sqrt{d}$$

where V = air velocity, fpm
G = specific gravity of the material
d = particle diameter, in.

This formula is useful only in making a rough check of air velocity, since in practice conveyed materials are not spherical or of uniform size. However, it will be observed from the formula that the air velocity increases with an increase in density of the material and that it also increases with an increase in the particle size. This general relation holds for all types and sizes of materials.

Experience is the best guide in determining which air velocities should be used, or a similar system may be available for actual tests of air velocities and air volumes.

In practice, it is customary to make no distinction between the velocities required for vertical and horizontal transport of materials. It is usually desirable to maintain a minimum velocity of about 3,000 fpm.

Materials that are commonly conveyed by a low-velocity fan system usually have sufficient surface area to be successfully conveyed by the velocities given in Fig. 22-6. The curve in this illustration is based on one variable only, *i.e.*, the bulk weight of a cubic foot of the material in the condition in which it is to be conveyed. If this material happens to be chips with many voids in a cubic foot, the bulk weight will be much less than the density. Also if the material is of a sticky nature, special consideration must be given to the problem. If the material contains moisture, the total weight including the moisture is used in determining the air constants.

It will be noted that the conveying curve gives no consideration to the size of the particles, and for that reason, if there are unusually large pieces of material, a check of the theoretical conveying velocity should be made, assuming an equivalent sphere of the largest piece. Actual conveying velocity should be 50 per cent over this theoretical velocity, or curve velocity, whichever is greater.

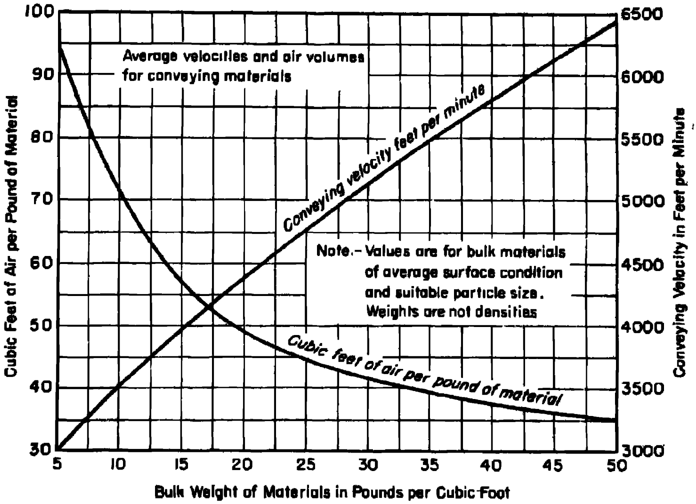

FIG. 22-6. Average velocities and air volumes for conveying material.

There is no theoretical approach to the determination of the amount of air required to convey a pound of material, and consequently it is necessary to depend upon experience and spot checks of existing systems. Experience has demonstrated that with low-velocity systems (3,000 to 7,500 fpm) the cubic feet of air required to convey a pound of material varies from approximately 90 ft of air per lb for light fluffy materials such as open cotton to around 35 to 40 cu ft of air per lb for heavy compact materials.

The curves in Fig. 22-6 are based on an even feed of material to the systems. If the feed is uneven, sufficient air must be provided to convey the maximum quantity of material which may be fed to the system during any short interval.

Within the range of low-velocity conveying, velocities higher than normal do not appear to have much effect on the quantity of air required per pound of material.

A special branch of low-velocity conveying is concerned with the removal of chips, sawdust, or other refuse material from production machines. The most prominent application is in woodworking plants where the machine cutters produce shavings and sawdust in such large quantities that, unless the waste material is removed by a suction system as fast as it is made, it soon clutters up the machines and surrounding floor space, thus necessitating frequent stops to clean the machines and clean away the refuse.

The design of refuse-removal systems for machines is usually governed by the requirements that sufficient suction and air exhaust must be provided around the cutter to draw the chips or shavings into the conveyor pipe. These conditions require excess air over the amount required to convey the material once it is gotten into the conveyor pipe. Similarly, if the conveyor pipes are not more than 20 to 25 per cent greater in area than the area of the pickup ducts, the required suction at the cutter hood usually results in sufficient conveying velocity.

In woodworking plants the size of connections serving the cutter hoods and the suction maintained at these connections will vary with the quantity of shavings made, kind of lumber, its condition, whether dry or wet, speed of the cutters, etc.

The suction required at the hoods varies from 2 to 5 in. of water, depending upon the type of plant, high- or low-speed machines, depth of cut, size of chips, and kind and condition of the lumber.

Two-inch water suction is sufficient for finish mills such as found in the East. Suction of 2½ to 3½ in. is used for heavy manufacturing, while heavy-duty planing mills manufacturing lumber on the Pacific coast use suctions up to 5 in.

In Table 22-1 are listed sizes of hood connections suitable for average conditions in finish mills, general manufacturing, and similar applications.

On very heavy-duty jobs it is desirable to check the air volume and velocities in accordance with the data for straight bulk conveying. This applies particularly to exhaust from hogs and relay systems.

High-velocity Conveying. High velocity or high-pressure pneumatic conveying has a much smaller field of economic application than the conventional low-velocity fan type and is used principally for the direct delivery of dry bulk materials such as soda ash, caustic lime, fullers earth, etc., from cars or boats to storage and process bins. Since the pickup nozzles can be arranged to be self-feeding, this method of conveying is particularly adapted to such materials as soda ash and the like which are extremely disagreeable to work around. Grain is also handled by high-velocity pneumatic systems, but since mechanical methods are more economical in power consumption, the high-velocity pneumatic systems find their principal fields of usefulness where special conditions control, such as unloading barges, picking up spills in grain elevators, etc.

In its simplest form, the high-velocity pneumatic conveyor consists of a positive pressure exhauster or centrifugal exhauster, a material collector or receiver, and the necessary conveying duct, terminating in a flexible hose with a suction nozzle for easy manipulation in the car or storage pile.

An important part of the equipment is the suction nozzle which should be light to allow for easy manipulation by the operator and should have some means of allowing a "free-air" inlet, making it impossible to choke the nozzle by burying the open end. It is an advantage to be able to regulate the free-air inlet according to the conditions existing with different materials.

The material collector usually comprises a centrifugal separator, into which the material is discharged and from which it is withdrawn through an air-lock rotary valve. Fines not separated centrifugally pass into a cylindrical bag filter having a number of compartments. The filter is arranged to offer a maximum of cloth filtering surface in proportion to the total area. The filters are usually cleaned by the automatic reversal of air flow through the cloth and the consequent partial collapse of the tubes. The air reversal is caused by the operation of slow-moving cams which control the opening and closing of the valves. In some types, compressed-air jets are also directed against the bags to assist in the cleaning operation. High-efficiency filters are a necessity since fines that could be passed through a fan and discharged to atmosphere must be excluded from positive-pressure blowers and centrifugal compressors.

In low-velocity conveying sufficient air must be introduced with the material so that the material is kept in suspension in the air stream. If this condition is not fulfilled, some of the material will collect in the bottom of the pipe and slowly build up until the systems become inoperative. Under these conditions the transporting power of the air appears to increase only moderately with an increase in air velocity above the critical velocity for the particular material involved. However, as the velocity is further increased, there appears to be an upper critical point, depending

Table 22-1. Exhaust-hood Connections
(Sizes of connections for woodworking machinery)

Machine	Type of application	No. of connections	Diameter of pipe, in.
Swing saws	Small size on dry-kiln lumber	1	4
	Large on wet	. .	4–6
Ripsaws	Dry-kiln material	1	5
	Not dry-kiln material	1	5
	Self-feed saws	1	5–6
Table saws		1	4
	For box factory work	1	4½
Mitre saws		1	5
Variety saws		1	4
Variety saws	With dado head	1	5
Double saws		2	5
Gang saw	(Dependent on size and number of saws)	.	6–9
Band saws	Blade under 2 in. wide	1	4
	Blade 2–3 in.	1	5
	Blade 3–4 in.	1	6
	Blade 4–6 in.	1	7
	Blade 6–8 in.	1	8
Jig saws		1	4
Tenoning	Single head	1	5
	Double head	2	7
	Double end, double head	4	10
Variety moulder or shaper	Single head	1	5
	Double head	2	7
	Double head, heavy work	2	9
Sanders	Belt, less than 6 in. wide	1	5 +4 on idler
	Belt, 6–8 and 10 in. wide	1	6 +5 on idler
	Belt, 12 and 14 in.	1	7 +5 on idler
	Drum, 24 in. long	1	5
	Drum, 30 in. long	1	6
	Drum, 36 in. long	1	7
	Drum, 48 in. long	1	8
	Drum, over 48 in.	1	10
	Disk, 24 in. diameter	1	5
	Disk, 26–36 in.	1	6
	Disk, 36–48 in.	1	7
	Arm sander	1	4
Planers, matchers, moulders, stickers, jointers, etc. (all top and bottom heads)	Knives 4– 6 in.	One Hood for Each Cutter	5
	Knives 6– 8 in.		6
	Knives 9–14 in.		7
	Knives 15–20 in.		8
	Knives 22–26 in.		9
	Knives 28–36 in.		10
Side knives	Under 4 in.		5
	5–7 in.		6
	8–10 in.		7
	10–12 in.		8
Hog	12 in. wide and under		9
	13–18 in.		10
	19–21 in.		14
	21–24 in.		16

on the characteristics and density of the material, where a given quantity of air will carry much more material. While this phenomenon is termed high-velocity conveying, the action appears to be a combination of moving the material in suspension together with pushing the material along in the pipe by the dynamic pressure of the high-velocity air.

There are practically no authentic published data covering velocity and air quantity required for high-velocity conveying of various material. However, it can be stated that the velocities range from 10,000 to 20,000 fpm and air volumes are one-fifth or less of those required for low-velocity conveying.

Pneumatic-tube Conveyors. The pneumatic-tube conveyor is a conveyor of small articles enclosed in a container which is built to fit the tube closely and which is forced through the conveyor tube by air pressure in a similar manner to the way the piston of a steam engine is pushed along by the steam pressure. Tubes are run between the points to be served, with a dispatch and receiving terminal at each end.

The tubes must be continuous, and preferably should be incased for protection against mechanical injury. Airtight joints and smooth interiors are absolutely necessary for successful operation.

The carriers in which the material to be conveyed is placed are made of metal or molded material covered at the ends with felt. The trailing end of the carrier is provided with a skirt of felt. The air pressure behind it forces it open, making a tight fit and preventing leakage of air past the carrier. The nose is usually provided with a felt buffer which also assists in maintaining an airtight condition.

Pneumatic tubes are worked either by pressure or vacuum. Pressures may range from 1 to 10 lb, and the vacuum system ranges from around a pound to several pounds vacuum. Air volumes depend on the design and tightness of the system. As an illustration, a well-designed single-station system using 2¼-in. tubes and operating on the vacuum principle would require around 50 cfm per station at 1 to 2 lb vacuum.

DUST CONTROL

Control of dust, gases, and fumes in industrial plants is governed in most states by law and supervised by state boards. Most of the basic laws are broad, delegating wide powers to the control boards to set up rules and regulations for the control of these hazards. New York State was a pioneer in this field, and many of the state codes are patterned after the New York code, although in some states there is considerable variation from the New York code, especially as to the suction required at grinding-wheel hoods. Usually definite and detailed specifications are given for the control of dust from grinding wheels and similar applications, while for other dust and fume hazards the rules are of a general nature, specifying that the noxious materials must be removed to a satisfactory degree and leaving the details of accomplishment to the designer of the system.

Where detailed codes are involved, the specific requirements must be followed. However, where there are no detailed code specifications, it is customary to follow the New York specifications in the design of grinding- and buffing-wheel exhaust systems.

The basis of the New York code consists of specifications covering the size of hood connections for given wheel sizes, together with the requirement that 2 in. of water suction must be maintained at the hood connections.

The New York code requirements (current at date of this publication) for minimum size of branch pipes for grinding and buffing wheels are as shown in Table 22-2.

Table 22-2. Branch Connections for Grinding Wheels

Diameter of wheel, in.	Maximum wheel surface, sq in.	Minimum diameter of branch pipe, in.
6 or less, not over 1 thick	19	3
Over 6 to 9 inclusive, not over 1½ thick	43	3½
Over 9 to 16, inclusive, not over 2 thick	101	4
Over 16 to 19, inclusive, not over 3 thick	180	4½
Over 19 to 24, inclusive, not over 4 thick	302	5
Over 24 to 30, inclusive, not over 5 thick	472	6

Where the maximum grinding surface exceeds 472 sq in., a proportion of 1 sq in. of pipe area to each 15 sq in. of grinding surface shall be provided for dust removal. Disk wheels of 24 in. or less in diameter shall be provided with branch pipes at least 4½ in. in diameter; wheels over 24 in. in diameter shall be provided with branch pipes not less than 5 in. in diameter.

If a wheel is thicker than those in Table 22-2, it shall have a branch pipe not smaller than is called for by its wheel surface in the proportion of 1 sq in. of pipe area to each 15 sq in. of grinding surface.

Table 22-3. Branch Connections for Buffing Wheels

Diameter of wheel, in	Maximum wheel surface, sq in.	Minimum diameter of branch pipes, in.
6 or less, not over 1 thick	19	3½
Over 6 to 12, inclusive, not over 2 thick	75	4
Over 12 to 16, inclusive, not over 3 thick	151	4½
Over 16 to 20, inclusive, not over 4 thick	251	5
Over 20 to 24, inclusive, not over 5 thick	377	5½
Over 24 to 30, inclusive, not over 6 thick	565	6½

If a wheel is thicker than those in Table 22-3, it shall have branch pipes not smaller than are called for by its wheel surface as specified.

Buffing wheels 6 in. or less in diameter used for jewelry work may have 3-in. branch pipe.

In grinding, polishing, and buffing auto bumpers, propellers, and other special work, more than one branch pipe from the hood may be necessary, according to the work being done. Similarly, on extremely wide face wheels where the amount of abraded material is comparatively small, a smaller sized branch pipe may be satisfactory.

The code requires that sufficient static suction shall be maintained in every branch pipe to produce a difference of level of at least 2 in. of water between two sides of a U-shaped water gauge. The point at which test is made shall preferably be 10 diameters of the pipe from the hood, but in no case less than 1 ft from the hood. The test is made by placing one end of rubber tube over a small hole made in the pipe, the other end of the tube being connected to one side of the water gauge. This test must be made with all branch-pipe connections open and unobstructed. Balancing devices which are placed on the upstream side of the test hole are not considered as obstruction. Test holes should preferably be drilled since a prick punch leaves an internal burr which is detrimental to accurate suction reading.

General Specifications for Piping. The area of the main suction pipe at any point shall not be less than the combined areas ("load areas") of the pipes joining it between such point and the tail end of the system, except that the area of main suction pipe used for removal of dust from grinding, polishing, and buffing wheels shall be not less than 20 per cent greater than the area of all pipes joining it. The area of floor sweeps, which are normally closed by blast gates, is not included in computing main pipe area.

For easy access to the fan, the length of pipe adjacent to the inlet should be a split sleeve approximately the same length as the diameter of the pipe, but not less than 18 in.

Pipe should be constructed of galvanized iron of not less than the gauges specified in Table 22-4. Elbows should be two gauges heavier than the corresponding straight pipe. For particularly abrasive conditions heavy black iron pipe may be required with elbows having replaceable backs. Where flexibility is required, telescopic pipe, ball joints, flexible metal hose, and reinforced rubber hose are used. Any pipe reinforcing should be made on the outside of the pipe.

All lap joints should be so constructed that the outlet of one length of pipe enters the inlet of the following length in the direction of air flow. All joints should be either lock-seamed, riveted, or spot-welded and in addition should be soldered airtight. Every branch pipe, except in special cases, should be connected at the top or side and at an angle not exceeding 45 deg and should be inclined in the direction of air flow at the junction with the main. Instead of entering the main at an angle shoe collars may be used since their resistance is about the same as a branch entering at an angle.

Increase in diameter of pipes should always be on a taper and not by an abrupt change. Branches should join the main in the tapered section. All pipes should be free from burrs, projections, and sharp edges. Unless space will not permit, every bend, turn, or elbow should be made with a center-line radius at least equal to twice the diameter of the pipe, and all horizontal pipes should be provided with cleanout doors spaced 10 to 20 ft apart and located near branches.

Where large chips or heavy pieces of solid material may be carried along with the air stream into the duct system, chip traps are advisable in the branch ducts, or they may be combined with the machine hoods. Chip traps which depend on air expansion for their effectiveness should have a cross-section area of at least eight times the area of the connecting duct.

Table 22-4. Net Weight of Round Galvanized Iron Pipe and Elbows of Proper Gauge for Blowpipe Systems

Diameter of pipe, in.	Gauge of iron and weight per sq ft	Area of pipe, sq in.	Weight			Diameter of pipe, in.	Gauge of iron and weight per sq ft	Area of pipe, sq in.	Weight		
			Per running foot, lb	Of full elbow	Of elbow two gauges heavier than gauge of pipe				Per running foot, lb	Of full elbow	Of elbow two gauges heavier than gauge of pipe
2	No. 24,	3.1	0 6	0.6	0.8	21	No. 20,	346	10.0	45	58
2½	1.16 lb	4.9	0.7	0.9	1.1	22	1.66 lb	380	10.5	49	64
3		7.1	0.9	1.1	1.4	23		415	11.0	53	69
3½		9.6	1.1	1.4	1.7	24		452	11.5	58	75
4		12.6	1.5	1.8	2.2	25		491	12.0	62	81
5		19.6	1.8	2.4	2.9	26		531	12 5	68	88
6		28.3	2.1	3.4	4.0	27		573	13 0	72	95
7		38.5	2.4	4.2	5.2	28		616	13 5	78	102
8		50.3	2.7	5.4	6.6	29		661	14 0	83	108
9		69.6	3.0	6.5	7.9	30		707	14.5	89	116
10		78 5	3.4	8.2	10.0						
11		95.0	3.7	9.5	11.5	32	No. 18,	804	20.0	131	161
12		113	4.1	11.5	13 5	34	2.16 lb	908	21.0	147	180
13		133	4.4	13.0	16.0	36		1,018	22.5	167	206
14		154	4 7	15.0	18.5	38		1,134	23.5	185	228
15		177	5.0	17.0	20.5	40		1,257	25.0	211	260
						42		1,386	26.0	229	282
16	No. 22,	201	6.6	23.5	27.5	44		1,521	27.5	240	307
17	1.41 lb	227	6.9	26.0	30.5						
18		255	7.3	29.0	34.0	46	No. 16,	1,662	35.0	336	413
19		284	7.7	32.5	38.0	48	2.66 lb	1,810	36.5	363	447
20		314	8.2	35.0	41.0	50		1,964	38.0	390	481

Buffing- and grinding-wheel systems should be segregated since sparks from grinding wheels may set fire to the lint and grease from buffing operation if both are carried through the same duct system.

The duct system should not pass through fire walls or barriers. This can sometimes be avoided by placing the main ducts on the outside walls of the building. Many times vertical risers can be placed in stair wells to avoid piercing important floors. Insurance regulations require that, where ducts do pass through fire walls, they be provided with automatic fire dampers on both sides of the wall.*

Duct systems for the removal of explosive or inflammable materials should be kept independent of other duct systems. The ducts should be as straight as possible, should lead directly to the outside of the building, and should never pass through other rooms. Automatic-sprinkler heads should be provided near the inlet and outlet of the duct system. If installed inside the ducts, they should be placed in an offset and not in the direct path of the air flow. Explosion-relief vents are usually required on duct systems handling explosive or readily flammable materials. Such systems should also be electrically grounded.

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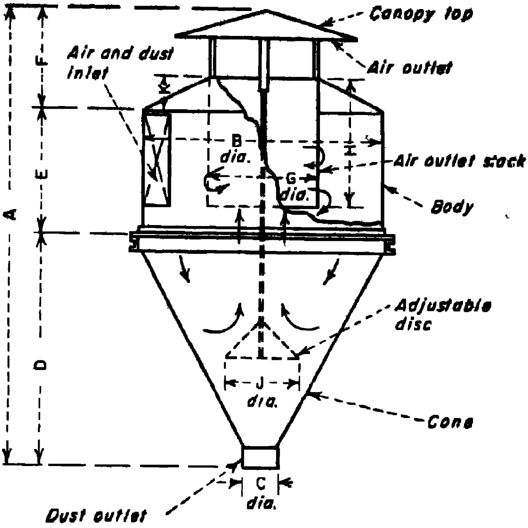
* For details of fire dampers, see regulations published by National Fire Protective Association, Boston, Mass.



Inlets, in.	Dimensions									
	A	B	C	D	E	F	G	H	J	K
2 × 7	3′ 10⅝″	2′ 0″	2″	2′ 2″	12″	8⅝″	8″	12″	6″	4″
3 × 10	4′ 10⅝″	2′ 6″	4″	2′ 8″	15″	11⅝″	12″	16¼″	9″	5″
4½ × 13	6′ 11¾″	3′ 6″	6″	4′ 0″	21″	14¾″	16″	20⅝″	12″	6″
6 × 15	8′ 6¼″	4′ 2″	10″	4′ 8″	2′ 4″	18¼″	20″	2′ 0″	15″	7″
7½ × 17	8′ 11½″	4′ 8″	10″	4′ 10″	2′ 4″	21½″	2′ 0″	2′ 3½″	18″	8″
8½ × 19	9′ 8½″	5′ 4″	10″	5′ 0″	2′ 8″	2′ 0½″	2′ 4″	2′ 7½″	21″	9″
9 × 24	10′ 9¼″	6′ 0″	12″	5′ 6″	2′ 11″	2′ 4¼″	2′ 8″	3′ 1¼″	2′ 0″	10″
9 × 30	12′ 1½″	6′ 8″	12″	6′ 2″	3′ 4″	2′ 7½″	3′ 0″	3′ 8½″	2′ 3″	11″
10 × 33	13′ 5″	7′ 4″	12″	6′ 10″	3′ 8″	2′ 11″	3′ 4″	4′ 1″	2′ 6″	12″
12 × 39	15′ 7¾″	8′ 8″	14″	8′ 1½″	4′ 2½″	3′ 3¾″	4′ 0″	5′ 0½″	3′ 0″	14″
16 × 45	18′ 2″	10′ 0″	16″	9′ 4″	4′ 10″	4′ 0″	5′ 0″	5′ 9½″	3′ 9″	16″
19 × 54	21′ 11½″	12′ 0″	16″	11′ 4″	5′ 10″	4′ 9½″	6′ 0″	6′ 10¾″	4′ 6″	19″
22 × 60	25′ 9½″	13′ 4″	18″	12′ 7″	7′ 9½″	5′ 5″	6′ 8″	8′ 4½″	5′ 0″	21″
27 × 65½	26′ 3″	13′ 8″	18″	12′ 10½″	7′ 9½″	5′ 7″	6′ 10″	9′ 6″	5′ 2″	21½″
32 × 71	26′ 9″	14′ 0″	18″	13′ 2½″	7′ 9½″	5′ 9″	7′ 0″	9′ 11″	5′ 3″	22″

NOTE—Proportion dimensions to determine other sizes.

FIG. 22-7. Cyclone-collector dimensions.

The exhaust fan, often known in the trade as a planing-mill exhauster, should be suitable for handling material through the wheel, which requires that the blades be radial or forwardly pitched.

Dust collectors of the cyclone type are usually located out-of-doors, on the roof or in some other place where a small amount of dust discharged from the stack will not be objectionable. A collector should not be located near open windows. If this cannot be avoided, the collector should be provided with a stack extending above the roof.

Cyclone collectors are usually applied at 3,000 to 3,200 inlet velocity, and under such conditions on grinding-wheel dust, the separation will usually be better than 95 per cent. However, a very small amount of extremely fine dust can create a considerable nuisance. Therefore, care must be exercised regarding the location of cyclone collectors.

The principal dimensions of cyclone collectors are shown by Fig. 22-7.

Where a high degree of separation is required or the cleaned air is to be recirculated, it is necessary to filter or scrub the air, as no centrifugal-type separator will remove fine dust to the degree required. If the dust loading is heavy, it is advisable to use a cyclone collector to remove the bulk of the dust and carry the exhaust from the stack of the cyclone to cloth-screen collector, scrubber, or other high-efficiency separator. Cloth-screen collectors are usually applied at around 3 ft of air per sq ft of cloth filter screen and under such conditions have an average resistance of 2 in. of water under normal operating conditions.

Hoods for the collection of coarse materials must be so shaped that the dispersed particles are thrown by the action of the cutter or wheel into the mouth of the hoods, since these large particles are thrown off with such a high velocity that their trajectories cannot be diverted by the inflowing air (see Fig. 22-8).

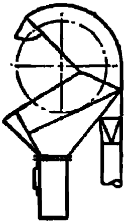


FIG. 22-8. Adjustable hood.

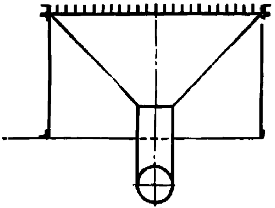


FIG. 22-9. Downdraft hood.

Dust produced by grinding and snagging operations on large and bulky castings may be controlled by downdraft hoods (Fig. 22-9) placed beneath the floor with a grating above them which serves as the work station. Air is drawn over the work down through the grating into the exhaust hood. Successful operation requires that the air velocity around the work be sufficient to remove the dust without objectionable drafts around the workmen's feet. A velocity of 200 fpm over the gross area of the grating will usually be sufficient. Where foundry shakeouts are handled by down draft, 250-ft velocity should be used for cold castings. If the castings are hot, it is necessary to overcome the tendency of the air and dust to rise, which will require around 400-ft velocity.

Foundry shakeouts, grinding, and similar operations may be conducted in booths having one end open and exhausting from the opposite end. The operations require from 100 to 200 fpm air velocity over the open end of the booth.

Processes such as belt and bucket conveyors, screws, etc., are controlled by the inward-leakage principle. The source of dust is enclosed as completely as practicable, and then sufficient air is exhausted from the enclosure to maintain an inward air leakage at all openings, cracks, and crevices. Where it is possible to compute the area of the openings, 150-ft velocity should be allowed over the total leakage area. In any case a liberal amount of exhaust air should be allowed.

Sand-blast rooms are provided with exhaust systems which remove the dust to facilitate the operations, but the dust level is still such that the operators are required to wear masks. Sand-blast rooms may be exhausted either by down draft or longitudinally. Usual air velocity is 70 fpm. Replacement-air inlets are required opposite the exhaust outlets.

Other than the definitely poisonous dusts such as lead, arsenic, mercury, etc., the most hazardous dusts to health are silica and asbestos. Control is usually by hoods and exhausted enclosures, with velocities ranging from 150 to 300 fpm.

Explosive Dusts. Practically all combustible materials when finely divided and suspended in air in the proper concentration can be ignited to produce a violent explosion. Dust explosions have occurred in plants handling coal, charcoal, grain, flour, feed, starch, sugar, powdered milk, wood dust, paper dust, cocoa, spices, hard-rubber dust, cork, sulfur, leather dust, and metal dusts. Of the metal dusts, magnesium and aluminum are particularly hazardous.

Where clouds of explosive dusts are given off, the hazard of a dust explosion can be much reduced by a well-designed dust-control system. However, unless suitable provisions are made in the design of such a system, it in itself may introduce an additional explosion hazard. Where explosive dusts are involved design should be in accordance with the detailed instructions given in "National Fire Codes for the Prevention of Dust Explosions."[1]*

FUME EXHAUST

In industrial plants and process industries, fumes, gases, vapors, and excessive heats are produced by machines, vats, tanks, furnaces, kettles, chemical processes, etc. Their presence represents an occupational health hazard as well as an obstruction to the efficient operation of the equipment. The fumes may be toxic or corrosive and require positive, complete, and continuous removal. On the other hand and according to the process, general and reasonable efficient removal may be all that is required. Between these extremes is a wide range of requirements, some relatively simple, others complex and difficult of accomplishment.

The usual methods of fume removal are general room exhaust, exhaust hoods at the points of contamination, and exhaust from booths and enclosures.

General exhaust is limited in its application (exhaust from hoods, booths, and enclosures is usually preferable) because it is necessary to remove such large quantities of air in order to reduce the percentage of contaminants in the room atmosphere to the proper level. Also, with general exhaust, there is a tendency for the air currents to short-circuit and fail to sweep out all areas. Where the fumes have a tendency to rise, as in foundries, exhaust through roof monitors is indicated, and where pickling and plating tanks are located along an outer wall, exhaust above the tanks and along the wall is quite effective. General exhaust should be so arranged that the natural movement of the fumes is towards the exhaust fans. This can be assisted by furnishing a positive replacement-air supply properly directed. Replacement air should enter the room at points where its velocity will assist the flow of fumes towards the exhaust fans. Cross drafts are to be particularly avoided. The secret of success of general exhaust is to handle large volumes of air, and for that reason direct exhaust without any or with very short ducts should be used. Under these conditions propellor or axial-flow fans, which handle large volumes of air at low pressures, are used. General exhaust requires an air change every 2 to 5 min, and in situations such as dye houses, heated replacement air must be supplied to absorb the moisture to prevent condensation and drip. Zoning the exhaust so that the greater portion of the exhausted air is taken from the most heavily contaminated areas is useful in keeping down the over-all amount of air removed from the building.

Side-wall exhaust above tanks requires a number of propellor fans spaced along the wall to obtain distribution. Satisfactory control requires the movement of around 750 cfm per lin ft of tank. Also, if the ceiling is high, a curtain or false ceiling about 8 ft above the tank is advisable so as to draw the air from the contaminated area. Open windows in the wall or other chances for air short circuit are to be avoided.

Fumes, smoke, vapor, and heated air which tends to rise from tanks, kettles, forges, furnaces, melting pots, and similar equipment are usually exhausted by means of canopy hoods (Figs. 22-10, 22-11, and 22-12). In addition to the usual funnel-shaped canopy hoods, there are double canopy hoods and those with side enclosures. The closer the hood is to the source of contaminant and the more completely the equipment is covered or enclosed, the better the results. The sides of the hoods should slope at an angle of approximately 45 deg with the vertical center line and should extend over the equipment by four-tenths the distance the hood is above the equipment. Air velocity over the face of canopy hoods ranges from 100 to 200 fpm. Duct velocities range from 2,000 to 3,000 fpm. With large hoods, connecting ducts should be branched so as to enter the hood at several points in order to obtain a uniform air flow over the face of the hood.

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* Superior numbers indicate specific reference at the end of this section.

Double canopy hoods are arranged with an inside liner forming a narrow slot about ¾ in. wide around the rim. They also have a central opening under the connecting duct having about one-half the area of this duct. This central duct opening may be provided with a damper for adjustment. The theoretical advantage of canopy hoods is that the high velocity through the slot at the rim has a tendency to prevent smoke and fumes from curling outward at the edge of the hood.

Canopy hoods with side enclosures are useful where the operations will permit closing off the sides of the hoods and are quite essential where side drafts are involved. If enclosures cannot be permanent, movable curtains may be used.

Hoods over cooking ranges are usually placed with the back of the hood against a wall. The exhaust duct can be part of the hood and located as shown in Fig. 22-13. Exhaust openings should be provided with protecting baffles arranged to reduce the entrance of grease into the duct system. Ducts from range hoods should lead directly to outdoors, should be well separated from all walls and partitions, or insulated with heavy asbestos, since there is always a possibility of intense heat as accumulation of grease may take fire. Quite a common error is to exhaust range hoods into chimney flues. This seldom works out satisfactorily as the flue resistance is almost always too high for the amount of exhaust air required properly to remove the smoke.

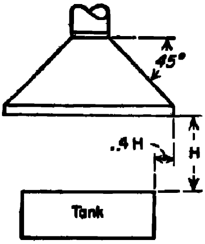

Fig. 22-10. Plain canopy hood.

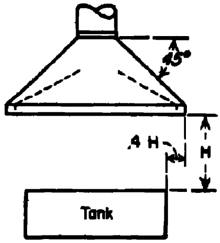

Fig. 22-11. Double canopy hood.

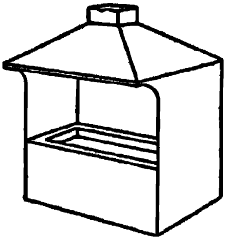

Fig. 22-12. Canopy hood with side enclosure.

Exhaust hoods should be placed as close to the source of the fumes as possible, without interfering with the process. Hoods should be arranged and shaped so that the fumes are directed by any natural air currents into the face of the hoods. Arrangement should be such that the operator is never between the source of the fumes and the hood. Cross drafts from open doors, windows, moving machinery, and heated objects may deflect the exhaust air currents requiring baffling or increased hood velocities. Convection air currents from furnaces and heated objects seldom exceed 75 fpm. Direct draft from open windows, doors, and moving machinery, however, may be of considerable magnitude.

Slotted hoods (Fig. 22-14) are used for exhausting from plating and pickling tanks where overhead cranes prevent the use of canopy hoods. The slots are usually 1¼ in. wide if flanged and 2 in. wide if not flanged. Air volume is based on a minimum of 150 cfm per sq ft of tank area. Slots should be located on three sides of the tank and, where tank is more than 3 ft wide, slot should extend along all four sides. Tanks over 5 ft wide are best handled by combined blow and exhaust hoods (Fig. 22-15). The pressure slot will range in width from ½ to 2 in. The exhaust air should be at least 200 cfm per sq ft of tank area. Air supplied by the pressure slot is one-quarter the exhaust air.

Downdraft exhaust is indicated when the vapors are heavier than normal air and there are no rising air currents to lift the vapors. However, if the vapors are hot,

Table 22-5. Average Air Velocities for Exhaust Hoods

Process	Type of hood	Average air velocity, fpm
Chemical laboratories	Enclosed hoods, sliding door	50–70 over gross opening
	Enclosed hoods, open front	70–90 over face of hood
	Downdraft table hoods	90–125 over fume-producing area of table
	Enclosed hoods in air-conditioned laboratories	Exhaust as above plus a supply of outside air equal to 90 per cent of exhaust air
Degreasing	Canopy hoods	125 over face of hood
	Slotted hoods for tanks up to 3 ft wide	1,500–2,000 through 2-in. slots
Disk grinding and sanding	Down draft through grating	200 over area of grating
Dry cans, padders, and slashers	Canopy and special hoods	25–35 cfm per sq ft of drying-cylinder surface, plus exhaust of 150 fpm over face area of mangle and size-box hoods
Electroplating	Canopy hoods for tanks up to 2 ft wide	125 over face of hood
	Canopy hoods for tanks over 2 ft wide	150 over face of hood
	Slotted hoods for tanks up to 2 ft wide	2,000 through 2-in. slots
	Slotted hoods for tanks over 2 ft wide	2,500 through 2-in. slots
Electric welding	Portable hoods	200 over face of hood
	Open-front booths	100 over face of booth
Foundry shakeout	Down-draft hoods	Hot castings 400 over hood Cold castings 250 over hood
	Open-front booths	150 over face of booth
Grain elevators	Bucket elevator boots	500 over face of hood
	Transfer points	400 over area of all openings
Granite cutting	Plain hood	Surfacing machine, rated on cfm, 540 cfm
	Flanged hood	Surfacing machine, 300 cfm
	Special portable hoods	Hand tools 200 at a point 6 in. from face of hood
Kitchen ranges	Canopy hoods	100–150 over face of hood, depending on size and use
Metal spraying	Open-front booths	Lead 200 over face of booth, zinc and other metals 125–150 over face of booth
Pickling metals	Canopy hoods	200 over face of hood
	Slotted hoods	200–250 cfm per sq ft of tank. Velocity = 2,000 fpm through slots
	Blow and exhaust	Exhaust 200–250 cfm per sq ft of tank. Velocity 2,000 fpm through slots. Blow 50–75 cfm per sq ft of tank. Velocity 3,000 to 5,000 fpm
Rubber mixing rolls	Canopy hoods	200 over face of hood
	Double hoods	2000 through 1½- to 2-in. slots
Steam kettles	Canopy hoods	150 over face of hood
Varnish kettles	Canopy hoods	200 over face of hood

NOTES: 1. Double hoods are usually applied at around 75 per cent of the face velocity of a corresponding canopy hood and with a velocity of 1,000 to 2,000 fpm through the slot at the periphery of the hood.

2. Canopy hoods with side enclosures are usually applied at 100- to 125-fpm velocity over operating area of hood.

3. Air that has been exhausted from a building must be replaced either by infiltration or, if of appreciable quantity, preferably by a fan supply system arranged with heaters to raise the outdoor air to room temperature.

even if heavier than air, the tendency will probably be for them to rise. Successful downdraft exhaust requires that the air velocity be sufficient to control the vapors but not so great as to create objectionable drafts around the workmen's feet.

Air velocities for exhaust hoods (see Table 22-5) vary with local conditions, the severity and toxicity of the fumes, the degree of hood enclosure, and the possibility of cross drafts.

Fan Requirements. *Air Volume.* The volume of air handled by an exhaust system is determined by the total effective connected load area and the suction at the hood connections. By load area is meant the area at the throat of the hoods. The hoods cause a restriction of the air flow which varies with the construction of the hoods and per cent of open area. Bell-mouthed hoods may have an effective area of 95 per cent of gross area, while a thin plate orifice will be around 60 per cent. While restrictive effect will vary from hood to hood, field tests have shown that 70 per cent of the total gross load area is a good average for the net effective load area of the usual exhaust system.

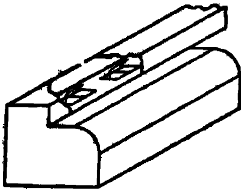

FIG. 22-13. Kitchen hood.

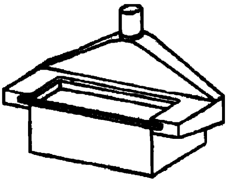

FIG. 22-14. Slotted hood

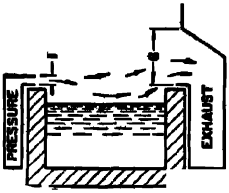

FIG. 22-15. Blow and exhaust hoods.

The velocity of air flow over this area is determined by the formula

$$V = 4{,}000\sqrt{i}$$

where V = air velocity, fpm

i = suction at the hood throat, in. of water

Cubic feet of air per minute (cfm) handled by the system is equal to total net effective load area (in square feet) times air velocity; or cfm = $0.70\,AV$, where A = gross load area in square feet, and V = air velocity in feet per minute

Air volume may also be determined by reference to Table 22-6. Individual volumes for each hood connection are taken from the table and added together to obtain the air volume which the fan must handle.

Table 22-6. Air Volume, Cfm, Handled through Average Collecting Hoods

(Cubic feet of air at 0.075 density handled per minute through average collecting hoods based on coefficient of entry of 0.71 with 10 per cent added for leakage)

Diameter of connection pipe, in.	Maintained suction-in. water gauge						
	1	1½	2	2½	3	4	5
	Vel. = 3,130	Vel. = 3,830	Vel. = 4,430	Vel. = 4,950	Vel. = 5,420	Vel. = 6,260	Vel. = 7,000
1½	38	47	54	61	67	76	86
2	68	84	97	108	118	136	153
2½	107	131	151	168	185	214	238
3	153	188	217	243	266	306	343
3½	209	256	296	330	362	418	466
4	273	334	386	431	473	546	609
4½	345	423	488	546	598	690	775
5	427	523	605	676	741	854	955
6	614	751	867	970	1,062	1,228	1,373
7	835	1,023	1,181	1,322	1,448	1,670	1,870
8	1,092	1,337	1,546	1,727	1,892	2,184	2,440
9	1,381	1,694	1,953	2,184	2,387	2,762	3,091
10	1,705	2,090	2,409	2,695	2,559	3,410	3,806

Exhaust from canopy hoods, slots, and similar applications is usually determined on the basis of a given air velocity over the particular opening. In applications of this type, it is usual to assume the effective orifice area as 70 per cent of the gross area.

Fan Pressure. As soon as air starts to flow in a duct system, a resistance to the flow is set up which must be overcome by the fan pressure in order for the air flow to continue. This resistance to flow is called maintained resistance, which is equivalent to the static pressure that must be produced by the fan.

The maintained resistance is composed of a number of factors, the most important ones being frictional losses in the ducts, losses caused by changes in velocity, work to get the air into the system and bring it up to the required velocity, and resistance of collectors and filters.

To determine the maintained resistance of a system, the point of maximum resistance either must be determined by inspection or trial. This may or may not be at the far end of the system, since a long branch of small diameter may have greater resistance than the additional main from junction point to its far end. Starting at the point of maximum resistance, the maintained resistance of the system is represented by the formula

$$R = W + P + K + E + N$$

where R = maintained resistance = fan static, in. of water
W = work required to get the air into the system and up to the required velocity
P = pipe friction
K = collector resistance
E = pressure drop due to changes in velocity
N = other losses if any

The work W required to get the air into the system will vary with the type of hood or opening into the duct and the suction or air velocity at this opening. Field experience and tests have shown that for average conditions W can be taken as one-half the suction at the hood. Or in the case of a velocity rating at the hood, W is taken as equivalent to the velocity head corresponding to the velocity in the suction pipe. Experience has shown that entrance coefficient on these systems will average close to 70 per cent, which is consistent with the method of determining work at hood entrance.

Pipe Friction. Experience has indicated that friction resistance figured as equivalent to one velocity head loss in 50 diameters of length of pipe will cover the normal loss with sufficient additional allowance to cover the effect of joints and unevenness of commercial pipe and flow of material in the air stream.

Expressed algebraically the pipe friction in straight round pipe, handling air at standard density is

$$P = (L/50) \times (12/d) \times i$$

where P = pipe friction, in. of water
L = length of pipe, ft
d = diameter of pipe, in.
i = velocity head = $(V/4{,}000)^2$
V = air velocity, fpm

For rectangular pipe the friction is increased by the ratio of the perimeter of the rectangular duct to the perimeter of a round duct of equivalent area.

Friction drop in elbows having a center-line radius equal to twice the pipe diameter is taken as equivalent to 8 diameters of length of pipe; for center-line radius equal to pipe diameter, 12 diameters of length.

Collector resistance for cyclone type of standard design, as shown in Fig. 22-7, is given by the formula

$$K = C(V/1{,}000)^2$$

where K = resistance, in. of water
V = entrance air velocity, fpm
C = constant—average value 0.2

Pressure Drop Due to Changes in Velocities. In abrupt enlargement of pipe and consequent drop in velocity, there is a pressure loss which can be taken as from one-half to the full difference in velocity heads corresponding to the initial and final velocities. The more abrupt the enlargement, the greater the percentage loss. If the enlargement is on a gradual taper, the loss is small and may be neglected.

Other Pressure Drops. Collectors of the cloth-bag or screen type run about 2 in. of water loss. Oil-type filters average $\frac{1}{10}$ to $\frac{3}{10}$ in. Paper-filter collectors run about the same as cloth-screen collectors.

Reference

1. "National Fire Codes for the Prevention of Dust Explosions," National Fire Protection Association, Boston, Massachusetts.

SECTION 23

CRUSHING AND GRINDING

BY HARLOWE HARDINGE AND E. B. DOELP

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CRUSHING AND GRINDING

BY HARLOWE HARDINGE

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FUNDAMENTAL PRINCIPLES

The subject of crushing and grinding covers a field so broad and with so many applications that it is difficult to cover the fundamental principles in a way that will apply to the majority of cases found in industrial and metallurgical practices. Crushing and grinding of various materials is carried on usually for one of two reasons: either so that reduction is effected in order that values may be extracted from an ore, or the entire material being ground is a process material for commercial purposes, such as silica and feldspar in the pottery trade, powdered coal for firing of furnaces, limestone and cement in the cement trade, and for many other uses too great in number to enumerate here.

The dividing line between crushing and grinding is not sharp and will vary with different materials being reduced as well as the type of machine effecting the reduction. For this reason certain machines are built to do crushing only, others to do grinding only, and others will perform a combination of both. With but one or two exceptions, if the range of reduction is large, no one machine is capable of efficiently reducing the material through the entire crushing and grinding range. As many as five different crushing and grinding operations are required to reduce a 4- to 6-ft-diameter boulder down to a slime or fine powder, and the method of making this reduction will be shown later when discussing the various types of equipment used.

The materials to be crushed and ground will vary greatly in hardness, and it therefore stands to reason that the hard materials will be much more difficult to crush and grind than will the relatively soft materials. For this reason different types of machines should be used to suit different conditions. To grind and crush soft materials, the more complicated and less expensive type of pulverizer is usually used; but for hard materials, the crushing and grinding equipment must be of sturdy construction and simple design to withstand the severe usage and strains encountered in effecting the necessary reduction. In general, the ores encountered in metallurgical practice are considered comparatively hard as compared with many materials crushed and ground for industrial uses.

CRUSHING

Crushing is a mechanical operation which may be defined as a process of breaking down a material by impact or direct pressure sufficient to cause the material to fracture.

Crushing, as applied in present-day practice, is limited to the size of machine that can be used and to the cost of reduction by this method. Crushers are now available for large-scale operations that will handle rock as large as 8 ft diameter, but where tonnage is low and labor is relatively cheap, it may be economically feasible to install a smaller crusher with a receiving opening to take approximately 70 to 80 per cent of the feed, then sledging or blasting the very large pieces to a size small enough to feed to the crusher. To disintegrate economically materials by the simple crushing action, products are seldom finer than ½ in., though on rare occasions they may be reduced to 10 or 12 mesh.

Crushing machines must be very sturdy and capable of withstanding severe shock and strain, as reduction by crushing requires a great deal of force. Literally hundreds of types of crushers and rolls have been tried out in the past, but of these only a few have survived.

It is necessary to maintain a very definite relationship between the size of feed opening of the jaw crusher and the size of the bottom or discharge opening. This relationship is termed "ratio of reduction." The most efficient ratio of reduction consistent with economical operation is about four diameters, *i.e.*, when the maximum size of feed to the crusher is 4 ft, the most effective work can be accomplished if the discharge is approximately 1 ft. However, the loss in efficiency by increasing the ratio of reduction is often offset by the fact that fewer stages will be required to reduce further the larger pieces, and for this reason, most crushers are designed to effect a greater ratio of reduction than the ideal. In practice, ratios up to 8:1 are sometimes used.

Crushers are designed to operate on the principle of a huge nutcracker. The material to be crushed is dropped between two surfaces so inclined in respect to each other that the opening at the top is larger than the largest piece of material, and the opening at the bottom may be adjusted to the size of product desired. One of the surfaces is made to move a short distance forward, and the material being crushed, being incapable of distortion, is broken under the tremendous pressure. However, the total power to do the crushing is far less than might be supposed, as the distance the crushing surface moves is comparatively slight. For instance, to crush run-of-mine rock from 6 to 1 ft at a rate of 400 tons per hour, about 250 hp is required; but to reduce this same quantity from 1 ft to 2 in. requires nearly 800 hp, with comparative increases on down the line.

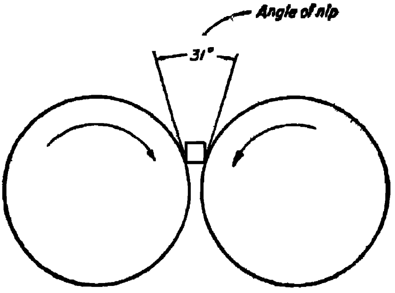

Fig. 23-1. Angle of nip.

Angle of Nip. In nearly all types of crushing devices, the crushing surfaces form an angle termed the "angle of nip." The angle between the crushing surfaces allows the pieces of material to be wedged and broken and then still further reduces the material by the wedging and breaking action as the smaller pieces of material progressively drop into the smaller space between the crusher jaws, until the material is finally discharged from the smallest opening between these jaws.

The angle of nip must be slight enough to prevent slippage of the material wedged between the crushing surfaces; but if the angle is too slight, the resulting ratio of reduction will be too small, and the crusher will not cover the necessary commercial range of reduction required.

The theoretical or critical angle of nip may be designated as that angle where the component of the forces that acts to hold the particles between the crushing surfaces by friction are just balanced by the components of the forces that tend to force the particle outward from the apex of the angle. The safe working angle of nip is therefore something less than the critical or theoretical angle. Theoretically, if there were no friction between the particle and the crushing surface, the only other force exerted would be by the weight of the particle to be wedged. The angle of nip would then have to be very slight to prevent slippage, which means that the safe angle of nip will be determined to a large extent by the friction of the particle against the crushing surfaces. The angle of nip will therefore vary widely depending upon the physical properties of the material being crushed, the character of the crushing surfaces employed, and the speed of approach.

When the crushing surfaces of the machines are round, *e.g.*, rolls, the angle of nip will vary at different points along the crushing surfaces. The angle of nip (Fig. 23-1) in this case is the angle formed by the intersection of two tangents drawn from the two points of contact of the material wedged between the crushing surfaces.

The angle of nip in jaw or gyratory crushers is nearly constant throughout the length of the crushing surfaces, and 31 deg is generally considered the safe working angle, although in an effort to make it possible to crush many different types of material with one machine, some manufacturers have adopted a lesser angle.

Rough rocks with sharp and abrasive characteristics or a rough or corrugated crushing surface will increase the frictional resistance between the crusher surface and the material, thus increasing the effective angle of nip.

Since the material fed to the crusher is not immediately accelerated to the speed of the crusher surfaces and may fail to be wedged between them, high speed of approach of the crushing surfaces will reduce the effective angle of nip.

The surface moisture characteristics of the material fed to the crusher also has some minor effect upon the angle of nip. Dry material is usually better than damp material since clay or fine particles adhering to the surface of damp material will tend to lower the coefficient of friction. However, the coefficient of friction is usually increased if the material sent to the crusher is very wet, owing to the washing away of the clay and fine particles and to the exposure of a clean surface.

TYPES OF CRUSHERS

The various types of crushing devices in practice are known as "primary breakers" and "reduction crushers." These may be either the moving-jaw type or the gyratory crusher. Where the range of reduction is large, these crushers may be operated in series; as many as three stages of so-called "primary breakers" in some cases may be used to advantage, the final product being 1 to ½ in. In general, gyratory crushers will have considerably more capacity when reducing to any given size than will jaw crushers of equal feed opening, and for this reason, when large pieces of material are to be fed, the jaw crusher is often used as the primary breaker. This selection is due to the fact that for a rated capacity the jaw crusher will provide a greater feed opening.

The reduction crusher may take feed as large as 4 to 6 in. and in one operation produce a product all passing ½ in. In practice, the types of reduction crushers generally used are the Dodge, reduction gyratory, rolls, and swing hammer. The latter two types are often used to effect a reduction down to 10 or 12 mesh and are sometimes used to produce an even finer product. This, however, is not generally recommended.

The advantages of adapting one or more of the various mechanical principles found in the crusher to pulverize material have been recognized. If certain of these mechanical advantages could be successfully applied, there would be a considerable saving in the amount of power required, and the character of the product could be more closely controlled than is now possible with the grinding devices currently in use. However, with the exception of some highly limited applications, no such device has been successful. Much difficulty has been experienced when attempts are made to effect fine reduction by the use of the several principles embodied in crushers. As the fineness is increased, the crushing surface must also be greatly increased. The crusher then becomes excessively wide at the point where the fine material is produced, and the movement of the crusher surface, even though small at this point, becomes exceedingly difficult to maintain mechanically. The crushing surfaces are subject to wear, and a very small amount of wear will cause a change in clearance, which in turn will have a major effect on the fineness of the product. In addition, serious complications will be presented by any moisture present in the material, which tends to cause the fines to agglomerate or pack, this being even more serious with reduction crushers when attempting to crush damp material close to their limiting size range.

Some of the grinding devices now in use (see page 1540) do utilize one or more of the basic crushing principles, but in an attempt to avoid the difficulties mentioned above, they will in most cases sacrifice many of the advantages and the simple mechanical construction of the so-called crusher.

Double-toggle Blake Crusher. Jaw crushers are the simplest and most commonly used type of crushing equipment and are used primarily as coarse breakers. Of the three main types of jaw crushers, the most common and successful is the double-toggle Blake-type crusher which is manufactured by many concerns in many modified forms and in many sizes. The largest will take rock up to 8 ft in diameter, and the smallest commercial size will take a feed of around 3 to 4 in. The ratio of reduction may be varied within practical limits to suit the capacity and size requirements of the installation.

The principle of operation is simple. As shown by Fig. 23-2, there is one stationary jaw and one movable jaw, set at an angle to each other, with the large opening at the top to receive the coarse feed. The movable jaw is hinged at the top and oscillated by a toggle near the bottom, and though the coarse material is squeezed less near the top than are the finer particles near the bottom of the jaws, there is still sufficient movement to cause fracture of the large pieces. The resulting pieces then drop down and are further reduced in size in the narrowing opening and finally through the opening between the jaws at the bottom.

The movable jaw is strongly hinged at the top and is actuated through the toggles by the pitman. The pitman is made to move up and down by an eccentric on the crusher drive shaft. At the lower end of the pitman are toggles which are so adjusted that, when the pitman moves up and down, the toggles will exert a horizontal pressure which is transmitted to the lower end of the movable jaw. As the crushing stroke is only half the total cycle, a large flywheel is used to aid the operation during the crushing stroke. The toggles are designed to be the weakest member of the crusher. In the event some tramp iron or hard nonfriable material accidentally gets between the jaws, one or both toggles will break, and no further damage to the crusher will be caused. The movable jaw is held against the toggles at all times by means of a spring which prevents the toggles from jarring loose and dropping out.

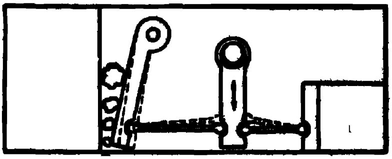

Fig. 23-2. Double-toggle Blake-type crusher.

The jaw plates are usually made of manganese steel to withstand the severe wear to which they are subjected, but several other alloy steels are also being used successfully. All wearing parts are constructed as simply as possible and are made easily replaceable.

To suit various conditions of the material being crushed, the size of the discharge opening, as well as the amount of swing of the movable jaw, may be adjusted. The end of the rear toggle is supported in a tail block, and the size of the discharge opening of the crusher may be adjusted by varying the thickness of shims between the tail block and the crusher frame. The length of stroke or movement of the crusher jaw may also be varied by adjustment of the tail block up or down by means of shims. The stroke is increased by raising the tail block which increases the angle of the rear toggle. Lowering the tail block by means of shims decreases the stroke. The swing of the jaw should be adjusted to suit both the nature of the material being crushed and the degree of fineness required. A long stroke is desirable for coarse crushing and maximum capacity, while a short stroke is more advantageous for fine crushing and to prevent the finer pieces from packing between the jaws.

Specifications of the Traylor-Blake-type double-toggle jaw crusher are given in Table 23-1.

Advantages. The outstanding advantage of the jaw crusher is that it has a larger feed opening, other conditions being equal, than do the gyratory or other types of crushers. For a rated capacity, a larger size of feed can be introduced into this crusher than into any other type. The required headroom for a jaw crusher is less than that of the gyratory crusher of equivalent size or capacity. The jaw plates can be readily inspected and are easily and quickly removed for replacement. The moving parts on the jaw crusher are more accessible for inspection and replacement than on the gyratory crusher. A variation in the length of stroke to suit the character and size of the crusher product can be readily obtained. The first cost of the jaw crusher is usually less than that of the gyratory crusher of equivalent size. The jaw crusher will operate on very hard, tough materials with a minimum of attention and can also handle many damp and sticky materials.

Disadvantages. As the jaw crusher performs useful work during only about one-half its operating cycle, *i.e.*, on the crushing stroke, the power required is usually somewhat greater than that required by an equivalent gyratory crusher. The jaw crusher will have a somewhat greater tendency to choke under heavy feed than will the gyratory crusher, this being caused partly by the large feed opening as compared with the capacity and partly because of the shape and motion of the crushing surfaces. The jaw crusher will not crush to so fine a size as will a gyratory crusher of equivalent capacity.

Table 23-1. Blake-type Double-toggle Jaw-crusher Data*

Size opening, in.	Approx shipping weight, lb	Max rpm driving pulley	Approx capacity per hr, tons of 2,000 lb				Size fly wheel, in.	Max hp required
			Largest		Smallest			
			Ring size, in.	Tons per hr	Ring size, in.	Tons per hr		
24 × 72	162,000	150	4	200	2½	110	78 × 10	150
36 × 42	108,000	175	6	144	4	76	78 × 10	115
36 × 48	115,000	160	6	160	4	86	78 × 10	125
42 × 48	171,500	150	8	260	5	118	96 × 12	150
48 × 60	265,000	120	10	450	6	235	120 × 12	180
56 × 72	410,000	95	10	535	7	310	144 × 14	250
60 × 84	490,000	90	12	980	8	450	144 × 14	300
66 × 86	560,000	90	12	1,110	8	510	144 × 14	300

* From Traylor Engineering and Manufacturing Company *Bulletin*

NOTE: Capacities are based upon materials weighing 100 lb per cu ft when crushed. Horsepower varies with the size of product, output, and hardness of material.

Dodge Crusher. The principle of operation of the Dodge jaw crusher is illustrated by Fig. 23-3. As with the Blake-type crusher, one plate is stationary and one movable, but the movable plate in the Dodge is hinged at the bottom rather than the top. Greater motion takes place at the feed opening than at the discharge opening. This type of construction prevents large pieces of material from dropping through the discharge opening on the backstroke, as occurs with the Blake; thus the Dodge crusher delivers a more uniform product.

Many features of the Dodge crusher are similar to those of the Blake crusher. Both of the crushing surfaces are covered with alloy-steel wearing plates which are quickly and easily removed for replacement. The movable jaw, which is hinged at the bottom on a large pin capable of withstanding very heavy strain, is moved by a strong lever which in turn is actuated by an eccentric on the main drive shaft. A heavy flywheel is used to reduce power consumption and balance the crushing operation. The adjustment of the opening is made by shimming backward or forward the packing blocks of the jaw shaft. Specifications covering the Dodge jaw crusher are given in Table 23-2.

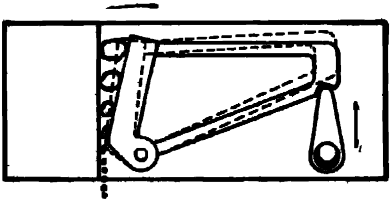

FIG. 23-3. Dodge crusher.

Advantages. The Dodge crusher is generally used as a reduction crusher, and it will produce a uniform product with a minimum of attention and adjustment. The angle of nip for a Dodge crusher is generally greater than that of the Blake type. The Dodge crusher is the simplest and least expensive of the standard-type crushers. This crusher is suitable for small-tonnage installations with a relatively high ratio of reduction.

Disadvantages. The Dodge crusher is less efficient than the Blake; it will choke if fed at too rapid a rate; and considering the floor space required and the power consumed, it does not have the capacity that several other types have.

Table 23-2. Dodge Crusher Data*

Size of jaw opening, in.	Capacity, tons of 2,000 lb/hr (1 ton = 20 cu ft), capacity based on ring size				Jaw motion, in.	Approx hp required	Size of tight and loose pulleys in.	Rpm	Weight complete, lb
	Size of product								
	½ in.	¾ in.	1 in.	1½ in.					
4 × 6	¼	½	1	..	½	3	16 × 4½	275	1,100
7 × 9	.	1	2	3	$^{15}/_{16}$	6	24 × 5½	235	3,250
8 × 12	..	1½	3	4	$^{19}/_{32}$	10	30 × 6½	220	5,900
11 × 15	..	2	4	6	1⅝	15	32 × 10	200	13,500

* From Allis-Chalmers Manufacturing Company *Bulletin*.

NOTE: Crushers may be operated in either direction. Capacities given are approximate and may vary either way according to the character of material to be broken and depend on distance the jaws are set apart and the speed. Hard stone or ore that breaks with a snap will go through faster than sandstone or similar material. Dodge crushers can be set up almost "tight" at discharge.

Single-toggle Jaw Crusher. The single-toggle jaw crusher, illustrated by Fig. 23-4, has one stationary jaw and a movable jaw operated by a mechanism which causes the jaw to be moved at the top as well as at the bottom. The movable jaw is hinged at the top and is mounted on an eccentric on the crusher drive shaft. The lower end of the movable jaw rests against a toggle, and the other end of the toggle is pivoted in a block on the stationary back frame of the crusher. Rotation of the eccentric shaft at the upper end of the movable jaw causes the jaw to move in and out and up and down. The discharge opening of the crusher may be regulated by placing shims or wedge blocks between the toggle block and the stationary back frame. In the event a finer product at lower capacity is desired, the in-and-out movement of the jaw at the bottom can be adjusted by raising or lowering the toggle block.

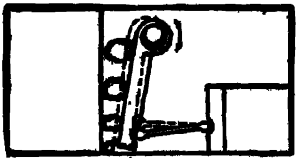

FIG. 23-4. Single-toggle crusher.

Operating data of Telsmith single-toggle jaw crushers are given in Table 23-3.

Advantages. The single-toggle jaw crusher will produce a more uniform product than the Blake type, and for the floor space required, the capacity is high. The construction is simple, and the moving and wearing parts are easily accessible. It is well suited for a moderate-sized installation. The ratio of reduction is good and will produce a good feed for fine-grinding equipment.

Disadvantages. Because the eccentric bearing must take all the weight of the jaw as well as the direct pressure caused by the crushing, there is high bearing pressure at this point. Owing to the heavy weight of the movable jaw, it has not proved practicable to build this type of crusher in large sizes. The rolling motion of the movable jaw plate, which causes some sliding action on the material being crushed, will generally result in a higher wear rate of the jaw plate than experienced in the Blake crusher.

Gyratory Crusher. The gyratory crusher, like the jaw crusher, is often used as a primary breaker and is a modification of the jaw-crushing principle to obtain greater capacity. This crusher is constructed in many sizes to meet many conditions, including large-scale operations. Gyratory crushers are sometimes operated in series, and the second stage and possibly the third stage may still be considered primary crushing if the final product is still in the nature of 1½ in. maximum size.

Table 23-3. Single-toggle Jaw-crusher Data*

Size of crusher (feed opening), in.	10 × 16	10 × 30	14 × 24	18 × 32	25 × 36	30 × 42
Net weight of crusher, lb, about	4,950	9,900	10,000	19,000	38,000	52,000
Horsepower required	8–15	15–25	20–30	50–60	75–100	125–150
Rpm	350	320	320	275	250	250
Capacity, tons per hr with:						
⅝-in. discharge opening	4–6					
¾-in. discharge opening	5–7	11–15				
1-in. discharge opening	7–9	14–20				
1¼-in. discharge opening	8–11	18–25	14–20			
1½-in. discharge opening	10–13	22–30	17–24			
2-in. discharge opening	13–18	29–40	23–32	35–48		
2½-in. discharge opening	16–22	36–50	29–40	44–60		
3-in. discharge opening	.....	.....	35–48	53–72	74–101	
3½-in. discharge opening	.....	.....	.....	62–84	86–118	
4-in. discharge opening	.....	.....	.....	70–96	99–135	116–157
5-in. discharge opening	.....	.....	.....	88–120	124–169	145–199
6-in. discharge opening	.....	.....	.....	.....	149–203	175–239

* From Smith Engineering Works *Bulletin*.

Note: Where no rating is specified in capacity table for any certain discharge opening, crusher cannot be operated economically at that opening. The capacities given are in tons of 2,000 lb and are based on limestone weighing loose about 2,600 lb per cu yd and having a specific gravity of 2.6. The horsepower varies with the size of the product, the capacity, and the hardness of the rock.

For close settings, all undersize material should be removed from the feed so as to eliminate packing and excessive wear on the jaw dies.

No crusher, when set to any given discharge opening, will produce a product all of which will pass a screen opening of the same dimensions as the given discharge opening. The amount of oversize will vary with the character of the rock.

The principle of operation of the gyratory crusher is illustrated by Fig. 23-5. The outer section is a truncated cone, known as the "concave," and is open at the top and bottom. A heavy vertical shaft or spindle carrying a conical crushing surface, known as the "mantle," is suspended in the center of the outer section and is hinged at the top. The lower end of the central shaft is gyrated in a path slightly off the center of the machine, which causes the surface of the mantle to approach the surface of the concave at different points during each revolution. The material to be crushed is dumped on top of the crusher and will lodge in all parts of the annular opening and will be crushed by the same pinching action as described for the jaw crusher. Owing to its circular action, every movement of the gyratory crusher is capable of crushing material, and it will discharge a product continuously; whereas the jaw crusher crushes only about one-half the time. As the crushing action and loading are more continuous than in the jaw-type crusher, the gyratory crusher does not require a heavy flywheel.

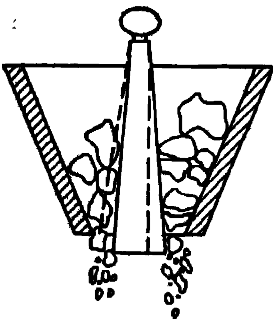

Fig. 23-5. Gyratory crusher.

The central shaft or spindle is rigidly supported and hinged on a swivel joint at the top. The crushing head is fastened to the central shaft and is constructed so that it may be readily replaced when worn. The shell liners on the outer stationary member of the crusher are also readily accessible for inspection and replacement. At the lower end of the central shaft is located the crusher driving mechanism, which consists of an eccentric on which is located a gear. This gear in turn is driven by a pinion fastened to the crusher drive shaft. As the eccentric rotates, it causes the lower end of the shaft to describe a small circle, while the upper end of the shaft is hinged at one point; and the gyrating movement of this shaft changes the distance once each revolution from maximum to minimum between the crushing head on the shaft and the adjacent point on the shell liners. The leverage thus secured by this principle is similar to the pinching action of the jaw crusher and is sufficient to produce a great crushing force. The product is discharged through the bottom annular space between the crushing head and the concave.

A shear pin is
to the drive shaft. This pin is designed to be the weakest member of the drive and will break first in the event any tramp iron or other substance impossible to fracture and too large to pass drops into the crusher. On smaller sizes of gyratory crushers used for intermediate and reduction crushers, the central head is sometimes supported hydraulically and is arranged to open momentarily to allow the passage of tramp iron or any other injurious materials.

Adjustment of the size of the discharge opening of the crusher is performed by raising or lowering the central shaft by means of an adjusting nut located at the top of the shaft. As the conical mantle or crushing head carried by the central shaft extends below the discharge opening of the concave, raising the central shaft will decrease the opening and decrease the size of product produced. Lowering the shaft will increase the opening and the size of the product.

Advantages. As the crushing action of the gyratory crusher is continuous and all motion is available for crushing, less power is required, and the capacity will be greater than the equivalent jaw crusher. It has a higher capacity in proportion to the floor space occupied. When operating, there is less vibration than with the jaw crusher. The gyratory crusher will handle a choke feed to better advantage than will the jaw crusher. It will reduce a slabby feed to better advantage.

Disadvantages. The gyratory crusher will require considerably more headroom than will the equivalent jaw crusher, and when all facts are considered, the gyratory crusher will usually have a higher first cost. The gyratory crusher will not take so large a size of feed as an equivalent-capacity jaw crusher, and wet or sticky materials tend to clog it. Data on Allis-Chalmers primary gyratory crushers are given in Table 23-4.

Table 23-4. Gyratory Primary-crusher Data*

Two feed openings, approx size of each, in.	Countershaft speed, rpm	Approx hp required	Approx net weight of crusher, lb	Approx capacity, tons per hr			
				Min		Max	
				Discharge opening, in.	Capacity	Discharge opening, in.	Capacity
2½ × 11	700	3	700	⅜	½		
8¾ × 35	450	15–25	20,000	1½	30	2⅝	51
10½ × 44	400	25–40	30,000	1¾	40	2¼	57
13⅝ × 44	375	50–75	45,000	2¼	65	3¼	140
15⅞ × 60	350	60–100	62,000	3	127	4	211
20 × 65	330	75–125	94,000	3½	173	4½	243
30 × 98	325	125–175	169,000	4	270	5	433
36½ × 124	300	175–225	263,000	4½	300	6½	606
42½ × 143	300	200–275	286,000	5	418	6½	755
50¼ × 169	250	225–300	575,000	5½	497	7	910
54⅛ × 170	250	225–300	630,000	6	515	7	1,000
59¾ × 196	250	225–300	725,000	7	824	10	1,680
59¾ × 195	250	300–500	1,000,000	7	1,070	10½	2,120

* From Allis-Chalmers Manufacturing Company *Bulletin*.

NOTE: Capacities tabulated above are estimated for average conditions and must be considered approximate. They are based on full continuous feed of quarry or mine-run material weighing 100 lb per cu ft, crushed. The size of crusher indicates the width of feed opening. For special conditions, minimum discharge opening may be decreased or maximum increased depending on characteristics of material to be crushed.

Reduction Gyratory Crusher. Reduction gyratory crushers, because of their high capacity, have reasonably large ratios of reduction. They have met wide use as secondary crushers and are sometimes used as tertiary crushers in very large operations. Where the initial feed to the crusher is relatively small, reduction gyratory crushers may in some instances be used as primary breakers. If the material undergoing reduction is friable and not sticky, this type of crusher may be adapted for crushing down to a size that begins to compete with rod or ball mills.

Reduction gyratory crushers are of basically high speed and are closely set as compared with the standard gyratory crushers.

Symons Cone Crusher. The Symons cone crusher is shown schematically by Fig. 23-6. It is in effect a gyratory reduction crusher but employs a modified principle of operation different from the standard gyratory crushers. The crushing head gyrates in much the same manner as the standard gyratory crusher, but it gyrates several times as far and at a much higher speed. This action of high speed and wide travel results in the material receiving a series of rapid hammerlike blows as it passes downward through the crushing chamber. The pieces of material being reduced pass through the crushing cavity in a series of bounces caused by the high speed of the crushing head, which, as it recedes from the closed position and from under the material, permits each piece of material to fall vertically a short distance before the crushing head comes up again for another crushing impact.

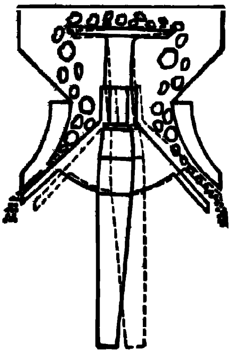

FIG. 23-6. Symons cone-crusher principle.

Because of the short time cycle consumed in one revolution of the crushing head, the material is subjected to a number of crushing impact blows before it can pass through the crushing chamber. As the material continues to move downward, it reaches the lower portion of the crushing chamber where the crushing surfaces are parallel for some distance. This parallel zone is of such length that the material in passing through will receive at least one impact from the crushing head, and all pieces discharged will have at least one dimension no larger than the minimum opening in the parallel zone. The width of opening at the parallel zone on the closed side will regulate the size of product discharged, rather than the open-side dimension as is the case with the gyratory crusher. Sizes, capacities, and power of Symons cone crushers are given in Table 23-5.

Symons cone crushers are protected against damage by a circle of heavy coil springs, which for normal operation hold the stationary crushing concave down firmly on the frame. When tramp iron enters the crushing cavity, these springs are compressed with each revolution of the head, allowing the concave to rise until the noncrushable material has passed into the discharge. These springs automatically return the concave to its normal position after the passage of the injurious material.

Advantages. The Symons cone crusher has much the same advantages as the gyratory crusher. It has a relatively large ratio of reduction, power consumption is economical, and it delivers a uniform size of product at high capacity per floor space required and power consumed. Maintenance is relatively low.

Disadvantages. The Symons cone crusher has a high first cost, particularly for relatively low capacity where the Dodge crusher can be used. It must be adjusted regularly to maintain a specified size of product, and it tends to choke with damp or sticky materials.

Rolls. Rolls, though a different principle from any of the types already described, may be classified as reduction crushers. They are primarily adapted for use where it is desired to crush a final product finer than the previously described types of reduction crushers can.

The ratio of reduction per roll should not exceed 4:1, and for this reason rolls are sometimes operated in series.

Generally, the size of feed is within the range of 3 to ½ in. In exceptional cases, rolls have been built to handle feed as coarse as 5 to 9 in., reducing this to a product of approximately 2 in. They can also be constructed to take a feed of as fine as ¼ in., discharging a product of 10 to 12 mesh, but this operation approaches that of the grinding mill. Only in exceptional cases are rolls recommended for fine reduction, as various types of grinding mills can accomplish the work to a much better advantage.

Table 23-5. Symons Cone-crusher Data*

Size of crusher, ft	Type of bowl	Recommended min discharge setting, in.	Feed opening, in.	Capacities, tons (2,000 lb) per hr at indicated discharge setting with material weighing 100 lb per cu ft								Full load, rpm	Hp
				3/16 in.	3/8 in.	1/2 in.	3/4 in.	1 in.	1½ in.	2 in.	2½ in.		
1⅜	Fine	3/16	1½	8	15	20	...	...	...	...	...	650	20–25
	Coarse	⅜	2⅝	..	15	20	30	40					
2	Fine	¼	2¼	.	20	25	35	...	...	...	...	575	25–30
	Coarse	⅜	3¼	..	20	25	35	45	60				
3	Fine	⅜	3⅞	..	35	40	70	...	...	...	...	580	50–60
	Coarse	½	5⅛	..	..	40	70	80	90	95			
4	Fine	⅜	5	..	60	80	120	150	...	...	...	485	75–100
	Coarse	¾	7⅜	..	...	..	120	150	177	185			
4¼	Fine	½	4½	..	..	100	140						
	Medium	⅝	7⅜	..	..	...	140	160	...	...	...	485	125–150
	Coarse	¾	9½	..	..	..	140	160	185	190			
5½	Fine	⅝	7⅛	..	.	...	200	275					
	Medium	⅞	8⅝	..	.	...	...	275	340	375	450	485	150–200
	Coarse	1	9⅞	.		.	..	275	340	375	450		
7	Fine	¾	10	.	.	..	330	450	600				
	Medium	1	11½	.	.	...	...	450	600	800	...	435	250–300
	Coarse	1¼	13½	.	.	..	...	...	600	800	900		

* From Nordberg Manufacturing Company *Bulletin*.

NOTE: The capacities given are neither minimum nor maximum but are a conservative estimate based on results secured in actual practice.

A diagrammatical view of a set of rolls is shown by Fig. 23-7. The principle of operation of rolls is that of two cylinders with axes and faces parallel and with adjacent faces revolving downward, so that when material is dropped between them it is broken by the pinching and crushing action produced. Either or both of the rolls may be positively driven. The capacity depends upon the size of feed, size of product desired, speed of the rolls, and width of roll face but will vary only slightly with the hardness of the material being crushed when operated under proper conditions. The bearings for one roll are fixed on the base, while the bearings for the other roll slide on the base and are backed by compression springs. This arrangement permits the rolls to move apart and pass any tramp iron or noncrushable material. After tramp iron has passed through the rolls, the compression springs will push the roll into its former operating position without interrupting the functioning of the machine. There should be no movement of the rolls or compression springs when the rolls are properly adjusted and crushing the material for which they were designed. The rolls are kept apart the proper distance by shims between the bearing bases. The shells of the rolls are made replaceable to provide for wear on the crushing surfaces. Bearings are made very heavy in order to withstand the high pressures which are exerted when the rolls are fed to capacity.

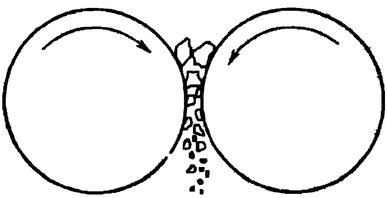

FIG. 23-7. Crushing rolls.

One of the main reasons why rolls cannot be used to better advantage for medium-fine crushing is that the surfaces of the rolls tends to corrugate, thus making it impossible to secure a uniform product when slightly worn. Various schemes have been employed in an attempt to correct this fault, and the method most generally used is to make one roll adjustable laterally. When corrugation starts, this roll is moved laterally slightly, thus placing different surfaces opposite one another and reducing the tendency to corrugate the surface. One type of roll provides for this lateral movement to take place continuously by means of a mechanically actuated end thrust bearing. Another method consists of placing a flat abrasive grinding stone against the ridges as they operate and wear them down. This method requires a great deal of attention, but it occasionally is the only alternative other than to remove the rolls and turn down or replace the faces.

Various specifications for Traylor crushing rolls are given in Table 23-6.

Table 23-6. Crushing-roll Data*

Size of roll diam by face, in.	Approx capacity, tons of 2,000 lb per hr			Rolls without automatic lateral adjusting mechanism		Rpm (usual range)	Hp required
	Size product, in.	Max feed, in.	Capacity, tons	Movable and fixed pulleys, in.	Approx shipping wt, lb		
18 × 10	⅛	⅝	5	24 × 8	5,600	250–300	8
24 × 12	⅛	⅞	10	48 × 6	11,800	180–230	10
30 × 16	⅛	1	20	60 × 10	20,000	130–180	15
36 × 16	¼	1¼	35	72 × 10	59,300	100–150	40
42 × 18	¼	1½	45	72 × 12	65,700	95–120	55
48 × 20	⅜	1⅞	60	84 × 12	81,500	75–105	65
54 × 24	⅜	2	75	84 × 16	103,200	70–95	75
60 × 30	½	2⅜	115	96 × 16	154,000	65–85	100
72 × 36	⅛	3	155	108 × 18	248,800	50–75	150

* From Traylor Engineering and Manufacturing Company *Bulletin*.

NOTE: Capacities are based on materials weighing 100 lb per cu ft when crushed. The horsepower figures shown in the table are for medium capacities only and are based on a medium hardness of product. The actual horsepower required will depend entirely on the work to be done.

Advantages. Rolls are economical of power consumption and require less headroom and floor space than crushers of equivalent capacity. As long as the rolls have smooth surfaces, they will give a more uniform product than it is possible to obtain from any other type of crusher. They fill the gap between reduction crushing and grinding better than any other type of equipment.

Disadvantages. To keep them efficient, rolls require constant attention, and the repair cost is higher than on other crushing devices to do the same work. Corrugation greatly reduces the uniformity of the product as well as the capacity of the rolls. It is generally not economical to produce a product finer than 4 mesh with rolls.

Roll Data. The theoretical capacity of rolls can be obtained by computing the volume of the material passing between the rolls. In practice, the effective capacity of the rolls should be not more than one-quarter the theoretical capacity owing to the unequal character of the feed. The following formula is generally used:

$$DWS/(1{,}728 \times 4) = \text{effective capacity, cu ft/min}$$

where D = distance roll faces are separated, in.
W = width of face, in.
S = peripheral speed, in. per min

The capacity in weight is computed by multiplying the above capacity in cubic feet per minute by the weight per cubic foot of the material being crushed.

Roll Crusher. The roll crusher is a modification of the standard roll and is used for crushing relatively soft materials, such as bituminous coal, alum, salt, and similar materials.

The single-roll crusher consists of a toothed or corrugated roll revolving close to a very heavy stationary breaker plate which is curved and set closer to the roll at the bottom than at the top. The corrugations on the roll tend to increase the angle of nip or actually dig into the coarse lumps as they are fed through the opening above the roll, and in operation the material is crushed as it is dragged downward by the toothed roll and wedged into the decreasing space between the roll and the breaker plate. The material is discharged at a size not greater than the gap between the flat portion of the roll and the base of the stationary plate at the lower edge. The top edge of the

SECTION 24

DRYING

BY R. J. RUSSELL

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DRYING

BY R. J. RUSSELL

Chief of Technical Staff, Hardinge Company, Incorporated

INTRODUCTION

In general plant-engineering practice, drying is the removal of water from a solid by vaporization. Frequently drying is preceded by mechanical dewatering, filtering, or natural drainage to reduce the total cost of preparing a dry product.

Consideration must be given to the design and selection of drying equipment in order to meet the widely varying conditions encountered. In some instances drying must be carried out rapidly to preserve the properties and characteristics required of the dried material. Some materials must be dried slowly with careful control of temperature and humidity conditions to maintain uniform shrinkage. Other materials are not harmed or subject to deterioration during drying, but drying costs can be considerable, so proper drier selection is important.

DRYING FUNDAMENTALS

Vaporization of water proceeds as the vapor pressure exerted exceeds that of the surrounding atmosphere, and the process is accompanied by the absorption of heat. The heat supply is transferred by conduction, convection, or radiation separately or in combination.

Conduction is the flow of heat by direct contact from a heated body or surface to the material being dried.

Convection is the transfer of heat from heated air or gas to the material by passing over or through it.

Radiation is radiant energy in part absorbed by the material and transformed into heat.

Increasing the temperature of the water raises the vapor pressure and accelerates vaporization. The drying rate is proportional to the difference between the vapor pressure of the water and that of the surrounding atmosphere.

In some instances the water is held mechanically in pores or interstices in the solid. When this condition exists, the drying can proceed only as rapidly as the water proceeds by diffusion toward the surface of the solid.

The diffusion process also enters into surface evaporation. The solid is surrounded by a film of gas, and water vapor from the solid diffuses through this relatively stationary film. As the relative velocity of the gas to solid is increased, the film thickness is decreased, and surface evaporation is accelerated.

Air or other gas provides a medium for removal of water vapor from the drying chamber. Exceptions are when the vapor pressure exceeds atmospheric pressure and when drying occurs in a partial vacuum.

A definite and known vapor pressure exists for saturated water vapor at each temperature. When this vapor pressure exists in a given quantity of air at any specific temperature, the air is saturated. The slightest drop in temperature results in condensation of part of the water vapor. Conversely, any increase in temperature permits an increase in the amount of water vapor. Although many drying operations require a control of the ratio of water vapor to air or gas (humidity), rarely is the gas discharged from the drying system saturated (100 per cent humidity).

When the exhaust gases from a drying system are below the boiling point, there is a definite minimum quantity of air or gas necessary for removal of the water vapor. Where the source of heat for drying is the gases carrying away the water vapor, the quantity required generally exceeds the minimum limit. For indirect-heat drying, the minimum air requirement is of importance.

TYPES OF DRIERS

Drying may be a continuous or intermittent process. The intermittent or batch drier is simple in operation, and exact control of drying conditions can be maintained. Generally, thermal efficiency is low and labor requirements are high. Nevertheless, some materials can be dried successfully only in intermittent driers. The continuous drier is preferred and can be used satisfactorily in most instances.

Intermittent driers are known as chamber, compartment, or tray driers. Such driers consist of a chamber heated by steam coils with air circulated by natural convection currents or heated air induced through the chamber. The material may be supported on trays or otherwise suspended in the chamber. Some papers, leathers, ceramic ware, and lumber are best dried in such driers. In many instances drying must be carried out in accordance with a definite time schedule and with exacting temperature and humidity control.

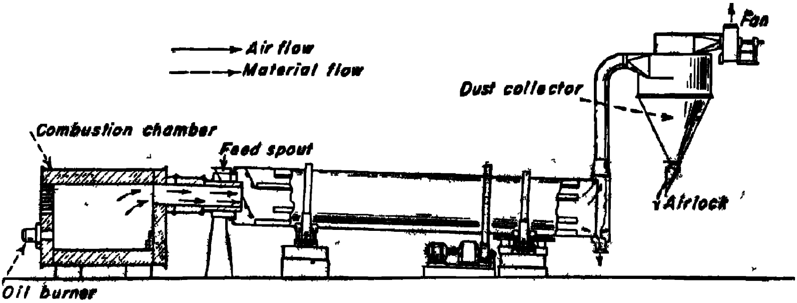

FIG. 24-1. Direct-heat parallel-flow drier. (*Hardinge Company, Inc.*)

Continuous driers are classified by the means for mechanically handling the material being dried. The following are the important types:

The *rotary drier* is a cylinder having internal baffles or flights to lift and lower the material being dried as the cylinder is rotated. Tires or riding rings are attached to the shell and supported on rollers or bearing wheels. One end is slightly elevated to accelerate passage of the material through the cylinder. A gear encircling the drier cylinder engages a driving pinion connected to a speed reducer and power source A fan or stack exhausts the drying gases.

Several types of rotary driers are available to meet a wide range of drying requirements. They are designated as direct-heat, indirect-heat, and reverse-current driers.

In the *direct-heat drier* the material being dried and the drying gases are in direct contact. The gases and material travel in the same direction in the parallel-flow drier (Fig. 24-1) and in opposite directions in the counterflow drier. A variation of the direct-heat drier passes the gases through the mass of material from louvers disposed around the periphery of the shell.

The *indirect-heat drier* is arranged to pass the heated gases through flues or ducts while the material is maintained in contact with the outer surfaces of the flues. Another arrangement has a bank of steam tubes suspended longitudinally within the shell and connected to a steam header as a heat source (Fig. 24-2).

The *reverse-current drier* consists of an inner flue supported within the drier cylinder through which the gases pass before reversing and flowing through annular space formed by the outer cylinder and the inner flue. Material travels through the annular space counter to the flow of gases (Fig. 24-3). An alternate design encases the drying cylinder in brickwork, passing the gases around the cylinder. They then flow through the cylinder counter to the travel of material. Another type has two or three concentric cylinders and transports the material in the gas stream through the smaller inner cylinder and then through the annular areas formed by the encircling larger cylinders.

Conveyor driers are of two principal types. Material is fed onto belts or screens which are mechanically driven to convey the material through a drying chamber, or electric vibrating conveyors may be used (Fig. 24-4). The heated gases pass over the material or through perforations in the conveying surface for the direct-heat types. Indirect drying is accomplished by heating the conveying deck on which the material is transported or by heat from a radiation source such as infrared lamps.

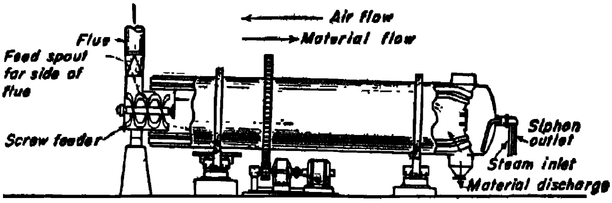

FIG. 24-2. Steam-tube drier. (*Hardinge Company, Inc.*)

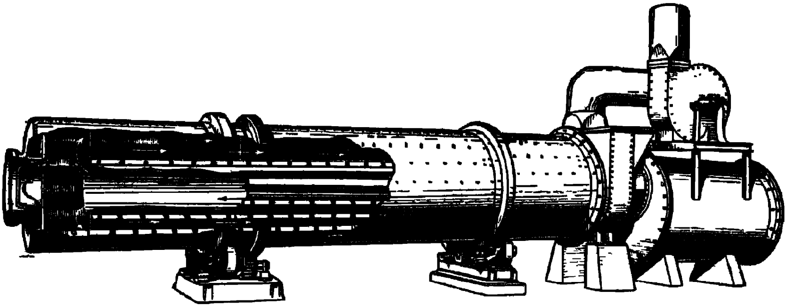

FIG. 24-3. Reverse-current drier. (*Hardinge Company, Inc.*)

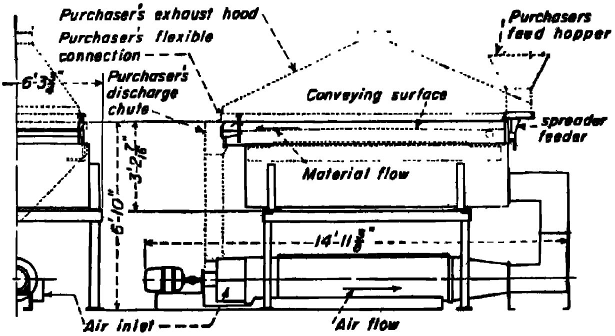

FIG. 24-4. Direct-heat conveyor drier. (*Jeffery Manufacturing Co.*)

Drum driers are used for fine materials usually in the form of a slurry before drying. The rotating drum is steam heated, and the material is fed to the drum from a weir at the top or by dipping the drum into the slurry. The thin film which adheres to the drum is scraped or brushed off after drying (Fig. 24-5). Speed is controlled to complete drying of the film in less than one complete revolution. Multiple drums or steam-heated rolls are used for drying paper, cloth, and similar materials.

Rapid vaporizing driers are known as flash, spray, and dispersion driers. Such driers are generally applicable for materials high in moisture and of very fine particle size. Drying is extremely rapid because of the large exposed surfaces of the material and the manner in which the material is brought into contact with the drying gases to produce a highly turbulent action (Fig. 24-6). The material is agitated by atomization from nozzles, dispersion from a rotating disk or wheel, or by introduction into a high-velocity gas stream.

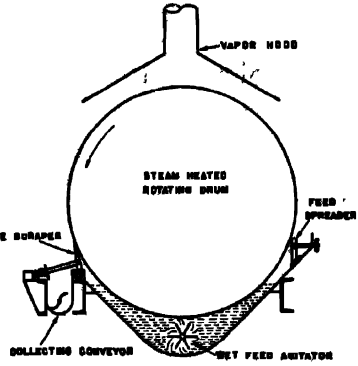

FIG. 24-5. Single-drum drier. (*Buffalo Foundry and Machine Co.*)

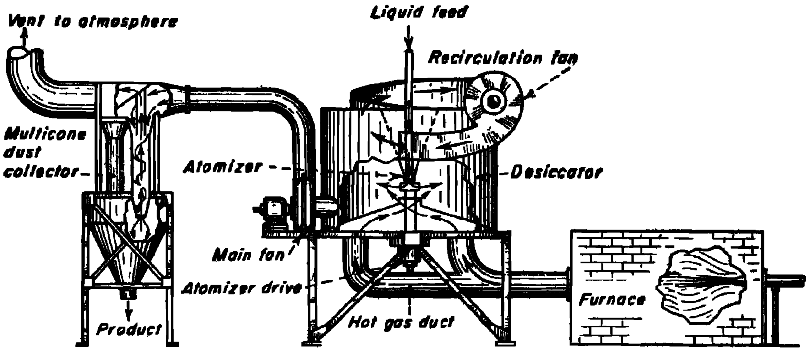

FIG. 24-6. Spray drier. (*Western Precipitation Co.*)

Vacuum drying may be used with special designs of spray, rotary, drum, and conveyor driers where the physical and chemical properties of the material require drying at low temperatures.

SELECTION OF DRYING EQUIPMENT

Every drying requirement is subject to certain limitations which in turn limit the types of driers which can be used. Although the drying capabilities of the several types are overlapping, no drier will handle each drying requirement equally well.

Certain physical and chemical properties of the dried material must be obtained or maintained to make it suitable for subsequent processing or marketing. These requirements are met by control in the drying operation of (1) final moisture content, (2) temperature, (3) drying time, (4) the means for handling the material, and (5) the drying medium.

The final moisture content is controlled by temperature, drying time, and controlled humidity of the drying medium. In rotary driers drying time is controlled by speed of rotation, slope of the shell, and the flight arrangement. The drying time of conveyor driers is also controlled by the rate of travel of the conveying mechanism.

Formed materials or materials destroyed by breakage must be dried by conveyor, tunnel, or batch driers. Rotary driers cause some breakage or reduction in particle size. Sheet materials such as paper require precise handling to allow for shrinkage in drying and yet maintain satisfactory strength qualities of the dry product.

Most driers can be adapted to use a suitable drying medium and can be equipped with automatic controls for temperature and humidity. Such controls often include as a part of the drying system recirculation of part of the gases and recycling part of the dried material.

HEAT REQUIREMENTS OF DRIERS

The theoretical heat requirements for a drying problem can be computed readily. Likewise, the minimum flow of gas required as a carrier for the water vapor can be determined. The basic laws governing flow of heat apply to all drying operations, but any attempt to apply them directly becomes quite involved. In commercial practice empirical formulas have been developed from the theory combined with experience factors.

Heat requirements for drying are

1. Heat required to raise the temperature of the water, vaporize, and in some instances superheat the vapor
2. Heat to raise the dry material to the discharge temperature above the entering temperature
3. Heat to the exhaust air or gases above the ambient temperature
4. Heat loss from the surfaces of the drying system to the surrounding atmosphere

The theoretical heat requirement for drying is that necessary to vaporize the water, or 970 to 1050 Btu per lb, depending upon the temperature at which vaporization occurs. In actual practice, driers require from 125 to 500 per cent of the theoretical heat with an average of 200 per cent.

Table 24-1. Drier Fuel Requirements and Capacities

Type of drier	Heat required, Btu/lb of water evaporated			Capacity factor		
	Temperature range of heat source			Temperature range		
	Low, 100–500°F	Intermediate, 500–1500°F	High, 1500–2500°F	Low, 100–500°F	Intermediate, 500–1500°F	High, 1500–2500°F
Rotary:						
Direct heat	2500–3500	2000–2500	1500–2000	0.3–1.0*	1.0–4.0	4.0–7.5
Indirect heat	3000–4500	2500–3000	2000–2500	0.2–1.0*	1.0–2.0	2.0–3.0
Indirect-direct	....	2000–2500	1500–2000	....	1.5–2.5*	2.5–4.0
Steam tube	1200–1500	....	....	1.0–2.5*		
Conveyor:						
Direct heat	2500–3500	2000–2500	....	1.0–4.0†	4.0–8.0	
Indirect heat	3000–4500	2500–3000	....	0.5–2.0†	2.0–4.0	
Drum	1100–1500	....	....	2.0–6.0†		
Rapid vaporizing	3000–4500	2000–2500	....	0.2–1.0*	1.0–3.0	
Batch	3500–4500	2500–3500				

* Pounds of water evaporated per hour per cubic foot of drier volume.

† Pounds of water evaporated per hour per square foot of drying surface.

NOTES: Exit gas or air velocities for rotary range from 50 fpm for fine low-specific-gravity materials to 900 fpm for granular and coarse materials.

Conveyor driers maintain air velocity through or over the drying bed of 50 to 800 fpm.

Dust-collection equipment (dry cyclone collectors, wet collectors, or bag filters) must be used when complete recovery of dust is required.

Table 24-2. Performance and Cost Data of Ruggles-Coles Driers*

Type of rotary drier	Reverse current			Parallel flow, direct		Counterflow				Steam tube		Warm air
						Direct		Indirect				
Size, shell diam, in., by length, ft	70 × 35	70 × 35	90 × 55	104 × 80	90 × 55	60 × 40	90 × 75	60 × 35	90 × 55	48 × 30	70 × 30	48 × 25
Material	Silica sand	Zinc slimes	Coal	Shale	Pebble phosphate	Manganese ore	Stone	Chalk	Kaolin	Rubber scrap	Corn gluten	Ammonium sulfate
Feed moisture, per cent	5.0	43.0	17.5	10 0	13 8	8.0	5.4	28.0	23.0	38.0	50 0	1.75
Product moisture, per cent	0.1	22.0	2 9	1.0	1.3	0.9	0.1	0.1	2.0	9.0	12 0	0.1
Delivery rate, tons/hr	15.0	5 5	17.8	90.0	55	15	50	1.5	7.5	0.5	0.67	2.65
Type of fuel	Oil	Oil	Coal	Oil	Oil	Oil	Coal	Oil	Coal	Steam	Steam	Steam
Fuel/hr, lb or gal	16	40.2	800	180	193	34	670	19.5	705	805	2,340	290
Fuel, Btu/lb or gal	150,000	151,000	11,300	150,000	150,000	150,000	13,500	140,000	13,600	895	880	945
Water evaporated Btu/lb	1,500	1,520	1,360	1,500	1,810	2,220	1,550	2,340	2,400	1,600	2,050	3,050
Total power, hp	25	19	61	145	85	22	70	18	65	6	10	9
Approximate factory cost†	15,000	15,000	30,000	40,000	26,000	10,000	26,000	15,000	36,000	8,000	12,000	5,000

* Hardinge Company, Incorporated.

† Factory cost covers driers, motors, and fuel-burning equipment.

Generally, when drying from a high initial moisture and delivering a product not entirely free of moisture, thermal requirements per unit of water evaporated are least. Drying to a low final moisture frequently is accompanied by a substantial heat loss as sensible heat in the dried product. A portion of this heat may be recovered economically from roasting or calcining operations, but rarely is it economically feasible in drying. Approximate heat requirements for drying are shown in Table 24-1.

DRIER DESIGN AND OPERATION

The factors that limit evaporating and drying capacity of a given drier are (1) the rate at which heat can be supplied or absorbed, (2) the conveying capacity of the drier, or the time required for drying, and (3) the velocity of the gases leaving the drier for direct-heat driers.

Laboratory tests are helpful in establishing a factor for drying time, and any changes in physical characteristics can be observed. Even though exact commercial drying conditions cannot be reproduced, data can be obtained which are useful in selecting a preferred type of drier and of value in approximating the size of drier for a specific drying requirement. A check is usually obtained by reference to known operations which are similar or comparable. Table 24-1 shows approximate design values for size determination.

DRYING COSTS

Fuel is the principal cost in thermal drying. From the heat requirements in Table 24-1 and a known or selected price for fuel, an approximate cost determination can be made. Power costs range from 10 to 20 per cent of the total drying cost. For other than batch or intermittent drying a single drier does not require the full-time attendance of an operator. Maintenance, lubrication, and incidentals rarely exceed 15 cents per ton of dry product and generally average about 5 cents. Equipment costs are as varied as the types of driers. The materials used in fabrication vary widely in costs, for where corrosive conditions occur, alloy metals are used. Table 24-2 gives data and approximate costs for several types of Ruggles-Coles rotary driers.

PERFORMANCE OF DRIERS

Frequently it is desirable to establish the performance of a drier installation. A batch drier test should complete four to six cycles, and the performance should be based upon an averaging of the results of each batch. For continuous driers the operation should be started and data recorded for test purposes after uniform conditions are reached. The test should then be carried out for not less than 8 hr and readings and samples taken at regular intervals.

Measurements to be taken and data recorded are

1. Amount of material entering and leaving drier
2. Per cent moisture content of wet and dry material
3. Quantity of heat (fuel, steam, or electrical heat supplied)
4. Power
5. Temperature and humidity of gases entering and leaving drier
6. Temperature of material entering and leaving drier

From the foregoing, performance results can be calculated and a heat balance determined. The difference between the heating value of the heat source and that totaled for raising the temperature, evaporating, and superheating the vapor of the water; sensible heat added to the product; and heat loss to exhaust gases is recorded as radiation, convection, and unaccounted-for losses. These losses range from 5 to 15 per cent of the total heat supplied.

SECTION 25

METALLURGY AND CORROSION

BY H. L. MAXWELL AND W. R. MYERS

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METALLURGY AND CORROSION

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INTRODUCTION

Information bearing on materials of construction, particularly those materials used in building corrosion-resistant chemical equipment, will be found in this section. The authors have screened and condensed available information into a brief form for the day-to-day use of plant engineers on practical materials problems. The material presented here, although brief, is authoritative, and while answers will be found to many practical corrosion problems, it is not the intention of the writers that corrosion testing be minimized. Rather the need for corrosion testing under controlled conditions is emphasized. Testing requires time but usually results in over-all economy.*

METHODS OF CORROSION TESTING

In the selection of a constructional material for service facilities and process equipment, all prospective candidates should be appraised on the basis of corrosion resistance and probable life, physical properties, prior performance in the same or similar applications, cost including fabrication and installation, availability in the shapes and forms required, applicable methods of fabrication and repair, anticipated maintenance costs, product contamination, desired life, and salvage value. The appraisal of materials of construction from the corrosion standpoint constitutes one of the most difficult phases of materials engineering. This is true particularly when a choice of materials must be made without benefit of service data and experience.

Materials of construction are either metallic or nonmetallic. The former class consists of metals and their alloys, and the latter includes such materials as glass, concrete, cement, refractories, rubber and rubberlike products, plastics, carbon, graphite, sulfur, silicon, wood, earthenware, stoneware, fused oxides, and many others. The methods employed in evaluating this large variety of materials, particularly in the laboratory or under artificial conditions, must be selected with great care. This applies in particular to the measures used in determining the degree of deterioration or attack.

The deterioration of metals and alloys sometimes takes the form of general attack in which the material is corroded more or less evenly over the exposed surface. Frequently metallic materials corrode by pitting. The corrosion in this case starts at selected areas, and the attack progresses inwardly from the surface. The total amount of metal affected is usually small even if failure is imminent. Corrosion of some metals also occurs through selective attack on certain constituents of the material. This type of corrosion is most frequently found in two-phase or polyphase alloys. It may be intergranular in nature, particularly when a secondary phase is present in or along the grain boundaries. Metals and alloys sometimes fail by stress cracking and stress-corrosion cracking. In failures of this type the actual quantity of metal affected by corrosion is often very small.

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* In cases where more detailed information is required than is given in this section, the reader is referred to the sources listed in the bibliography at the end of this section.

In the nonmetallic group the various materials also deteriorate or are attacked in many ways. Rubber and rubberlike materials and plastics may embrittle, swell, or become tenderized. Bond failure sometimes occurs in rubber-lined equipment, while at the same time the sheet lining itself is affected negligibly. The resins used in making impervious graphite and carbon impose limitations in temperature and service conditions. Wood often fails by swelling or by reactions to form cellulose derivatives. The cement in transite reacts with certain acids leaving an asbestos skeleton. There may be a weight gain or a weight loss depending on whether the salts formed are soluble or insoluble in the corroding solution. This is also true in the reaction of refractories and concrete with acids.

The ideal corrosion-testing equipment would be designed to duplicate conditions to which the materials of construction are exposed under operating conditions. Such equipment would incorporate features making it possible to determine the effect of temperature, velocity, aeration, turbulence, entrainment, impingement, galvanic couples, plastic deformation, residual and applied stresses, and joining and fabrication methods on the corrosion rate. The cost of such equipment would approximate if not exceed that of a small-sized plant. Less costly methods are consequently employed. Those commonly used in the laboratory are the total-immersion test, the alternate-immersion test, the partial-immersion test, and the salt-spray test, the last being employed extensively for protective coatings.

The total-immersion test is the most widely used method. Test data obtained by this method are very valuable, although it is desirable to confirm results by special tests and by semiworks or plant exposure if this is possible.

Total-immersion Tests. *General Description.* The method of total-immersion tests consists essentially of immersing prepared and weighed samples in the test solution for a number of periods of usually two or more days' duration and determining the weight loss by cleaning and reweighing the samples after each period. The corrosion rate of metals and alloys is expressed as weight loss per unit area or as a penetration-rate factor, whichever is desired. If pitting, intergranular or other selective corrosion, or stress-corrosion cracking occurs, it is necessary to modify the method of reporting test data.

Equipment Required. Metal-cutting saws and disks and belt-polishing equipment are required to prepare the samples. If boiling tests are to be made, the testing equipment is generally constructed of glassware consisting of flasks equipped with water-cooled condensers. If the tests are made below the boiling temperature, air condensers or cover glasses may be sufficient to prevent loss of vapor during the test. Electrically or gas-heated hot plates or other suitable heating equipment are required for the boiling tests Constant-temperature baths or furnaces are suitable as heat sources for tests made below the boiling temperature. Glass cradles are generally used for supporting the samples in the test flask or beaker. Micrometers or calipers and an analytical balance are needed to determine the surface area and the weight and density of the samples. In some cases glass testing equipment is not suitable. Testing apparatus is then constructed of hard rubber, plastics, or suitable metals. Protective coatings are sometimes used on glass or metals depending on test conditions. It is also desirable, and in some cases necessary, to have a binocular and a metallurgical microscope and fine polishing equipment.

Size and Finish of Samples. The size and shape of samples should be in harmony with the testing equipment used. Samples that can be polished and measured quickly are preferred. For most purposes metal samples are finished on a 120-grit belt.

Volume of Test Solution. The quantity of test solution used should fulfill the following conditions. It should be sufficiently large that the decrease in the concentration of the active constituent or constituents is not appreciable during the test period. It should be sufficiently large that the accumulation of corrosive products during the test period is small. The volume of solution should be not less than 250 cc for a sample having a surface area of 30 sq cm. A volume of 1,000 to 1,500 cc is preferred for a sample of this size. Flasks of 1,000 cc capacity containing 600 to 700 cc of test solution and samples having a surface area of 10 to 20 sq cm make a satisfactory combination for boiling tests.

Temperature of Corroding Solution. Total-immersion tests may be run over a wide range of selected temperatures. The temperature of tests made below the boiling point should be controlled thermostatically to plus or minus 1°C. Fluctuation in barometric pressure will cause variation in the temperature of boiling tests. Such variations are normally disregarded.

Starting the Test. Metal samples are prepared and finished on a 120-grit belt or paper. They are washed and degreased if necessary, dried, measured, and weighed and the density determined if this is not known. The samples are placed in the cradles. They may be either immersed in the test solution before or after the testing temperature is reached. In boiling tests the samples are often placed in the flasks before the hot plates are turned on. Samples of dissimilar materials should not be placed in a common flask. It is permissible to test duplicates of the same material in a common flask, but the individual samples should not be in contact.

Duration of Test. Tests are generally run for at least 240 hr. This interval should consist of three to five periods of approximately 48 hr each. Fresh test solution should be used for each period. The time count should be started when the solution has reached the test temperature.

Cleaning Samples after Test. At the completion of a period the samples are removed from the test solution and cleaned. Cleaning in many cases can be accomplished by rubbing the samples with a rubber stopper or by brushing with a fiber bristle brush. The dislodged corrosion products are washed away with water. In certain cases the above treatment is not satisfactory.

In some such cases electrolytic cleaning is used, the sample being made the cathode of a cell. Inhibited 5 per cent sulfuric acid makes a suitable electrolyte for general use. Blanks should be run to determine metal weight loss in the cleaning operation, if any. In very troublesome cases either scraping or polishing may be required to clean the samples. The clean samples are finally degreased, if necessary, dried, and reweighed.

Calculation of Corrosion Rate. The corrosion of the samples may be expressed in a variety of ways. One common method employs the so-called "mdd rate" (milligrams per square decimeter per day). Another employs the ipy rate (inches penetration per year) or ipm (inches penetration per month). Some engineers prefer mils per month or mils per year instead of inches per month or inches per year. The calculation of the rate, inches penetration per month, is made as follows:

$$R = (283 \times W)/(A \times D \times t)$$

where R = corrosion rate, in. penetration per month
W = weight loss, grams
A = area, sq cm
D = density, grams/cc
t = time, hr

If pitting or intergranular corrosion or cracking has occurred, the above methods of reporting corrosion data have serious limitations. In the case of pitting, information on size and the number of pits per unit area and the maximum and average depth should be included. If intergranular corrosion occurs, it is often necessary to determine the penetration by microscopic methods. This can be done by fine polishing and examining a section cut from a corrosion coupon. This procedure may also be required if cracking has occurred.

Other methods are also employed in determining degree of deterioration during the test. The loss in tensile strength during test is sometimes used and is an excellent method if the material for tests can be obtained in suitable form for preparing tensile specimens. The decrease in electrical conductivity has been used for measuring intergranular corrosion of austenitic stainless steels.

When testing nonmetallics by the total-immersion tests, the changes in hardness, tensile strength, elongation, and dimensions are of more value than the weight loss or gain.

Special Total-immersion Testing. The effect of oxygen or other gases on the corrosion rate can be determined by introducing the desired gas into the test solution through a diffuser of porous glass or alundum, which is connected to an outside supply source. The supply line is introduced through the condenser tube or through corks or stoppers, if these are used. Oxygen has a marked effect on the corrosion of many metals. This should always be taken into consideration in making tests. The corrosion cracking of metals can be determined semiquantitatively by using U-bend specimens constrained with through bolts. Corrosion creep tests have been made on lead and lead alloys and some of the light metal alloys. Erosion corrosion tests are made to determine the effect of velocity and impingement on the rate of corrosion. It is also possible to obtain data on the corrosion of dissimilar metals in contact with the total-immersion-test method. Two total-immersion tests have been widely adopted for testing the stainless steels—the nitric acid test (ASTM A-262-44T) and the acid copper sulfate test (Strauss and modified Strauss).

Alternate-immersion Test. In this testing procedure special mechanical equipment is required which permits automatic immersion in and withdrawals of the samples from the test solution. The test is designed to determine the effect of alternate exposure to the corrodant and to air, such as occurs in many tanks, pipe lines, and on roofs. The tests are normally conducted at room temperature or only slightly above. The test period is usually longer than is employed in total-immersion testing. The procedure for preparing test samples and determining the degree of deterioration is the same as that used in the total-immersion test.

Partial-immersion Tests. In this test the samples are only partially immersed in the test liquid. The object of this test is to determine the rate of corrosion at the liquid-vapor level. This is important in constant-level tanks and in partially filled pipe lines.

Salt-spray Test (ASTM B-117-44T). This test is widely used for determining the corrosion resistance of protective coatings. It is also employed for testing light metal alloys and stainless steels. It finds its greatest use in selecting materials of construction for exposure to industrial and marine atmospheres and as a control for acceptance tests for certain finishes.

Precautions in Corrosion Testing. In addition to close control of temperature and of the composition of the corrodant, it is necessary to give careful attention to the composition of the material being tested. A small quantity of copper oxide in copper used at high temperatures under reducing conditions will cause rapid failure. The quantity of silicon normally present in semikilled and killed steels causes a rapid increase in the rate of corrosion by fluorine between 200 and 400°C. Lead in zinc-base zinc–aluminum alloys needs to be very carefully controlled to prevent corrosion. Many other examples are known.

A corrosion engineer should remember that to be useful a material of construction must be fabricated into the required shapes. Fabrication often requires hot or cold forming and joining parts by riveting, welding, soldering, brazing, bolting, cementing, and expanding or shrinking. The method of fabrication often determines to a considerable degree how the equipment performs corrosionwise once it is in service. In making corrosion tests these matters must be taken into consideration. The weld-decay corrosion of certain grades of stainless steel is an excellent example of how fabrication procedure may destroy the corrosion resistance of the material of construction. Crevice corrosion and the caustic embrittlement around rivet holes illustrate further what may happen as a result of the fabrication method.

Plant and Field Testing. Much corrosion testing is done in operating plants. In general, the same principle applies here as in the laboratory tests. It is often permissible to test simultaneously a variety of metals, particularly if the tests are made where recycling is not involved. Often the chief difficulty is in designing and building a suitable test rack. In making such racks it must be remembered that the samples must be insulated from the rack and from each other. ASTM A-224-41 outlines recommended practice for conducting plant tests.

SPECIFIC PROPERTIES OF SOME OF THE MORE COMMON METALS AND ALLOYS

Although upwards of four thousand metals and alloys have been listed by name, it may be said that the usual engineering functions of plant design, construction, and operation are concerned with a relatively small number of those listed. The authors have brought together useful information as regards physical properties and corrosion resistance on about sixty of the more commonly used metals and alloys, seventeen of which are classed as stainless alloys.

Brass. Brasses are alloys of copper and zinc containing up to 45 per cent zinc, usually produced in the wrought form in many compositions and in strengths ranging from 35,000 to 125,000 psi. The great variety of compositions, colors, and physical properties make the brasses one of the most useful classes of industrial alloys. *Gilding metal* (95 per cent copper, 5 per cent zinc) is gold-colored and used for jewelry and decorative purposes. *Red brass* (85 per cent copper, 15 per cent zinc), *low brass* (80 per cent copper, 20 per cent zinc), *cartridge brass* (70 per cent copper, 30 per cent zinc), *high brass* (66 per cent copper, 34 per cent zinc), and *Muntz metal* (60 per cent copper, 40 per cent zinc) are the more common varieties used for screws, rivets, condenser tubes, flexible hose, eyelets, springs, musical instruments, drawn shapes, valve stems, etc. Brasses are resistant to atmospheric corrosion but, in general, have slightly less corrosion resistance than pure copper or the copper–tin alloys. The higher the zinc content, the poorer the corrosion resistance to the more severe exposures. Brass suffers two types of corrosion which must be guarded against. *Dezincification* may occur when brasses are exposed to sea water or acidic river water under certain conditions. This peculiar corrosion is caused by dissolving of the brass and redeposition of the copper, resulting in a net loss of zinc. Dezincified areas then consist of a spongy copper deposit, which eventually penetrates the metal. *Season cracking*, often described as stress-corrosion cracking, causes cold-worked or externally stressed brass to crack spontaneously when exposed to moist ammonia atmospheres or mercury salt solutions. Annealing after cold forming will prevent season cracking.

Brasses, like copper, will work-harden after cold-forming operations. The tendency to work-harden increases as the zinc content increases. Hence, articles to be severely cold-worked should preferably be made of one of the low-zinc alloys in order to avoid unnecessary interstage annealing treatment. Brasses are usually tough machining materials, although the machinability increases as the zinc content increases. Free-machining brasses containing a small amount of lead are available and are known as "leaded brass."

Bronze (Tin Bronze). The term "bronze" generally refers to alloys of copper and tin, sometimes containing zinc or lead, which should be distinguished from other alloys such as aluminum bronze (copper–aluminum) and silicon bronze (copper–silicon). Bronzes are reasonably strong (tensile strength 30,000 psi) alloys generally used as castings. *Coinage bronze* contains 92 to 96 per cent copper, with the remainder tin to increase hardness and wearing qualities. *Government bronze*, composed of 88 per cent copper, 10 per cent tin, and 2 per cent zinc, is used for large castings. The addition of zinc increases the fluidity of the metal, producing more dense castings. Another composition is known as "85 and 3 fives" containing 85 per cent copper and 5 per cent each of tin, zinc, and lead, in which the lead is distributed throughout the metal, thus improving machinability. *Bearing bronzes* may be a phosphor bronze, in which phosphorus up to 1 per cent is added as a nonabrasive hardening agent, or a leaded bronze containing from 10 to 30 per cent lead uniformly distributed as small drops throughout the metal. Leaded bronzes are often used for severe bearing services, such as in locomotive journals. In general, the corrosion resistance of bronze is similar to that of copper. Bronze is used in the chemical and food industries for making cast articles to be used in mildly corrosive conditions, *e.g.*, as agitators or paddles in copper equipment. Strongly oxidizing conditions, such as nitric acid, are to be avoided. Bronze fittings are very resistant to atmospheric corrosion and find wide use in domestic and marine hardware.

Cast Iron. Gray cast iron, one of the most useful commonly used metals, is termed "gray" because of the typical color of a newly fractured surface. Gray cast iron is cheap, readily cast to shape, and easily machined. It is a brittle material but if carefully made may have a tensile strength of 60,000 psi with little or no elongation. Ordinary material has a tensile strength of 30,000 psi. Two valuable properties are good damping capacity and freedom from galling and seizing under friction. Gray cast-iron articles varying from a fraction of a pound to several tons are used as automotive cylinders and pistons, machine-tool castings, car wheels, flywheels, water pipe, and hardware.

By small changes in composition or by chilling during casting, *white cast iron* is produced. White cast iron is extremely hard and brittle and valuable for combating certain types of abrasive wear such as is found in plowshares, rolls, car wheels, dies, and wearing plates of various sorts.

Alloy Cast Iron. Nickel, chromium, and molybdenum are sometimes added as alloying elements. Nickel is added up to 5 per cent to improve machinability and to obtain improved combinations of strength, hardness, freedom from porosity, and wear resistance. The addition of 10 to 18 per cent nickel produces a nonmagnetic alloy having high corrosion and scale resistance.

Chromium is added to increase the tendency to chill or to form white iron. Up to 3 per cent chromium is used to produce additional hardness and wear resistance. Nickel-chromium cast iron is produced with various combinations of alloying elements to suit specific requirements.

Molybdenum, added in quantities from 0.25 to 1.25 per cent, increases the tensile strength, transverse strength, and hardness.

Cast irons may be welded successfully by the oxyacetylene or electric-arc processes, if preheated to avoid local hardening and cracking. Other joining methods are conventional brazing and silver brazing.

Malleable Iron. Malleable iron is a ductile form of cast iron, produced by extended annealing of a white cast iron. A tensile strength of 55,000 psi with an elongation of about 20 per cent (compared to elongation of 1 per cent less for gray cast iron) can be obtained. Malleable iron is used for castings that must withstand shock, such as in automobile parts, railway equipment, and agricultural machinery.

Copper. Copper is a strong, ductile, malleable metal which is reddish in color and which, with tin, zinc, nickel, silver, and many other metals, forms numerous commercial alloys of wide industrial application. Many of the alloys of copper possess physical properties such as tensile strength, elastic limit, reduction of area, and elongation that are a marked improvement for commercial purposes over pure copper. In the absence of oxygen, copper is quite resistant to dilute acetic acid. Copper is widely used in electrical equipment where high electrical conductivity is required. Of all the metals, copper is exceeded only by silver as an electrical conductor. Copper can be welded by electric arc, Heliarc arc, or by the oxyacetylene process. A deoxidized copper rod containing some residual phosphorus or boron is preferred. Copper may be brazed or silver brazed. If a joint of high strength is not required, the contact surfaces may be joined by soldering after first "tinning" the surfaces.

Ingot Iron. Ingot iron is a commercially pure grade of basic open-hearth iron, at least 99.8 per cent iron, with the carbon content at approximately 0.012 per cent. The tensile strength is 45,000 psi with an elongation of 45 per cent. Ingot iron is principally rolled into sheets and used for construction work where a ductile, comparatively rust-resistant metal is required. A large amount of ingot iron is sold as galvanized sheets. When alloyed with 0.4 per cent copper and 0.05 per cent molybdenum, ingot iron becomes *Toncan iron* and is claimed to have superior rust-resisting properties.

Silicon Iron. Silicon iron containing approximately 14.5 per cent silicon is available as Duriron or Corrosiron and is widely used as a cast alloy for the manufacture of corrosion-resistant pumps, valves, piping, and other equipment. The alloy can only be cast and is so hard as to be practically unmachinable. Finishing operations are usually done by grinding. The alloys are extremely brittle and cannot be handled roughly or subjected to severe thermal shock. However, in spite of these apparent disadvantages, when properly fabricated, silicon irons are unexcelled for the economical handling of hot nitric, sulfuric, and, to a lesser extent, hydrochloric acid. Silicon irons should not be used with alkali or hydrofluoric acid.

Lead. Lead is a soft, malleable, heavy metal (specific gravity 11.38) of relatively low tensile strength (3,000 psi), but it is widely used because of its resistance to corrosion by sulfuric acids and sulfates. It is readily fabricated by "lead burning" into various shapes for use in chemical equipment. Oxyhydrogen flame is the preferred means of heating. Lead is widely used as pipe and cable coverings, for process equipment handling sulfuric acid and sulfates, and in a variety of bearing alloys and related applications. One of the lesser known uses for lead is in "leaded" brass. During cooling and rolling of brass containing, say, up to 2 per cent of lead, the soft metal separates out, and the globules of lead are distributed throughout the body of the brass serving as a lubricant to the cutting tool and are the basis for "free-machining brass."

Monel. Monel metal is the trade name of a nickel-copper alloy obtained by smelting a natural ore of such a composition that an alloy of about 67 per cent copper, 28 per cent nickel is produced. Monel is a silvery white metal which may be cast, rolled, or forged. The tensile strength is 80,000 psi for annealed material and 140,000 psi for cold-drawn material. A free-machining grade (R-monel) having slightly increased sulfur content, is commercially available.

Monel is widely used in the food and chemical industries for its corrosion resistance both to atmospheric conditions and the more severe services such as dilute sulfuric acid, hydrofluoric acid, and alkali. Monel sheet is used in domestic kitchen furnishing and in architectural decorative work.

Monel is easily fabricated by drawing, stamping, rolling, and forging, although the material work hardens fairly rapidly. As with nickel, machining is best done with sharp high-speed or carbide tools, and care must be taken to avoid sulfurous atmospheres during annealing. Monel may be joined by fusion welding, using the oxyacetylene or electric-arc processes, or by resistance spot welding. Phosphorus-containing silver brazing alloys are to be avoided, but other silver brazing alloys flowing below 1400°F may be used. Soft solders are acceptable in certain services.

Nickel. Nickel is widely used in the commercially pure condition as wrought plate, bar, sheet, tube, and wire, and as shot for alloying purposes. Nickel is a silvery white metal having a tensile strength of about 70,000 psi which can be increased to 115,000 psi by cold working. Nickel is a tough metal, having high impact strength which is retained even at subzero temperatures. Nickel and nickel alloys constitute a large fraction of the "stainless" or rust-resisting metals.

Pure nickel is widely used for its corrosion resistance. It is generally resistant to nonoxidizing acid, neutral, and alkaline solutions; it is suitable for handling food products and may be used either as solid metal, clad plate, or electroplate. Nickel is extremely resistant to caustic solutions and nonoxidizing salt solutions but is readily attacked by sulfur at elevated temperatures.

Fabricating nickel requires special precautions in machining and annealing. Because nickel is strong and tough, high-speed steel or cemented carbide tools in good cutting conditions are required. The work should be well flooded with cutting lubricant and cut at moderate speeds and feeds. Annealing should be done in a sulfur-free atmosphere. Sulfur-bearing fuels or traces of sulfurized cutting oils on the work will cause severe attack on nickel during annealing. Nickel may be fusion welded by oxyacetylene or electric-arc methods or joined by silver brazing alloys (except phosphorous-bearing alloys) or soft soldering.

Silver. Pure silver has the whitest color and the highest electrical and thermal conductivity of all the metals. Pure silver (fine silver) is too soft for many mechanical uses. Hence it is usually alloyed with copper to increase physical strength and hardness. Sterling silver contains 7.5 per cent copper; coin silver, 10 per cent copper. Silver is used in the electrical industry for heavy bus bars and heavy-duty contact and also in the chemical industry because of its resistance to attack by acetic acid, brewery liquid, formaldehyde, phenol, and dilute sulfuric acid. The use of both cast and wrought silver in jewelry is well known. Silver is commercially available in sheet, plate, rod, tubing, and as clad plate, combined with base metals such as copper, nickel, or steel. Fabricated shapes are made by the fusion welding, using atomic-hydrogen torch or Heliarc process. Molten silver readily absorbs oxygen, creating "gassy" welds if improperly made. Annealing at 480°F will soften silver.

Silver Solder. Silver solders or, more properly, silver brazing alloys are perhaps the most convenient materials available for making reasonably high-strength joints between most of the common metals and alloys. Silver brazing alloys include a wide range of compositions containing from 5 to 80 per cent silver, combined with varying amounts of copper, zinc, cadmium, or phosphorus. One of the principal advantages of silver brazing alloys is the free-flowing nature of the melted alloy. Actually, the strongest joints are made with very small clearances. Silver brazing alloys possess about the same corrosion resistance as brass, copper, nickel silver, and monel. The procedure for using silver brazing alloys consists essentially of (1) removing all scale, grease, and dirt; (2) applying the proper brazing flux; (3) maintaining desired alignment of the pieces to be joined; and (4) heating (1200 to 1600°F) by oxyacetylene flame, induction or resistance heating, or by furnace heating. Aluminum or alloys that melt below 1250°F cannot be joined by silver brazing.

Solder. The term "solder" or "soft solder" usually refers to the tin-lead alloys of from 15 to 60 per cent tin. Additions of antimony, bismuth, and cadmium may be made to reduce cost or to obtain special properties. Soft solders are low-strength low-melting alloys (400 to 600°F) used for joining a large number of the common metals and alloys. The higher grades of solder contain the larger amounts of tin. Solder for electrical work may contain 60 to 90 per cent tin; the most common solder for general purposes contains 50 per cent tin. Wiping solder (so named because of the application technique) contains 40 per cent tin; low-grade solder for filling dents in automobile bodies contains 15 per cent tin. During the war emergency, tin was conserved by resorting to high-lead alloys for many soldering applications. The atmospheric corrosion resistance of tin-lead solders is in general satisfactory. Under more severe conditions, or when solder is used on copper or monel, accelerated corrosion may occur. Success in the use of soft solders depends on obtaining a clean surface on the metals to be joined, applying the proper flux, and heating the metal to the melting point of the solder. The heating operation may be accomplished by wiping the joint with semiliquid solder, by use of a soldering iron, blowtorch, solder bath, resistance or induction heating.

Because solders are inherently weak material, soldered seams should be strengthened by crimping, interlocking, or riveting before soldering.

Tantalum. Tantalum is a white, lustrous metal of high specific gravity—16.6—possessing unusual resistance to acid corrosion. It is practically chemically inert toward all materials except hydrofluoric acid and concentrated caustic. It is ductile and malleable and can be drawn into fine wires, rolled into thin sheets, or formed into a variety of sizes of tubing. Tantalum is welded successfully by the electrical-resistance method. Owing to its affinity for gases, the welding is usually carried out either under the surface of carbon tetrachloride or in the presence of a 100 per cent carbon tetrachloride vapor. It possesses the unusual property, owing to its natural inertness, of remaining clean under a variety of chemical service conditions. This permits its use in certain chemical equipment where heat transfer is a paramount factor and where the over-all economics will justify a metal of the high cost of tantalum.

Stainless Alloys. Stainless alloys may be divided, for purposes of approximate classification, into three groups: (1) ferritic alloys, which are essentially iron-chromium alloys and in certain cases with small amounts of nickel and molybdenum, the alloying element amounting up to 30 or 36 per cent of the total and with varying amounts of carbon; (2) austenitic alloys, containing from 16 to 25 per cent of chromium together with 7 to 20 per cent of nickel (these may contain up to 1 per cent columbium as a stabilizing agent); and (3) complex alloys, containing, in addition to the chromium and nickel, varying amounts of molybdenum, silicon, copper, manganese, each alone or in admixtures. All the alloys in these three classifications are quite resistant to oxidizing agents, and certain ones find wide use under corrosive conditions involving sulfuric acid, nitric acid mixtures, and nitrates. Table 25-1 contains a great deal of information that is pertinent to the various grades of stainless alloys.

1. Ferritic alloys are used in nitric acid service and certain general oxidizing service conditions. The composition range in this classification is broad. The low end of the range begins in commercial-alloy composition at 11.50 to 13.0 per cent chromium together with 0.12 maximum carbon and extends upward to an alloy with 30 per cent chromium and 0.35 maximum carbon. All the alloys in this range, *i.e.*, AISI types

Table 25-1. Commercial Stainless Irons and Steels—Analyses, Properties, and Characteristics

American Iron and Steel Institute type Nos.	403	416	430	444	329	446	304	303	302 B
Approximate chemical composition, per cent	C 0.12 (max) Cr 11.50–13.00 Si 0.50 (max) Mn 0.50 (max) S 0.03 (max) P 0.03 (max)	C 0.12 (max) Cr 12.00–15.00 Si 0.50 (max) Mo 0 60 (max) Mn 0.50 (max) S 0.07 (min) P 0.03 (max)	C 0.12 (max) Cr 14.00–18.00 Si 0.50 (max) Mn 0.50 (max) S 0.03 (max) P 0.03 (max)	C 0.08–0.20 Cr 18.00–23.00 Si 0.50–3.50 Mn 0.50 (max) S 0.03 (max) P 0.03 (max)	C 0.01 (max) Cr 25.00–30.00 Ni 3.00–5.00 Mo 1.00–1.50 Si 0.50 (max) Mn 0.50 (max) S 0.03 (max) P 0.03 (max)	C 0.35 (max) Cr 23.00–30.00 Si 0.50 (max) Mn 0.50 (max) S 0.03 (max) P 0.03 (max)	C 0.08 (max) Cr 17.50–19.00 Ni 8.00–9.50 Si 0.75 (max) Mn 0.60 (max) S 0.03 (max) P 0.03 (max)	C 0.08–0.20 Cr 17.50–19.00 Ni 8.00–9.50 Si 0.75 (max) Mn 1.25 (max) Se 0.15–0.30 S 0.03 (max) P 0.03 (max)	C 0.20 (max) Cr 17.50–19.00 Ni 8.00–9.50 Si 2.0–3.0 Mn 1.50 (max) S 0.03 (max) P 0.03 (max)
Physical properties: density gm/cc	7.78	7 76	7.70	7.60	7.75	7.54	7.92	7.92	7.92
lb/in.	0 281	0.280	0.278	0.274	0.279	0.272	0.286	0.286	0.286
Modulus of elasticity, 10^6 psi	29	29	29	29	29	29	28–29	28–29	28–29
Structure	Martensitic	Ferritic, Martensitic	Ferritic	Ferritic	Ferritic-nonmagnetic in hardened state	Ferritic	Austenitic sometimes very slightly magnetic	Austenitic, sometimes very slightly magnetic	Austenitic, sometimes very slightly magnetic
Specific heat, Btu/°F/lb (32–212°F)	0.11	0.11	0.11	0.11	0.11	0.11	0.12	0.12	0.12
Thermal conductivity Btu/(sq ft)(hr)(°F/ft), 200°	14 6	13 3	14.1	......	......	12.1	9 4	9.4	9.0
1000°	16.8	15 2	15.1	......	......	14.1	12.5	12.5	12.1
Linear coefficient of thermal expansion per °F × 10^6, 0–200°F	5 7	5.7	5.4	5.4	......	5 3	9.0	9.0	9.0
0– 600°F	6.1	6.1	5.6	5.6	......	5.5	9.8	9.4	10.0
0–1000°F	6 7	6 7	6 1	6 1	7.0	6 0	10.2	10.2	10.8
0–1500°F	7.0	7 0	6.3	6.4	......	6.2	11.1	11.1	11.1
Melting point, deg F	2720–2790	2700–2750	2710–2730	2700	......	2650–2700	2550–2590	2550–2590	2500–2550
Nominal values of mechanical properties, hardness: Brinell number, annealed	135–170	145–185	145–190	150–196	230	150–190	135–165	130–170	150–180
Heat-treated	180–365	190–350							
Cold-rolled	......	......	185–270	190–270	......	......	180–340	170–340	
Rockwell number, annealed	75–85B	80–95B	80–95B	80–95B	......	80–90B	75–90B	70–85B	80–90B
Heat-treated	30–40C	8–35C	......	......	40C				
Cold-rolled	......	......	90–105B	94–104B	......	......	5–45C		
Ultimate tensile strength, psi, annealed	60,000–80,000	70,000–85,000	65,000–90,000	70,000–90,000	105,000	75,000–95,000	80,000–90,000	80,000–90,000	80,000–100,000
Heat-treated	85,000–190,000	80,000–160,000							
Cold-rolled	......	......	80,000–130,000	90,000–145,000	......	......	100,000–200,000	90,000–150,000	
Yield point, psi, annealed	35,000–45,000	40,000–50,000	40,000–55,000	45,000–55,000	75,000	45,000–60,000	30,000–40,000	30,000–40,000	40,000–50,000
Heat-treated	60,000–150,000	60,000–130,000							
Cold-rolled	......	......	65,000–120,000	65,000–130,000	......	......	50,000–180,000	70,000–130,000	
Per cent elongation in 2 inches, annealed	40–30	35–25	35–20	35–25	20	32–20	65–50	60–40	60–50
Heat-treated	25–15	25–10							
Cold-rolled	......	......	20–2	20–2	......	......	50–4	50–4	
Per cent reduction in area, annealed	75–55	65–60	60–45	60–50	50	55–45	70–50	65–50	75–60
Heat-treated	65–60	60–40							
Impact strength—Izod value, ft lb, annealed	110–80	100–60	60–5	40–5	20–5	5–1	115–70	110–70	120–80
Heat-treated	100–20	100–10							

Ductility, Olsen, in., annealed	0.40–0.30	...	0.43–0.30	...	...	...	0.50–0.40		
Erichsen, mm, annealed	8–7	...	9–7	9–7	...	...	14–11		
Creep strength, psi, for a life of 10,000 hr with 1 per cent elongation, at 1000°F	11,600	...	8,500	...	...	5,500	18,000		
At 1100°F	...	...	5,000	...	...	2,700	12,000		
At 1300°F	1,500	...	1,400	...	...	500	4,000		
At 1500°F	...	...	800						
Tensile strength at elevated temperatures, short time tests, psi, 1300°F	16,000	15,000	15,000	...	...	20,000	36,000		
1500°F	10.000	8,000	8,000	...	...	10,000	17,000		
1700°F	8,500	5,000	5,000	...	...	6,000	11,000		
Heat-treatment, annealing	Furnace cool from 1500–1100°F or air cool from 1350–1400°F	Furnace cool from 1550–1100°F or air cool from 1400–1450°F	Furnace cool from 1550–1100°F or air cool from 1400–1450°F	Air cool or water quench from 1450–1600°F	Quench from 1700°F	Air cool from 1450–1650°F over 3/16 in., water quench	Water quench from 1950–2050°F	Water quench from 1950–2050°F	Water quench from 1950–2050°F
Hardening	Oil quench from 1750–1825°F	Oil quench from 1750–1825°F	May be hardened slightly if quenched from above 1550°F*	May be hardened slightly if carbon approaches 0.20%	Hold at 1350°F for 48 hr and furnace cool	May be hardened slightly if carbon is near the max	Nonhardening	Nonhardening	Nonhardening
Tempering	Draw 400–1300°F†	Draw 400–1300°F†	...	...	...	...	Nonhardening	Nonhardening	Nonhardening
Forging	Start 2100°F, finish at 1500°F	Start 2100°F, finish below 1500°F. Cold finish very important.	Start 2000–2050°F, finish 1400°F. Cold finish very important	Start 1900–2050°F, finish 1300–1600°F	1800–2000°F	Start 2050°F., finish 1500°F. Cold finish very important	Start at 2250°F, finish above 1700°F	Start 2250°F, finish 1750°F	Start 2300°, finish 1700°F
Heat resistance, scaling temperature, continuous service	1300°F	1300°F	1500°F	1800°F	About 2000°F	2000°F	1650°F	1650°F	1700°F
Intermittent service	1400°F	1400°F	1600°F	1900°F	...	2100°F	1500°F	1500°F	1600°F
Welding properties	Fair, generally should be annealed‡	Poor, brittle welds. Likely to crack. Usually not recommended	Fair, brittle welds. Generally should be annealed	Fair, brittle welds. Generally should be annealed	Fair, brittle welds. Generally should be annealed	Fair, brittle welds. Generally should be annealed	Very good. Tough welds	Fairly good. May crack. Usually not recommended	Good, tough welds.
Machining properties	Fair	Good. About 80–85 per cent compared to screw stock	Fair, tough	Fair	Fair	Fair, tough	Fair	Good. About 65 per cent of screw stock	Fair, tough
Drawing or stamping	Fair	Poor	Good	Good	Good	Poor	Very good	Fairly good	Good

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* Hardenability decreases as (a) chromium content increases and (b) carbon content decreases.

† Embrittlement and loss of corrosion resistance occur if drawn in range of 900° to 1100°F.

‡ Welding characteristics improved by presence of certain elements, such as Ti and Cb, which render the material non-air-hardening.

Table 25-1. Commercial Stainless Irons and Steels—Analyses, Properties, and Characteristics (*Cont.*)

American Iron and Steel Institute type Nos.	321	347	316	317	309	310	311	325
Approximate chemical composition, per cent	C 0.10 (max) Cr 17.00–19.00 Ni 7.00– 9.50 Ti 5 times C Si 0.75 (max) Mn 0.60 (max) S 0.03 (max) P 0.03 (max)	C 0.10 (max) Cr 17.00–19.00 Ni 8.00–12.00 Cb 10 times C Si 0.75 (max) Mn 0.60 (max) S 0.03 (max) P 0.03 (max)	C 0.10 (max) Cr 16.00–19.00 Ni 9.00–14.00 Mo 2.00–3.00 Si 0.75 (max) Mn 0.50–2.00 S 0.03 (max) P 0.03 (max)	C 0.10 (max) Cr 18.00–20.00 Ni 12.00–14.00 Mo 3.00–4.00 Si 0.50 (max) Mn 1.00–1.75 S 0.03 (max) P 0.03 (max)	C 0.20 (max) Cr 22.00–26.00 Ni 12.00–14.00 Si 0.60 (max) Mn 0.50–2.00 S 0.03 (max) P 0.03 (max)	C 0.25 (max) Cr 24.00–26.00 Ni 19.00–21.00 Si 0.50–2.00 Mn 1.50 (max) S 0.03 (max) P 0.03 (max)	C 0.25 (max) Cr 19.00–21.00 Ni 24.00–26.00 Si 0.50–3.00 Mn 1.50 (max) S 0.03 (max) P 0.03 (max)	C 0.25 (max) Cr 7.00–10.00 Ni 21.00–23.00 Cu 1.00–1.50 Si 0.50–1.50 Mn 0.50–1.50 S 0.03 (max) P 0.03 (max)
Physical properties: density, gm./cc	7.94	7.94	8.02	8.04	7.94	7.89	7.84	7.98
lb/in	0.287	0.287	0.290	0.291	0.287	0.287	0.283	0.289
Modulus of elasticity, 10⁶ psi	28–29	28–29	28–29	28–29	28–29	28–29	28–29	28–29
Structure	Austenitic, slightly magnetic	Austenitic, slightly magnetic	Austenitic, slightly magnetic	Austenitic, slightly magnetic	Austenitic, sometimes very slightly magnetic	Austenitic	Austenitic	Austenitic
Specific heat, Btu/°F/lb (32–212°F)	0.12	0.12	0.12	0.12	0.12	0.12	0.12	
Thermal conductivity Btu/(sq ft)(hr)(°F/ft), 200°	9.3	9.2	9.0	...	7.7	8.5		
1000°	12.8	12.5	12.3	...	9–4			
Linear coefficient of thermal expansion per °F × 10⁶, 0–200°F	8.8	8.8	9.0	...	8.3	8.0	8.0	9.3
0– 600°F	9.2	9.4	9.8	...	8.6	8.5	8.5	10.0
0–1000°F	10.3	10.2	10.5	...	9.6	9.2	9.4	11.1
0–1500°F	11.3	11.3	11.0	...	10.8	10.6	10.0	12.1
Melting point, deg F	2550–2590	2550–2600	2500–2550	2500–2550	2530–2570	2500–2600	2650–2700	2600–2700
Nominal values of mechanical properties, hardness: Brinell number, annealed	135–185	135–185	135–200	135–185	140–185	140–210	160–190	160–190
Heat-treated								
Cold-rolled	...	...	180–300	...	170–375			
Rockwell number, annealed	70–90B	70–90B	70–90B	...	82–90B	80–95B	84–92B	84–92B
Heat-treated	...	...		...				
Cold-rolled	...	...	10–30C	...	5–40C			
Ultimate tensile strength, psi annealed	80,000–95,000	80,000–95,000	80,000–95,000	80,000–95,000	90,000–110,000	90,000–110,000	90,000–110,000	85,000–95,000
Heat-treated								
Cold-rolled	...	...	100,000–200,000	100,000–200,000	110,000–200,000			
Yield point, psi, annealed	35,000–45,000	35,000–45,000	35,000–50,000	35,000–55,000	40,000–60,000	40,000–60,000	45,000–50,000	45,000–55,000
Heat-treated								
Cold-rolled	...	...	50,000–150,000	50,000–150,000	65,000–150,000			
Per cent elongation in 2 in., annealed	60–50	60–50	70–50	70–50	60–40	55–45	40–30	40–30
Heat-treated								
Cold-rolled	...	...	50–10	50–5	35–5			
Per cent reduction in area, annealed	65–55	65–55	70–60	75–60	65–40	60–40	45–35	55–45
Heat-treated								
Impact strength—Izod value, ft lb annealed	Charpy 77	...	120–70	...	...	Charpy 80–40	90–50	Charpy 80
Heat-treated								

Approximate chemical composition, per cent	C 0 10 (max) Cr 17 00–19 00 Ni 7 00– 9 50 Ti 5 times C Si 0 75 (max) Mn 0 60 (max) S 0 03 (max) P 0 03 (max)	C 0 10 (max) Cr 17 00–19 00 Ni 8 00–12 00 Cb 10 times C Si 0 75 (max) Mn 0 60 (max) S 0 03 (max) P 0 03 (max)	C 0 10 (max) Cr 16 00–19 00 Ni 9 00 14 00 Mo 2 60 3 00 Si 0 75 (max) Mn 0 50– 2 00 S 0 03 (max) P 0 03 (max)	C 0 10 (max) Cr 18 00–20 00 Ni 12 00–14 00 Mo 3 00– 4 00 Si 0 50 (max) Mn 1 00– 1 75 S 0 03 (max) P 0 03 (max)	C 0 20 (max) Cr 22 00–26 00 Ni 12 00–14 00 Si 0 60 (max) Mn 0 50–2 00 S 0 03 (max) P 0 03 (max)	C 0 25 (max) Cr 24 00–26 00 Ni 19 00–21 00 Si 0 50– 2 00 Mn 1 50 (max) S 0 03 (max) P 0 03 (max)	C 0 25 (max) Cr 19 00–21 00 Ni 24 00–26 00 Si 0 50– 3 00 Mn 1 50 (max) S 0 03 (max) P 0 03 (max)	C 0 25 (max) Cr 7 00–10 00 Ni 21 00–23 00 Cu 1 00– 1 50 Si 0 50– 1 50 Mn 0 50– 1 50 S 0 03 (max) P 0 03 (max)
Ductility, Olsen, in , annealed								
Erichsen, mm , annealed								
Creep strength, psi, for a life of 10 000 hr with 1 per cent elongation, at 1000°F			23,000		17,000	17,000	17,000	18,000
At 1100°F			14 000		13,000	13,000	10 500	9,500
At 1300°F			5 000		4,800	4,500	5 000	4,200
At 1500°F			2,000		850	1,000		850
Tensile strength at elevated temperatures short time tests psi, 1300°F	36,000		42,000		54,000	51,000	50,000	55,000
1500°F	17 000		25 000		32,000	34,000	25 000	40,000
1700°F	11,000		18,000		16 000	16,000	12 000	18,000
Heat-treatment, annealing	Water quench from 1950–2050°F for max softness Welding stock should be held at 1550–1650°F 4 hrs for max corrosion resistance	Water quench or air cool from 1950–2050°F Stress relieving Air cool from 1550°F min	Water quench from 1950–2050°F	Water quench from 1950–2050°F	Water quench from 1950–2050°F	Water quench from 1950–2050°F	1850°F Air cool for thin sections Water quench for heavier sections	1900°F Air cool for thin sections. Water quench for heavier sections
Hardening	Nonhardening	Nonhardening	Nonhardening	Nonhardening	Nonhardening	Nonhardening	Nonhardening	Nonhardening
Tempering	Nonhardening	Nonhardening	Nonhardening	Nonhardening	Nonhardening	Nonhardening	Nonhardening	Nonhardening
Forging	Start 2250°F finish 1700°F	Start 2250°F, finish 1800°F	Start 2200°F finish 1750°F	Start 2200°F, finish 1750°F	Start 2250°F, finish 1800°F	Start 2250°F, finish 1800°F	Start 2100°F, finish below 1900°F	Start 2050°F, finish above 1600°F
Heat resistance, scaling temperature, continuous service	1650°F	1650°F	1650°F	1650°F	2000°F	2000°F	2000°F	1800°F
Intermittent service	1500°F	1500°F	1500°F	1500°F	1900°F	1900°F	1900°F	1500°F
Welding properties	Very good 18-8-Cb rod usually recommended	Very good	Very good	Very good	Very good	Good	Good	Good
Machining properties	Fair tough	Fair, tough	Fair, tough	Fair, tough	Fair	Fair, tough	Fair, tough	Fair
Drawing or stamping	Good	Good	Fair	Poor	Good	Good	Fair	Good

403, 416, 430, 444, 329, and 446, are weldable with some difficulty and require a prescribed heat-treatment subsequent to welding in order to restore full stainless qualities. The handling, fabrication, and heat-treatment of these grades may be carried out on most plants, but previous experience is necessary for best results. The alloys in this classification are usually available in a wide variety of commercial shapes and find application in chemical plants within the scope of the physical and chemical properties of the materials.

2. Austenitic alloys are widely used in nitric acid service and certain general oxidizing service conditions. Austenitic alloys in the more conventional compositions contain the essential chromium and nickel (see Table 25-1 for types 304, 303, 302B, 316, 317, 309, 310, 311, 321, 325, and 347; types 321 and 347 contain addition agents, titanium and columbium, respectively, which have a marked effect on the heat-treating procedures and resulting properties). The welding properties of the austenitic alloys are much better than those of the ferritic types. The increased alloy content of the austenitic types results, under the majority of service conditions, in an increased resistance to corrosion. Several of the austenitic grades overlap in application, but certain ones, such as types 316 and 317, possess particular resistance to acetic acid hot and cold, with or without small amounts of sulfuric acid. Tests should always be run in advance under the specific service conditions. All the above compositions referred to are available commercially in some forms. However, the number of shapes is sometimes limited. If the amount required is less than enough to warrant a special processing lot, it is desirable to select a suitable alternate material on the basis of the properties and characteristics of other austenitic materials described in Table 25-1.

3. Complex austenitic alloys for sulfuric acid and certain sulfuric-nitric mixtures are closely related to the conventional austenitic alloys and differ principally in the increased amounts and number of metals added. These constitute a large number of complicated alloys, both cast and wrought. The addition of molybdenum and copper, usually together with manganese and silicon, to the conventional chromium-nickel austenitic alloys imparts increased corrosion resistance, particularly under corrosive conditions involving sulfuric acid, sulfites, and sulfates. Numbers of commercial alloys of this general type are available in cast and wrought forms. Often these alloys are made up into standard equipment such as pumps, valves, centrifuges, and filter frames by the manufacturer and are recommended by the producers for specific plant service conditions. Notwithstanding this confidence on the part of the manufacturer, it is desirable and even necessary that the customer conduct corrosion tests under the specific plant conditions in order to ensure that no unforeseen condition exists which will shorten the life of the equipment.

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SECTION 26

TRADE WASTE DISPOSAL

BY H. L. JACOBS

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TRADE WASTE DISPOSAL*

BY H. L. JACOBS

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INTRODUCTION

This section is a general guide to the problem of proper treatment and disposal of industrial wastes and not a discussion of the solution to specific problems. Rarely are two industrial waste problems alike in all details, and each one should, therefore, be handled as an individual situation.

The field of industrial waste treatment is relatively new compared with the amount of study that has been devoted to the treatment of sanitary sewage. It is true that a considerable number of problems have been solved to the satisfaction of pollution-control authorities, but many more exist which have not been studied or for which reasonable treatment methods have not been developed. This situation is partially due to the feeling on the part of most plant managers that waste treatment will add to the cost of their products and should, therefore, be avoided as long as possible. While this may be true in many cases, it is also true that many waste problems have been solved by the recovery of materials at a profit. As a matter of fact, in a few instances, the value of the recovered material has become extremely important in reducing the manufacturing cost and hence the selling price of the primary product.

The time is rapidly approaching when all industries and municipalities will be required to meet rigid specifications for the quality of their water-borne wastes. It is to be hoped that these specifications will permit *reasonable* amounts of contamination to be handled by the self-purification abilities of the receiving waters. The movement to clean up the nation's streams has been progressing during the past 50 years. Legislative activity in the field of stream-pollution prevention has become vigorous since 1940. During the years 1943 and 1944, in spite of the burdens imposed by the Second World War, stream-pollution legislation was introduced in 27 of the states. In 12 of these, the existing laws were amended, while the states of Tennessee, Washington, Minnesota, and North Carolina covered stream pollution in their statutes for the first time. The state of Virginia has since followed suit. The creation of several interstate commissions, with the primary object of reducing the pollution throughout large drainage areas, has resulted. Such commissions are now functioning in connection with the Delaware, Potomac, and Ohio River basins.

On many plants provisions have not been made for the segregation and treatment of sanitary wastes from office buildings and change houses. In too many instances no thought of the ultimate necessity for treatment of these wastes was given at the time the sewerage system was laid out, and as a result, sanitary and industrial wastes are mixed indiscriminately prior to discharge. In certain cases the sanitary sewage can be treated with the industrial wastes to advantage, but in many others the proper procedure will necessitate segregation and separate treatment. The methods for treatment of sanitary sewage have become standardized, and no discussion of them is contemplated here. Advice should be obtained from the appropriate health agency before the problem is attacked. Nearly all states require that plans be submitted for approval to the state health department.

DEFINITION OF POLLUTION

A stream becomes polluted when waste matter is introduced in sufficient amounts to alter the water quality so as to make it unsuitable for domestic or industrial supply or for the proper support and growth of marine life. This may occur because of the presence of dissolved salts or organic matter, materials carried in suspension, floating oils, acids, alkalies, or toxic substances.

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* Superior numbers indicate specific references listed at the end of this section.

EFFECTS OF VARIOUS WASTES

Acids and Alkalies. Mineral acids such as sulfuric, nitric, and hydrochloric will neutralize the natural alkalinity of the receiving waters with a resultant lowering of the pH value. Below pH 5.0, the more delicate fish such as trout and bass will be adversely affected, while at values below pH 4.0 nearly all types will die. At the same time, the effect on the food supply of the fish will be adverse owing to the decrease or death of marine organisms. Sulfuric acid or sulfates will increase the permanent hardness of the receiving waters, while hydrochloric acid or chlorides may render the water nonpotable even though the proper pH is maintained by the addition of alkali. The presence of mineral acidity in a stream creates a sterilizing effect which interferes with normal biological activity tending to purify the water of its organic content.

Any discharge of alkali which increases the pH value of the receiving waters above 9.0 will also prove detrimental to fish life.

Suspended Solids. Solids in suspension in a plant effluent may prove detrimental whether inorganic or organic in character. If they are heavy enough to settle out in the relatively low velocity of the receiving waters, bottom life will suffer, and the stream characteristics will be altered. If the settleable material is organic in nature, anaerobic decomposition will take place with the production of hydrogen sulfide and methane gases with consequent foul odors. In navigable waters, channels will be filled up, and dredging will be required more frequently.

Floating Matter. A common complaint in regard to certain effluents is the presence of floating matter Greases, oils, and tars are sometimes discarded to the sewer system and eventually show up as contamination in the receiving waters. These materials, particularly oils, spread out in very thin films on the water surface and interfere with the normal oxygen absorption from the atmosphere. Usually they are stable compounds which oxidize slowly, and the effect is, therefore, of long duration. Fish life may be damaged either by suffocation or at least by the absorption of tastes and odors in the flesh so that its value as human food is impaired. Floating oils will also be damaging to the paint on boats. In general, the source of such pollution is easily traced and may become a beacon which will lead the authorities to other sources of contamination.

Organic Matter. The organic matter present in clean and polluted streams is oxidized by the dissolved oxygen in the water with the formation of simpler compounds such as water, carbon dioxide, sulfates, and nitrates. This process of oxidation takes place with the assistance of bacteria which are present in most receiving waters. These bacteria are introduced by the discharge of raw or unsterilized sewage or by pickup of soil bacteria by storm waters. If the quantity of organic matter is small, the rate of reduction of dissolved oxygen will be overcome by the normal reaeration of the water. When this is the case and the dissolved oxygen remains above 5 ppm, no damage results, and the water is not considered to be seriously polluted. In many cases, however, the organic load is sufficient to cause serious depletion of the dissolved oxygen. At oxygen values below 5 ppm, the finer types of game fish cannot exist satisfactorily, and at values below 4 ppm, nearly all fish life will be affected adversely. When the dissolved oxygen content drops to 2 ppm and below, nuisance conditions are created, and foul odors are given off owing to anaerobic decomposition of the organic matter.

The term in general use for representing the effect of organic matter in the depletion of dissolved oxygen is known as the "biochemical oxygen demand," usually contracted to BOD. The BOD of a given effluent can be measured by means of a standard test which has been developed for sewage effluents. This same test can be applied to industrial wastes. The test consists of diluting to various degrees samples of the waste water with a well-aerated dilution water which has been "seeded" with a small quantity of sewage. The several dilutions, together with the required "blanks," are incubated at 20°C for 5 days, at which time the dissolved oxygen is determined by standard titration methods. The difference in oxygen content of the samples and "blanks" is then calculated to parts per million in the original waste and reported as BOD.[1]

Toxic Materials. Many materials that are toxic to humans and other animals are also detrimental to marine and bacterial life. Metallic or alkali cyanides in concentrations as low as 0.2 to 0.3 ppm will kill fish and will interfere with and sometimes completely inhibit digestion of sewage solids in sewage-treatment plants.

Soluble copper compounds are effective sterilizing agents and in concentrations as low as 0.25 ppm will be lethal to fish life. Sanitary sewage containing as much as 1.0 ppm of copper cannot be treated satisfactorily by biological processes because of the sterilizing effect on the bacteria.

The list of compounds that are detrimental to fish and bacterial life is a long one and includes such materials as hydrogen sulfide, chlorine, barium chloride, chromic acid, aldehydes, benzene, gasoline, and many others. Much information on the allowable limits for various compounds with respect to fish life is contained in a government publication,[2] and a large amount of work on fish mortality has also been carried out by the Michigan Experiment Station.

Taste and Odor Producers. Some compounds, usually organic in nature, are capable of producing serious taste and odor nuisances in potable water supplies. Of these the most common are phenol and some of its derivatives contained in wastes from coke-oven operations and certain other chemical processes. Phenol by itself can be tolerated by the human system to the extent of several parts per million, but upon chlorination the presence of a few parts per billion makes the water unpalatable. Many other compounds will impart undesirable tastes or odors which can be classed as "musty," "saline," "earthy," etc. It is frequently necessary for the water-plant operator to resort to excessive chemical dosage or to treatment of the water with activated carbon in order to produce a palatable water. Frequently the water consumer must put up with poor-quality water for a time because of the inability of the water-plant operator to eliminate completely the taste and odor.

PLANT SURVEY OF WATER-BORNE WASTES

Measurement of Volume. Importance of accurate measurement of the volume of the contaminating effluent cannot be stressed too greatly. Without such measurements, very little progress can be made in the study of the proper treatment method to be applied or of the quantity that may be discharged without damage to the receiving waters.

The method used to obtain accurate volumetric measurements will vary with the physical nature of the plant and process. In cases where the operation is compact and effluents are all collected and discharged through a single sewer, meter readings of the plant water consumption will sometimes suffice. In a complicated layout with several effluents entering a common sewer or with a sewer system split up into multiple outlets, it will be necessary to devise special methods for accurately determining the individual and over-all flows.

Determination of flow in an underground sewer is difficult to carry out. If the size and slope of the sewer are known, charts[3] are obtainable that will give the flow at various depths. If a location can be selected at which the flow level is unaffected by turbulence or velocity increase at the entrance to a manhole, depth measurements will provide reasonably accurate flow rates. For continuous readings, a low-pressure recorder can be used to record the pressure required to bubble air through the liquid in the sewer. Pressure recorders capable of reading less than 1 in. of water pressure can be obtained. The air stick used for this purpose should be carefully made so that the air opening will not be constricted at the bottom of the sewer. It is a good idea to place the opening at a known distance from the bottom of the pipe and to correct the readings accordingly so that the pipe can be rested on the inside bottom of the sewer at the exact center line. In using this method it is important to know that the sewer line ahead of the point of measurement is free of sediment or obstructions.

Any number of other standard methods of flow measurement can be applied when the proper conditions prevail. For individual discharges from pipes under pressure, some type of orifice plate may be suitable. For sewer lines, a weir box with a square, V-notch, or Cippoletti weir may be applicable. In order to obtain accuracy with a weir, it is necessary to observe carefully the dimensional requirements so that the *vena contracta*, approach velocity, and other factors will be correct. If measurements are to be made of an effluent containing appreciable solids in suspension, the installation should be made with a gate or opening at the bottom of the weir plate, or the weir plate should be removable so that the solids may be flushed out from behind it. Perhaps the biggest disadvantage of the weir is the relatively high head loss which must be overcome.

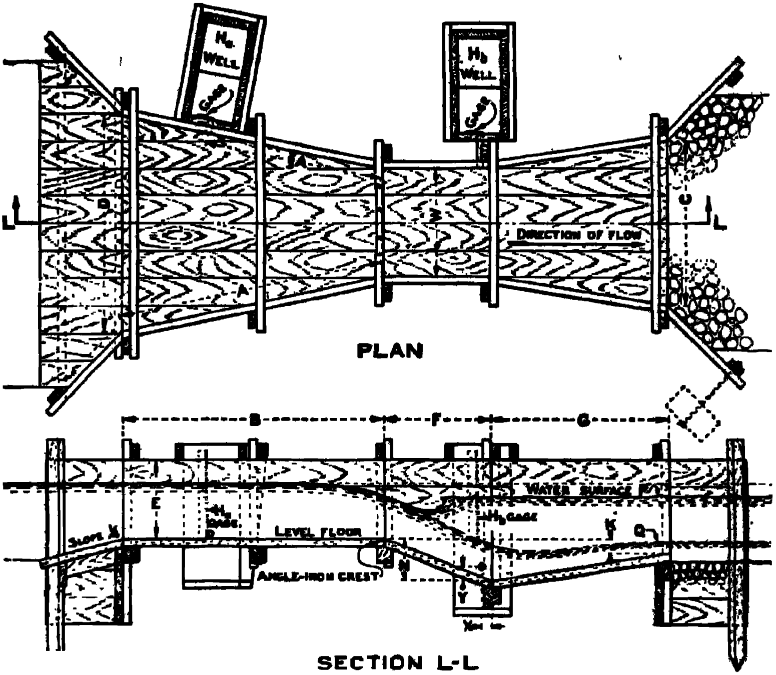

Fig. 26-1. Parshall flume.

The most satisfactory flow-measuring device for installation in a sewer line is the "Parshall measuring flume" (see Fig. 26-1). This flume was developed and tested by R. L. Parshall at the Colorado Experiment Station. It was developed primarily for measuring flows in irrigation ditches, but it is finding increasing use in sanitary sewage and industrial-waste-treatment plants. Its accuracy is sufficient to meet practical needs, and there is no point at which solids can collect and settle. Head loss is lower than for any other type of open measuring device. Flume liners made of sheet iron can be purchased in almost any required size. Fabrication of a liner from stainless-steel sheet which can be electric-welded is relatively simple. The easiest method for constructing the flume is to use the liner as the inside form for a concrete structure, leaving it in place for permanent use. For temporary use, wood construction is satisfactory. Complete details of construction and operation can be found in Reference 4.

Sampling. During the initial stages of a waste-disposal problem, grab samples are helpful for a quick analysis of the situation and for guides to a more careful sampling program. In most cases it is necessary to establish sampling procedures that will yield representative samples. If the problem is complicated, it may prove necessary for sampling to extend over a considerable period of time, and it will be most practicable to install continuous sampling equipment. It is beyond the scope of this section to cover the various means for obtaining such samples, but the equipment should be designed so that sampling rates are correlated with the flow. Where a substantial head loss can be taken, various types of splitting devices can be installed which will divert a small proportion of the total flow to a suitable container. For use with weirs or Parshall flumes, mechanical rotating samplers which will dip proportionate samples from the head chamber and lift them to a receiver can be purchased or built. One such device is known as the "Trebler sampler." Proportioning pumps can also be used in conjunction with flowmeters to draw off representative samples.

Table 26-1. Dimensions and Capacities of the Parshall Measuring Flume for Various Throat Widths, *W*

(Letters refer to dimensions, Fig 26-1)

W		*A*		⅔*A*		*B*		*C*		*D*		*E*		*F*	*G*	*K*	*N*	*X*	*Y*	Free-flow capacity	
																				Max	Min
Ft	In.	Ft	In	Ft	In.	Ft	In	Ft	In	Ft	In	Ft	In	Ft	Ft	In.	In	In.	In	Sec-ft	Sec-ft
	3	1	6 3/8	1	1/4	1	6	0	7	0	10 3/16	1	4	1/2	1	1	2 1/4	1	1 1/2	1 1	0 03
	6	2	7/16	1	4 5/16	2	0	1	3 5/8	1	3 5/8	2	0	1	1 1/2	3	4 1/2	2	3	3 9	0 05
	9	2	10 5/8	1	11 1/8	2	10	1	3	1	10 5/8	2	6	1	2	3	4 1/2	2	3	8 8	0 09
1	0	4	6	3	0	4	4 7/8	2	0	2	9 1/4	3	0	2	3	3	9	2	3	16 1	0 35
1	6	4	9	3	2	4	7 7/8	2	6	3	4 3/8	3	0	2	3	3	9	2	3	24 6	0 51
2		5	0	3	4	4	10 7/8	3	0	3	11 1/2	3	0	2	3	3	9	2	3	33 1	0 66
3		5	6	3	8	5	4 3/4	4	0	5	1 7/8	3	0	2	3	3	9	2	3	50 4	0 97
4		6	0	4	0	5	10 5/8	5	0	6	4 1/4	3	0	2	3	3	9	2	3	67 9	1 26
6		7	0	4	8	6	10 3/8	7	0	8	9	3	0	2	3	3	9	2	3	103 5	2 63
8		8	0	5	4	7	10 1/8	9	0	11	1 3/4	3	0	2	3	3	9	2	3	139 5	4 62

Analyses. It is rarely necessary to analyze a particular plant effluent for all the constituents listed below, but it is wise to have several complete analyses made of the over-all plant discharge so that the effect of the wastes on the receiving waters can be accurately estimated. In extreme cases of pollution, it may also be necessary to sample and analyze the receiving waters above and below the point of discharge. Usually, however, the necessity for correction of a waste discharge is created by a complaint from a downstream consumer, by fish mortality, or by an obviously unsatisfactory condition below the outfall, and the offending contaminant can be traced by the effect it creates. The analytical tests should include any of the following that will help in locating it:

1. Solids, suspended
2. Solids, total
3. Solids, volatile
4. Solids, fixed
5. pH
6. Acidity or alkalinity
7. Color
8. Grease or oil
9. Biochemical oxygen demand (BOD)
10. Oxygen-consumed value

Although the test for "oxygen-consumed value" has been used for years, no correlation exists between the values obtained in the test and the oxygen consumption that the waste will exert on the receiving water. The test for BOD is to be preferred and is rapidly becoming the criterion of organic pollution used by stream-control authorities. The "oxygen-consumed" test can be used to advantage in rough determinations of the dilution ratios that should be used in performing the BOD test.

In carrying out the analytical work the methods outlined in "Standard Methods for the Analysis of Water and Sewage," published by the American Public Health Association, should be followed. The BOD test is the only one of the above in which special apparatus and techniques are involved. With certain wastes it is found that chemicals are present that are toxic to bacteria, and consequently, the correct BOD will be difficult or impossible to determine. In such cases, the wastes must be investigated for the presence of known bactericides.

WASTE RECOVERY AND UTILIZATION

Although no two waste-disposal problems are alike in all respects, the approach to their solution should start along recovery lines. Frequently the recovery of values will prove to be a relatively simple matter when a thorough sampling and analytical program has located the major sources and points of discharge of the wastes.

The first step in the recovery program should be to check every possible outlet where wastes can enter the sewer system inadvertently. This check should include such obvious points as pump glands, pipe joints, and improperly tightened seals on pressure equipment. Frequently it will be found that a general tightening up or the elimination of "sloppy" operating habits will result in greatly improved conditions. Sometimes materials are deliberately thrown into the sewer when they could just as well be returned to process or put through a special treatment to convert them to a salable though substandard product. In any plan for general improvement of plant practices, it is wise to institute an educational program for supervising and operating personnel to advise them that an investigation of means for reducing contamination is under way and of the steps that have been taken in furthering the program. Possibly some valuable and usable suggestions will be forthcoming from both groups. In so far as preventable contamination is concerned, the cooperation of the operators is essential.

After the proper steps have been taken to decrease to a minimum the preventable waste, a thorough study of the process or processes responsible for the remaining contamination should be made. In addition to an engineering analysis of the process to determine optimum operating conditions with respect to losses, yield studies should be made under all feasible operating conditions in order to determine whether or not changes in temperatures, time cycles, reflux ratios, concentrations, etc., will result in decreased losses to the sewer. Equipment is often operated for such long periods at high throughputs that the rates for which it was designed are forgotten. While it may be economical to run certain equipment at higher than rated capacities, it is usually done at the expense of yields. It may be cheaper to enlarge the equipment, thereby reducing losses, than to install collecting and treating equipment to condition the wastes prior to discharge.

Wastes that cannot be eliminated at the source can sometimes be recovered for reuse or for sale on the outside. Sulfuric acid can be recovered economically by concentration if it is available at strengths of about 20 per cent H_2SO_4 or higher. If a use can be developed for a strength lower than that of commercial grades, strengths under 20 per cent can be considered for recovery possibilities. Three commercial acid concentrators are in general use today: the Simonson-Mantius vacuum evaporator, the du Pont falling film concentrator, and the Chemico drum-type concentrator. The first two operate under vacuum and are heated indirectly with high-pressure steam, although Mantius units utilizing hot oil for heating are available. The Chemico concentrator operates at atmospheric pressure with concentration being obtained by the passage of hot combustion gases through the acid. All these concentrators will successfully produce sulfuric acid of 92 to 93 per cent H_2SO_4 content. The falling film unit is applicable to the recovery of relatively clean acids, while either the Mantius or the Chemico will handle sludge acids satisfactorily. In calculating the economics of recovery by concentration, it should be recognized that acids lower in strength than that mentioned above can be considered when neutralization is the alternative since the cost of the latter will be avoided. If the sulfuric acid is contaminated with inorganic sulfates, these will usually precipitate at some intermediate strength and can sometimes be removed by settling, centrifuging, or filtration. If the removed salts have to be washed, the washings can be returned to the concentrator. If undesirable organics are present, they will sometimes be destroyed during concentration. If not, this result can usually be accomplished by holding the final acid at a high temperature for a required time. Destruction of the organic matter results in the decomposition and consequent loss of a certain amount of the sulfuric acid.

The usual method for recovering a usable grade of nitric acid is to mix it with concentrated or fuming sulfuric acid and then to distill the nitric acid. The sulfuric acid retains the water which is then removed from it by the usual methods of concentration. Sufficient sulfuric acid must be used to yield an acid of 70 to 72 per cent H_2SO_4 after nitric removal.

Methods for the recovery of hydrochloric acid are more complicated since dilute acids must be concentrated to the constant boiling mixture of 20.24 per cent and then that mixture distilled and condensed.

Organic acids are usually recovered by distillation or fractionation. In some cases it may be possible to saponify the acid with caustic and to effect a separation of the soap. The organic acid may then be recovered from the soap by treatment with a mineral acid such as sulfuric. It may also be possible to form a calcium soap of the organic acid which will separate more easily than the sodium soap. Organic acids are often a troublesome source of BOD, and neutralization does not correct that undesirable feature.

The conversion of the waste acid to a useful mineral salt should receive consideration before thoughts of recovery are abandoned. Satisfactory outlets can sometimes be developed for such common salts as ammonium nitrate, ammonium sulfate, ammonium chloride, sodium nitrate, and sodium sulfate. The concentration and crystallization of the neutral salt may present less of a problem than the recovery of the corresponding acid.

TREATMENT METHODS

Removal of Suspended Solids. The type of solids contained in the waste discharge will determine the line of attack to be followed in correcting the quality of the effluent. If it is a valuable or useful material and is not badly contaminated with other matter, steps should be taken to retrieve it before it reaches the receiving waters or even before it becomes mixed with other plant wastes. The process from which the material is lost should be examined to determine whether or not the recovered solids can be safely returned to it at some point. Obviously, plain sedimentation should be given primary consideration. Simple laboratory tests will provide sufficient information to determine such possibilities. If the results are promising, consideration should then be given to the type of settling tank that will be most suitable. The most important problem, aside from space requirements, is the one of selecting the proper solids-collecting and -removal equipment. If the solids are relatively heavy and will not be adversely affected by violent agitation, a centrifugal pump may be used for removing the underflow from the sedimentation tank, and the type of collecting mechanism is not important. If the recovered solids require careful handling, equipment should be selected that will not subject them to abrasive action. Plunger and diaphragm pumps are better selections for such service. Standard sedimentation equipment is manufactured by such companies as Dorr, Permutit, Link-Belt, Chain Belt, and others. Should it be desired to concentrate the solids still further, consideration should be given to filtration or centrifuging, depending on the nature of the solids. In many instances, filtration or centrifuging of the unsettled waste will prove effective.

If a clear effluent is required and if it cannot be obtained by sedimentation, flocculating agents such as alum or iron salts may prove helpful. Selection of the proper agent may be dictated by the nature or degree of contamination which the recovered solids can stand before their value as recovered material is lost. The selection of the best coagulating agent is again a matter for laboratory tests to decide. Alum will usually work well in the pH range of 6.0 to 8.0 but has the disadvantage of creating a rather light floc which does not settle too rapidly. Iron salts such as ferric sulfate, ferric chloride, and chlorinated copperas have the advantages of weight and a somewhat wider pH range for successful application.

In some cases, the solids in the effluent will be of such a nature that recovery is not warranted. When conditions require a clear effluent, these cases can be most easily solved by the use of a settling pond or lagoon of the largest practical size. If low-lying ground is available, it may be surrounded by an earth dike to form the pond. It is a good idea to divide the pond into two equal parts with a center dike, so that one-half of it can be drained and cleaned when necessary.

Neutralization of Acids and Alkalies. After all recovery possibilities have been investigated, there remain those wastes which must be neutralized prior to disposal. If the material is an alkali, the problem is relatively simple since sulfuric acid is the cheapest acid for such a purpose and can be conveniently shipped, handled, and fed in standard equipment. If the resulting neutral salt is insoluble at the concentration employed, additional dilution should be introduced, or provisions should be made to remove the precipitate from the effluent.

In the neutralization of waste acids, the cheapest form of alkali which can be purchased will be high-calcium or dolomitic limestone, hydrated lime, or quicklime. Typical analyses of good grades of high-calcium and dolomitic limestones are given in Table 26-2. When these stones are calcined to make quicklimes and are then hydrated, the calcium and magnesium ratios are unchanged. Hydration of a dolomitic quicklime does not result in a mixture of calcium and magnesium hydroxides, however, because the magnesium does not hydrate to an appreciable extent at atmospheric pressure. Whether for this or some other reason, it is a fact that dolomitic limes and limestones require a longer time to react with acids than do the high-calcium varieties.

Table 26-2

	High-calcium	Dolomitic
Calcium carbonate ($CaCO_3$), per cent	94.5	54.0
Magnesium carbonate ($MgCO_3$), per cent	1.0	42.0
Iron and aluminum oxides, per cent	1.0	1.0
Acid insoluble, per cent	3.5	3.0
Total, per cent	100.0	100.0

Limestone Beds. With certain problems, limestone beds for acid neutralization are convenient, successful, and highly desirable. The flow of solution may be upward or downward through the bed. In any case, the tank or structure confining the bed should be provided with a false bottom or free space below the bed where unreacted limestone and inert matter may collect. This free space must be flushed out periodically to remove the sediment. The acid solution must be relatively clear in order to prevent clogging of the bed. The depth of limestone in the bed will depend on the flow rate through it and the pH value desired in the effluent. If the proper precautions are taken in sizing the installation, excellent results can be attained with very little operating attention. No control equipment is required, and flow rates need not be controlled carefully if the bed has been designed to accommodate the maximum. The pH of the effluent will rarely exceed 5.0, but proper aeration of the effluent to drive off CO_2 will result in pH 7.0 or higher.

Limestone beds are ideally suited to the neutralization of *hydrochloric* and *nitric* acids when the quantities are not too great. When large quantities are to be handled, the size of the bed approaches the point where special design is entailed to ensure proper distribution of the solution, or the flow must be split to multiple beds. The design problems are surmountable, but the more complicated layouts can become very expensive. Flow rates and acid concentration of the feed will vary inversely, and insufficient data have been reported to establish their true relationship for varying depths of bed. It can be said, however, that passage of 2 gpm per sq ft of bed area of 5 per cent nitric or hydrochloric acid through a 6-ft bed of limestone will yield satisfactory neutralization. Stronger acids may be neutralized in this manner, but difficulties will be encountered with foaming. To determine the necessary design factors, it is much safer to carry out small-scale tests.

The neutralization of *sulfuric* acid involves more severe limitations because of the low solubility of the resultant calcium sulfate. The upper limits of acid concentrations are 0.3 per cent H_2SO_4 for high-calcium limestone and 0.5 per cent for dolomitic. Above those strengths, the stone will become inactivated because of the deposition of calcium sulfate. If it is impractical to dilute the feed acid to the required concentration, another neutralizing agent should be selected. Figure 26-2 indicates flow rates and bed depths for ¾-in. lump dolomitic limestone and 0.3 per cent sulfuric acid. These curves are also indicative of results that can be obtained with high-calcium limestone at the lower acid strength, but here again it will be safer to run tests with the actual acid and limestone that are concerned.

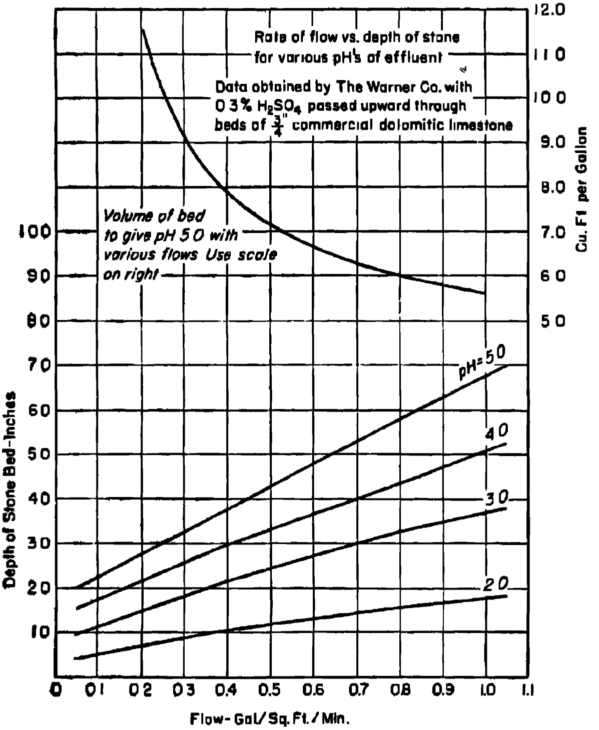

FIG. 26-2. Limestone bed performance. (*Warner Company.*)

Very occasionally, *hydrofluoric* acid may have to be disposed of by neutralization. The solubility of calcium fluoride is so low that the use of limestone for this purpose is impractical. Besides, the high toxicity of fluorides in general will usually prevent their disposition as water-borne wastes. The calcium fluoride should be precipitated with lime and removed as a sludge for disposal on a dumping ground.

Organic acids will lend themselves to neutralization in beds of limestone when the resultant calcium salts are relatively soluble. It should be remembered, however, that the BOD of the organic radical is unaffected by neutralization and that it may still become an objectionable waste.

Ground Limestone. Limitations imposed on the use of ground limestone for acid neutralization are rather severe, although it will find application in certain cases. These are confined to situations where the required limestone feed rate can be predetermined. Control of the feed by means of a pH instrument is not feasible for two reasons: the reaction time is too long, and the lack of active alkali plus the formation of CO_2 limit the pH rise. With dilute acids and with a limestone ground to pass a 20-mesh screen, reaction times of 30 to 60 min are required when the theoretical quantity of high-calcium limestone is used. Dolomitic limestones react so slowly that their use in the ground form is rarely worth consideration. In cases where the quantity of the discharged acid can be controlled, the use of ground limestone is practical. The material in the screen size mentioned will handle and feed with ease provided the moisture content is below 1 per cent. It can be stored in silos or bins without hanging up or becoming caked, and it is easily removed with a screw conveyor or other type of feeder. In finer sizes, however, it loses this characteristic and approaches hydrated lime in difficulty of handling. Facilities for retaining the acid-limestone mixture in a well-agitated state for at least 30 min should be provided. This may take the form of an agitated tank or tanks, a series of baffled flumes, or, where conditions permit, a sewer line at the proper slope and long enough to satisfy the retention requirements. Retention time may be shortened materially by utilizing an excess of stone, but this usually calls for the addition of facilities for the removal of the unreacted material. One should not forget that sulfuric acid requires dilution to 0.3 per cent before successful neutralization can be accomplished with limestone. Figure 26-3 shows reaction-rate curves for three different ground limestones. The stones were used in laboratory tests to neutralize mixed acid containing approximately 3,000 ppm H_2SO_4 and 2,600 ppm HNO_3. It is obvious that sample *C* was superior in reaction rate to *B* and *D*. The curves emphasize the importance of small-scale tests with the actual materials to be used in the plant.

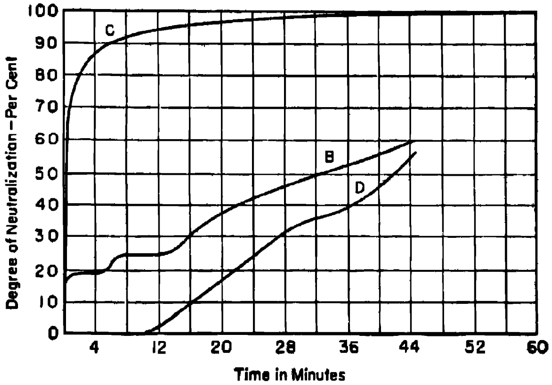

FIG. 26-3. Limestone reaction rates.

Hydrated Limes and Quicklimes. These materials are considered under a common heading since the chemical form in which they are used for neutralization is the same. Quicklimes must be properly hydrated before use. Quicklime should be considered whenever the neutralizing job is of sufficient magnitude to warrant the expense of slaking equipment and the operation of it. Quicklimes have higher neutralizing values than their corresponding hydrates to the extent of about 28 per cent for high-calcium and 14 per cent for dolomitic (see Table 26-3).

Table 26-3

Material	*Equivalent H_2SO_4, lb per lb*
High-calcium limestone	0.93
Dolomitic limestone	1.06
High-calcium hydrate	1.26
Dolomitic hydrate	1.63
High-calcium quicklime	1.68
Dolomitic quicklime	1.85
Calcium oxide (100%)	1.75

Delivered costs of quicklimes and hydrates in bulk are about the same per net ton. An advantage for quicklime is that bulk shipments are common, while hydrate is usually shipped in bags at a premium of about $1 per ton. Quicklime is more dangerous to handle than hydrate since it must be kept dry, and serious burns to the skin can occur if carelessness in handling is permitted. With the use of proper equipment for unloading and storing, however, these dangers are overcome, and the quicklime will generally store and handle better than hydrate.

The big advantage in the use of hydrated lime (purchased as such or prepared from quicklime) is that the alkali content is very active and is adapted to the use of pH-control equipment. In the usual neutralization problem, high-calcium lime will react completely in 1 to 2 min, while the dolomitic variety requires 6 to 10 min depending on the conditions under which it was calcined. Some authorities claim that shaft-kiln lime will react more slowly than that from a rotary kiln. The difference is not great, however. The curves in Fig. 26-4 indicate the reaction rates which may be expected with hydrated limes. Curve *A* represents a typical high-calcium lime; *B*, a dolomitic hydrate; and *C*, a dolomitic lime hydrated under 30 psi pressure. For the latter material it is claimed that the magnesium is completely hydrated. The tests supplying data for these curves were made with dilute acid containing 2,000 ppm H_2SO_4.

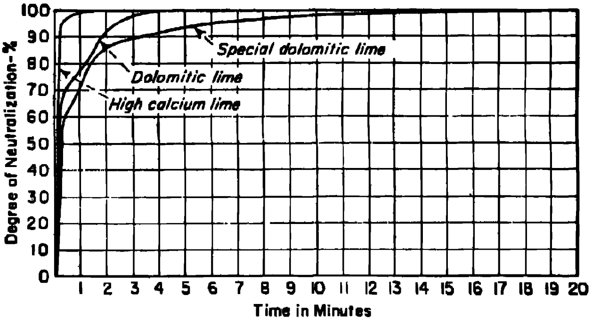

Fig. 26-4. Reaction rates of lime.

In general, hydrated lime will successfully neutralize acids at any reasonable concentration, and the pH of the resulting mixture can be controlled with automatic equipment. In some cases, particularly with sulfuric acid, heat is developed from the reaction and from the diluting effect of the added water. If the acid is too strong, the heat may be sufficient to dry out the reaction mixture until it becomes solid. This problem does not usually arise since acids of that strength can be recovered economically. Obviously, if the reaction products are not soluble, arrangements to separate and dispose of the solid matter will have to be made.

Figure 26-5 indicates equipment that will work satisfactorily where quicklime is to be used for acid neutralization. The quicklime in ground or pebble form is stored in a hopper or silo of either concrete or steel and is fed to the slaker at a constant rate by means of one of several types of feeder. Several types of lime slakers are made, and any of them sold by a reputable manufacturer will be all right provided it is properly sized. The principal requirement is that it must provide for the retention of the initial mixture at the proper lime: water ratio for a sufficient time and at a suitable temperature (95 to 98°C) to obtain complete hydration. A diluting compartment is usually provided in which the thick slurry is thinned to a solids content of 10 to 15 per cent for actual use. In cases where the demand is variable, it is necessary to provide intermediate slurry storage which can be utilized as a feed tank. The lime feeder and water valves to the slaker can be turned on and off electrically as needed by high- and low-level contacts in the slurry tank. The slurry is charged to the neutralizing tank by means of a centrifugal pump and a recirculating line. Recirculation of the slurry is required because of the necessity for maintaining a high velocity in the supply line to the control valve (at least 5 fps) to prevent stoppage. The control valve is regulated from a pH recorder controller, the electrodes of which may be suspended directly in the liquid in the mixing tank or installed in a chamber through which a portion of the overflow water passes. Several companies manufacture reliable pH-control instruments, but it is necessary to make sure that the control valve is of a type approved by the instrument supplier. The acid solution and lime slurry are introduced in separate streams to the downcomer of the neutralization tank. This tank is agitated by means of a sweep-type agitator and is equipped with baffles for turbulence. The volume of retained liquor is determined by the factors mentioned previously.

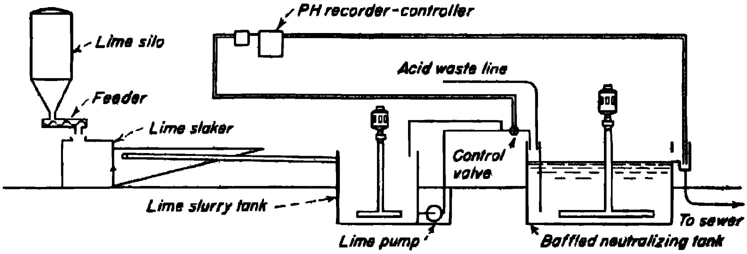

FIG. 26-5. Acid-neutralizing installation.

Color Improvement. If the effluent from a plant is sufficiently colored to affect the receiving waters adversely, the source should be determined and appropriate steps taken to correct the condition. Textile mills and dye plants are most frequently troubled by this problem, although tanneries, sulfite pulp mills, and some other industries must cope with it. The nature of the color will, of course, determine the steps which must be taken to remove or to destroy it. If the material is present as solid particles, means for settling it out or of precipitating and removing it must be devised. If it is in solution, as is usually the case, it must be absorbed, or its chemical composition must be altered in such a way that it is no longer noticeable. Sometimes simple adjustment of the pH prior to discharge will improve the situation since some dyes act as indicators.

The most positive approach will involve some form of chemical treatment either to reduce or oxidize the compound responsible for the color. The more common reducing agents such as SO_2 and the sulfites should be tried as well as acidification of the solution followed by the addition of metallic iron. The usual oxidizing agents consist of the dichromates, permanganates, hypochlorites, or liquid chlorine. The latter will be the most economical from the chemical-cost standpoint, although special precautions are necessary in handling it. If the color is due to a soluble sodium salt, it can frequently be converted to an insoluble compound by acidification followed by treatment with lime or some other alkali metal.

Floating Oils. Difficulty from floating oils can usually be overcome by the application of preventive measures at the equipment or process that contributes the contaminant. If the contamination cannot be prevented by reasonable means, the oil-bearing effluent should be isolated at the point where the smallest possible volume will require treatment. Once the oil has been mixed with the effluent water, the best method for separating it is to pass the effluent through a suitable oil separator where the liquid is retained at a low velocity such that the oil will rise to the surface where it can be removed by skimming. Formulas for optimum design of oil separators for use in the petroleum industry have been worked out (see Fig. 26-6).[4] These data as well as other valuable information are available in pamphlet form.[5] A line sketch of an oil separator taken from the pamphlet appears in Fig. 26-7.

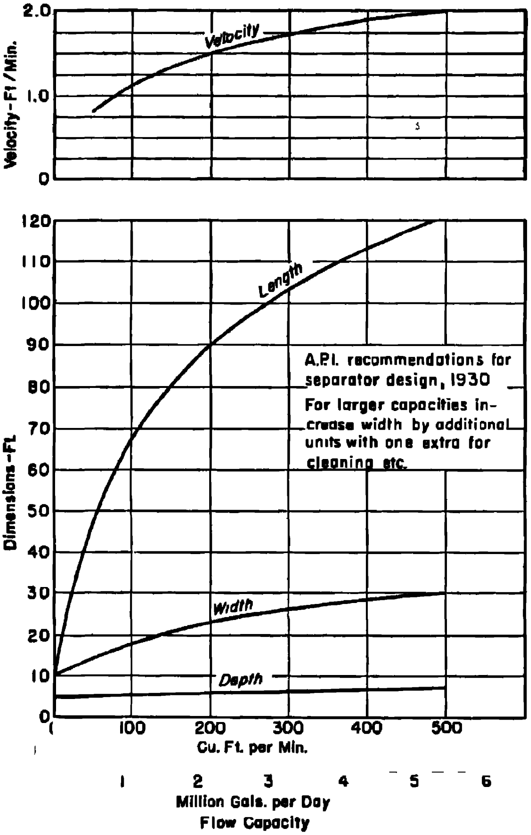

FIG. 26-6. API separator-design factors.

Problems will arise where oil is present in an emulsified or thoroughly dispersed condition. Oil separators are of no value in these cases unless steps are taken to "break" the emulsion beforehand. This cannot be accomplished by the same method in every case, but since the particles of oil are surrounded by a colloidal film containing an electrical charge, the job is one of neutralizing the charge or altering the nature of the film so that the oil particles will coalesce. Sometimes this may be accomplished by changing the pH of the water, by heating, or by the use of a coagulant. A laboratory investigation of the problem is essential.

Reduction of Biochemical Oxygen Demand. Biochemical oxygen demand (BOD) of an industrial waste can be due to organic matter in suspension and in solution. Material in suspension should be removed by methods discussed on pp. 1594 to 1595.

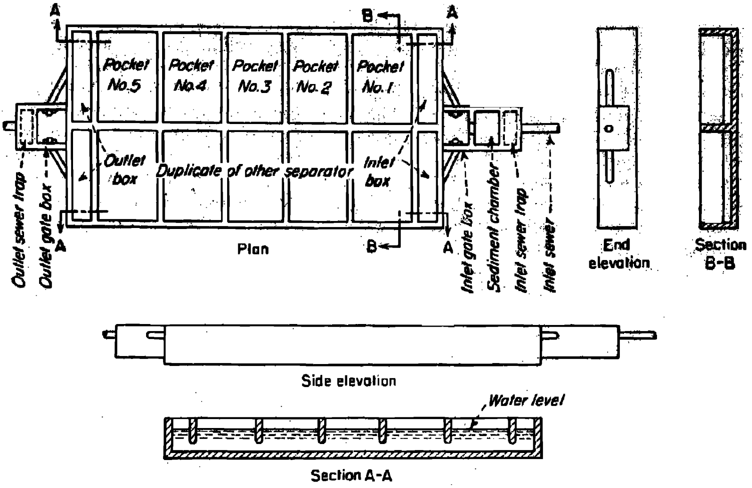

Fig. 26-7. Design of typical refinery separator.

Organic matter in solution requires treatment somewhat outside the usual experience of the chemical engineer. It is in the field of BOD reduction or destruction that a great deal of investigation is needed.

Some idea of the extent of the pollution created by various organic compounds can be gained from Table 26-4. The BOD values were determined in industrial laboratories during routine testing of plant effluents. Industries such as dyeing, beet-sugar extracting, canning, wood pulping, tanning, packing, and distilling are examples of those producing wastes characterized by high BOD values.

Perhaps the outstanding development in the industrial-waste-treatment field has been the adaptation of the trickling filter to the reduction of BOD content of industrial effluents. The trickling filter and its variant, the biofilter, were first applied to the purification of clarified sewage in which the BOD is present in solution and in the form of finely divided or colloidally dispersed particles of organic matter. Application of this principle has been widely used in the treatment of milk wastes and certain wastes from distilleries. At least one commercial biofilter is working satisfactorily in this country in the treatment of organic wastes from the manufacture of viscose rayon. The Dow Chemical Company at Midland, Michigan, has had a battery of four trickling filters operating for several years in the destruction of phenols in certain of their waste waters. Results of laboratory and semiworks scale tests indicate that

Table 26-4. BOD of Pure Compounds

Compound	Lb BOD per lb	Compound	Lb BOD per lb
Methyl alcohol	0.82	Sodium formate	0.04
Ethyl alcohol	1.00	Acetone	0.69
n-Propyl alcohol	0.47	Ethyl acetate	0.86
Isobutyl alcohol	0.07	Dibutyl phthalate	0.43
Formic acid	0.19	Butyl alcohol	2.04
Acetic acid	0.88	Phthalic anhydride	0.72
Methacrylic acid	0.89	Cornstarch	1.25
Methyl methacrylate	0.14	Casein	1.17
Calcium acetate	0.43	Gelatin	0.88
Formamide	0.00		

many problems involving organic contamination can be successfully solved by the use of trickling filters or biofilters. The latter will doubtless find wider application because of their ability to withstand "shock" loads as well as their greater flexibility in operation at variable flow rates.

A trickling filter is a bed of stone placed over a large surface composed of drainage tile or other liquid-collecting system. The stone is usually sized from 2 to 4 in. and should be of a variety that fractures unevenly. The underdrain system can be one composed of special tile members, or it can be a simple grid system for suspending the stone over a drainage space which is properly sloped to collect the liquid at a suitable point for discharge to a settling basin. The stone is built up to a depth of 5 to 8 ft and is surrounded by a wall or enclosure of wood, cement block, or concrete. The retaining wall can be eliminated by allowing the stone around the outer edge to seek its normal angle of repose. This naturally calls for greater bottom dimensions for the filter, and a larger quantity of stone must be used. The shape of the bed in plan may be square, rectangular, or round. Fixed or stationary distributors are used for square and rectangular filters, while rotary distributors are used with round filters. Round filters are usually installed in modern designs.

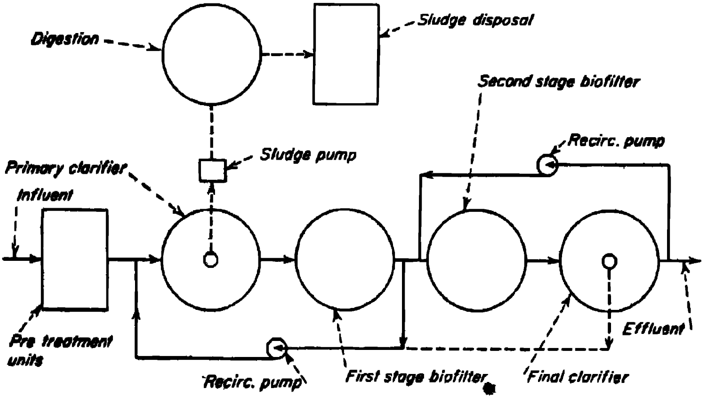

FIG. 26-8. Two-stage biofiltration.

The biofilter is a shallow trickling filter, and the details of construction are practically the same for each except that the depth of stone in the biofilter is held to 3 or 4 ft. In operation, the liquid is recirculated over the filter by means of pumps in varying amounts depending on the degree of reduction of BOD desired. The quantity of recirculated liquid will vary from one to ten times the volume of the feed. The sewage or waste being applied to the filter must be settled prior to application to avoid clogging of the interstices between the stones. Final settling of the filter discharge is also required, and if more than one filter is used in series, intermediate settling may be required. A line sketch of one type of biofilter installation is shown by Fig. 26-8. Several variations of equipment placement and recirculation practices such as parallel, series, and series-parallel are possible.

Trickling filters are operated on a once-through basis of the liquid being treated. The application rate for sanitary sewage and for organic wastes of comparable strength is usually set at 2 million gallons per acre per day, or about 45 gal per sq ft per day. At this rate, the BOD load with normal sewage will be 0.5 to 0.8 lb per cu yd of stone per day. Biofilters are designed for application rates as high as 30 million gallons per acre per day with BOD loadings up to 4.0 lb per cu yd per day. Owing to the low rate of application to trickling filters, the action is intermittent with alternating periods of wetting and drying of the film on the stones. With the biofilter, however, the object of the high application rate is to approach as nearly as possible continuous wetting. With rotary distributors, a short period of nonapplication between wettings is inevitable, although certain distributor designs permit liquid application approaching the continuous.

The functions of the filter are (1) to provide a medium upon which the purifying bacteria can grow and multiply; (2) to supply oxygen for use of the bacteria in oxidation of organic matter; and (3) to provide ample food for the bacteria continuously. The bacteria are present in raw and settled sewage, and it is necessary only to provide them with suitable conditions to carry out their functions. The bacteria establish themselves on the surface of the stone in the form of slimy growths. Air is drawn up through the bed by natural draft, and the applied liquor passes downward, spreading in a thin film over the surfaces of the stone, thereby coming into intimate contact with the slimes. The phenomena by which the organic matter in sewage is destroyed in passing through a biological filter are not definitely understood, but the nature of the process has been variously described as colloidal precipitation, absorption, and changes in surface tension. It is certain, however, that one function of the bacteria is to utilize the oxygen from the air in converting the complicated organic compounds into the simple end products, carbon dioxide and water. A portion of the ammonia and other nitrogen compounds end up as nitrates, nitrites, and free nitrogen. There is evidence to indicate that certain bacteria utilize organic nitrogen as a source of energy since a disappearance occurs as the sewage is treated.

Many organic compounds found in industrial wastes can be treated successfully for BOD reduction on trickling filters or biofilters. Most of the compounds listed in Table 26-4 have been treated on experimental filter units with BOD reductions of 70 to 75 per cent in most cases. In fact, any organic compound on which a successful BOD analysis can be carried out should lend itself to biological treatment. If the waste organics originate in chemical processes, they will probably be "sterile," and inoculation with sewage prior to biotreatment will be required.

In a short time after a biological filter has been put into operation, a "sloughing" or "unloading" of the slime growth occurs. In a trickling filter this action usually takes place intermittently, but in a biofilter it is practically continuous. This "sloughed" material contains a portion of the BOD which has been removed from the liquid as well as excess bacterial colonies. It is necessary, therefore, for this material to be removed from the filter effluent before it can be discharged to the receiving water. It is customary in sewage-treatment practice to settle out the slimes and pump them back to the primary sedimentation tank from which they are removed to the digestion tank. The sludge undergoes digestion under anaerobic conditions and is finally discarded as a dry solid after treatment on the sludge beds. In biological treatment of industrial wastes, final sedimentation and sludge digestion must be provided even though preliminary settling ahead of the biological filter is not required.

Disposal of Cyanides. The disposal of cyanide wastes is a problem largely confined to the metals industry, although it does exist in the coke industry and in some strictly chemical operations. The metal-plating industry uses the cyanides of copper, nickel, and zinc along with the alkali cyanides of sodium and potassium. The wastes are produced in two forms: spent plating or heat-treating bath and dilute wastes from the wash tanks. If disposal of the cyanides as a water-borne waste is practiced, the concentration of the cyanide as NaCN should not exceed 0.25 ppm in the receiving waters. Special precautions must be taken to assure that rapid and complete mixing with the receiving waters occurs so as to prevent the formation of zones of pollution which will be lethal to fish and marine organisms.

Strong wastes may be placed in disposal ponds for ground absorption and evaporation, but there is a real danger from contamination of underground water supplies. Solid wastes can be sent to the dumping ground, but, there again, a hazard exists unless the dump is located in a place inaccessible to the public and where rain will not carry the soluble matter to a stream. Ponding of dilute wastes for several months will result in the disappearance of the cyanide radical as oxidation and decomposition take place. The bottoms of such ponds should be sealed so that leakage does not occur.

Treatment of a cyanide solution with ferrous sulfate results in the formation of a corresponding ferrocyanide which lacks the lethal properties of the straight cyanide.

The ferrocyanides are generally highly colored salts and may cause complaints when discharged without further treatment to a receiving water. Unless a large excess of ferrous sulfate (500 to 600 per cent) is used, the resulting solution will not be completely free of cyanides. When molecular equivalents are used, the cyanide will not be reduced much below 50 ppm. The ferrous sulfate method should be used only under conditions where receiving-water quality is not critical.

Most cyanides will convert to hydrogen cyanide (HCN), one of the most lethal of gases, on being acidified with a mineral acid. This chemical reaction is used to advantage in one method of disposing of waste cyanide plating solutions. The wastes are placed in an acidproof tank, covered, and made airtight except for an exhaust stack. Air is forced through the liquid through perforated pipes laid on the bottom of the tank. Strong sulfuric acid (60 or 66°Bé) is charged to the tank through an overhead pipe after the solution to be treated has been run in. The rate of acid addition controls the evolution of the HCN gas, and the air agitation assists in driving it out of solution. The stack discharges the air–HCN mixture at a height sufficient to prevent toxic mixtures from reaching the ground. An induction fan discharges to the base of the stack to provide positive draft and additional dilution. This method is unsafe in congested areas, and the rate of acid addition must be carefully calculated and controlled.

The lime–sulfur method is frequently utilized for the treatment of strong cyanide solutions and waste cyanide solids from heat-treating installations. The solution is heated by steam and held at a temperature of about 60°C while commercial lime–sulfur solution is slowly added with continuous agitation. A yellow precipitate is first formed, and the end of the reaction is indicated by a permanent orange-red color. The precipitate can be settled and removed for disposal to a dumping ground. The supernatant liquor should be allowed to leach into the ground at a suitable location or discharge to the receiving water at a low rate, provided the volume is not too great.

Cyanides can be converted to cyanates by treatment with sodium or calcium hypochlorite. The alkali cyanates are relatively harmless to fish and will not revert to cyanides in the receiving waters. The metal cyanates are no more toxic than other compounds of the same metal. In using the hypochlorite-treatment method, care must be taken to maintain sufficient alkalinity to prevent the formation of the toxic gas, cyanogen chloride. It is customary to use a molecular equivalent of free alkali for each mol of cyanide calculated as NaCN. Oxidation of the cyanide to cyanate is the most positive method for the destruction of its toxic properties, and there are usually no remaining products requiring special treatment except copper or nickel under some conditions.

The Wallace & Tiernen Company has perfected a method for oxidizing cyanides which is a further development of the method just described.[6] According to unofficial reports, the process involves chlorination of the cyanide-bearing solution by means of a standard chlorinator with precautions taken to keep the pH of the resulting mixture above 8.5 at all times. The use of hypochlorite or chlorine will find particular application in the treatment of wash waters and other dilute wastes.

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SECTION 27

HEAT TRANSFER AND HEAT-TRANSFER PROPERTIES OF MATERIALS

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HEAT TRANSFER*

BY WARREN M. ROHSENOW, D. ENG.

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There are in general three kinds of heat transfer: conduction, convection, and radiation.

Conduction is the transmission of heat by molecular vibrations from one part of a body to another part of the same body or from one body to another body in physical contact with it, without appreciable displacement of the particles of the body.

Convection is the transmission of heat from one point to another within a gas or a liquid by the mixing or circulation of the fluid. If the fluid is circulated by the action of gravity due to differences of density of the hotter and colder portions of the fluid, the process is called "natural convection"; if the fluid is circulated by a pump, blower, or other mechanical means, the process is called "forced convection."

Radiation is the transmission of heat in the form of electromagnetic waves. All bodies give off heat in the form of radiant energy. When this radiant energy falls on another body, part of it is absorbed by the body, and part of it is reflected. The difference between the amount of radiant energy which a surface gives off and the amount it absorbs from other surrounding surfaces which it "sees" is the net heat transferred by radiation.

In most cases the actual heat transfer is accomplished by more than one of these three types of heat transfer; however, in some cases the major part of the heat transfer is accomplished by one of the types, and the heat transferred by the other types may be neglected. For example, in the furnace cavity most of the heat transferred to the first row of boiler tubes is accomplished by radiation; in later tubes both convection and radiation heat transfer must be considered; in the economizer and condenser convection heat transfer is most important.

Most heat-transfer problems applied to a power plant are of a steady-state type. The operating conditions, such as temperature and pressure, do not vary appreciably with time, and the rate of heat transfer from or to a particular surface is constant with time. We shall consider only this type of problem—one of steady-state operation.

NOMENCLATURE

A = area of heat-transfer surface, sq ft

b = height of plate, ft

C = conductance, Btu/hr/sq ft/deg F

cp = specific heat at constant pressure, Btu/lb/deg F

D = pipe diameter, ft

D' = pipe diameter, in.

D_c = coil diameter, ft

F_{12} = angle factor

$\mathfrak{F}_{12}$ = factor relating angle factor and emissivity

G = mass velocity, lb per hr per sq ft of cross section occupied by fluid

g = 4.18×10^8 ft per hr per hr

h, h_v = local individual film coefficient of heat transfer, Btu/hr/sq ft/deg F difference

h_s = coefficient of heat transfer for scale deposit

$(h_v + h_r)$ = combined coefficient of conduction, convection, and radiation between surface and surroundings.

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* Superior numbers indicate specific references listed at the end of this subsection.

k = thermal conductivity, Btu/hr/sq ft/unit temperature gradient, deg F per ft
L = length of pipe, ft
N = number of tubes through which fluid flows
q = rate of heat transfer, Btu per hr
r = radius, ft
t = temperature, deg F
t_s = condensing temperature of steam in a condenser
t_{w1}, t_{w2} = inlet and outlet temperature of cooling water
T = $t + 460$, temperature deg F absolute
U = over-all coefficient of heat transfer, Btu/hr/sq ft/deg F
V = average velocity, volumetric rate of flow divided by cross section filled by fluid, ft/sec
w = mass rate of flow, lb per hr
x = thickness of conductor or wall, ft
X = temperature rise of cold fluid ÷ inlet temperature of hot fluid minus inlet temperature of cold fluid
Y = factor that depends upon the surface arrangement of a heat exchanger
Z = temperature fall of hot fluid ÷ temperature rise of cold fluid
ϵ = emissivity of a surface
Γ = mass rate of flow: $w/\pi D$ for vertical; w/L for horizontal tubes
ρ = density, lb per cu ft
σ = 0.1723×10^{-8}, Stefan-Boltzmann constant
θ_1 = temperature difference between hot fluid entering and colder fluid leaving
θ_2 = temperature difference between hot fluid leaving and cold fluid entering
μ = viscosity at bulk temperature, lb per hr per ft = 2.42 times centipoise
μ_s = viscosity of fluid at temperature of the wall

CONDUCTION

Flat Solid Wall. The rate of heat transfer through a flat solid wall of a single material is calculated from the equation

$$= kA \; t_1 - t_2 \tag{1}$$

where k is called the thermal conductivity of the material and is a property of the kind of material and the average temperature of the material. It is the amount of heat transferred per hour through 1 sq ft area of a wall 1 ft thick with a 1°F temperature difference across the wall. Values of thermal conductivity k for various materials are given in the next subsection, Heat-transfer Properties of Materials.

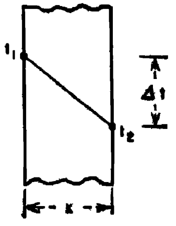


FIG. 27-1. [See Eq. (1).]

Cylindrical Solid Wall. The rate of heat transfer through a cylindrical solid wall of length L is calculated from the equation

$$q = 2\pi Lk \frac{t_1 - t_2}{\ln\ (r_2/r_1)} \tag{2}$$

where r_2 and r_1 are outside and inside radii in feet or inches, ln is logarithm with base e.

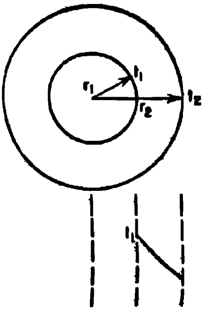


FIG. 27-2. [See Eq. (2).]

Composite Wall. A composite wall is one composed of a number of layers of different materials such as a building wall built up of layers of plaster, stone, and brick or an iron pipe covered with a layer of high-temperature insulation followed by a layer of 85 per cent magnesia. For these walls thermal conductance C of the wall is defined by and the rate of heat transfer is calculated from the following equation:

$$q = CA(t_1 - t_n) \tag{3}$$

where t_1 and t_n are the inner and outer surface temperatures of the composite wall.

For a flat wall composed of $(n - 1)$ layers of material:

$$\frac{1}{(x_A/k_A) + (x_B/k_B) + (x_C/k_C) + } \qquad (4)$$

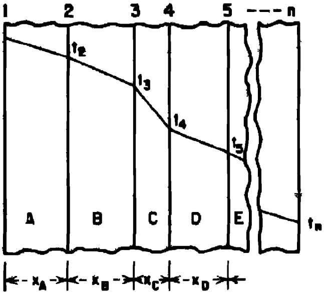


Fig. 27-3. [See Eq. (4).]

where x_A and k_A refer to the material layer A, etc. As many terms appear in the denominator of Eq. (4) as there are layers of different materials.

Cylindrical Composite Wall. For a cylindrical wall composed of $(n - 1)$ layers of material, the product of surface area times thermal conductance is

$$CA = \frac{1}{\dfrac{\ln (r_2/r_1)}{2\pi L k_A} + \dfrac{\ln (r_3/r_2)}{2\pi L k_B} + \cdots} \qquad (5)$$

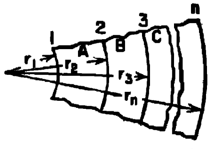


Fig. 27-4. [See Eq. (5).]

The rate of heat transfer q through a flat composite wall is then calculated by substituting the value of C from Eq. (4) in Eq. (3); for a cylindrical composite wall substitute the value of the product CA from Eq. (5) into Eq. (3).

CONVECTION

When a fluid flows parallel to a wall, the thin layer of fluid which is touching the surface clings to the wall and does not move. The fluid just next to this stagnant layer moves in sheets parallel to the wall; this is called the laminar layer. At the outer edges of the laminar layer the fluid in the transition zone tends to break away from its sheetlike formation to enter the turbulent region where it has extremely random motion. Heat transferred between the surface and the main stream must

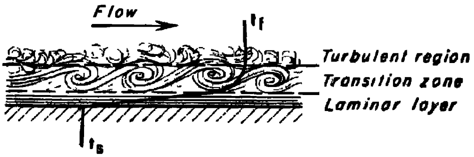


Fig. 27-5. Turbulent flow of a fluid flowing parallel to a surface.

be transferred through these various layers. Heat is transferred by conduction through the laminar layer and by convection through the transition zone. The laminar layer and transition zone combined are known as the film.

Since it is impracticable to separate the effects of these two types of heat transfer in this case, the rate of heat transfer between the surface at temperature t_s and the main body of the fluid at temperature t_f is calculated from the equation

$$q = hA(t_s - t_f) \tag{6}$$

which defines the film coefficient of heat transfer h having the units Btu per hour per deg F temperature difference per square foot of heat-transfer surface. Exact values of this film coefficient are not known in many cases, but empirical equations based on experimental results for the more common cases are given under the heading Film Coefficients, this Section, page 1613.

CONDUCTION AND CONVECTION

An important case of heat transfer is that from a hot fluid on one side of a solid wall to a cooler fluid on the other side. The wall may be a cylindrical or a flat wall of a single material or composite wall of layers of different materials. The rate of heat transfer is calculated from the equation

$$q = UA(t_h - t_c) \tag{7}$$

where t_h and t_c are the temperatures of the hot and the cold fluids, respectively.

Flat Composite Wall. Equation (7) is used to calculate the rate of heat transfer from a hot fluid successively through the hot-side film, layers of solid material of the wall, and the cold-side film to the cold fluid. For this case U of Eq. (7) is

$$U = \frac{1}{(1/h_h) + (x_A/k_A) + (x_B/k_B) + \cdots + (1/h_c)} \tag{8}$$

where h_h and h_c are the hot- and cold-side film coefficients of heat transfer, and the other terms are as defined in Eq. (4).

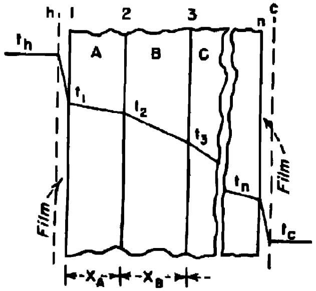


FIG. 27-6. [See Eq. (8).]

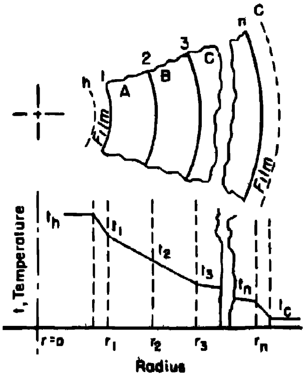


FIG. 27-7. [See Eq. (9).]

Cylindrical Composite Wall. Equation (7) is also used to calculate the rate of heat transfer from a hot to a cold fluid through a composite cylindrical pipe wall. For this case the product UA of Eq. (7) is

$$UA = \frac{1}{\dfrac{1}{2\pi r_1 L h_h} + \dfrac{\ln (r_2/r_1)}{2\pi L k_A} + \dfrac{\ln (r_3/r_2)}{2\pi L k_B} + \cdots + \dfrac{1}{2\pi r_n L h_c}} \tag{9}$$

Surface Temperature. Sometimes it is necessary to know the temperature of the inner and outer faces of the wall and also the temperature at the interface between layers of different materials within the wall. When these temperatures cannot be measured conveniently, they may be estimated by calculation. Consider the flat composite wall shown in Fig. 27-6. The rate of heat transfer q through the wall is calculated from Eq. (7) and (8). In the case of steady-state heat transfer the rates of heat transfer through the films and through each layer of material of the wall are all equal and given by this value of q. The rate of heat transfer through the films is given by Eq. (6) and through each layer of the composite wall by Eq. (1). The temperature of the hot and cold fluids may be measured; and values of film coefficients h_h and h_c may be found under the heading Film Coefficients, this section, page 1613. Thermal conductivity k_A and k_B may be found in Sec. 27, Heat-transfer Properties of Materials, page 1618. Then the surface temperatures t_1 and t_n may be calculated from Eq. (6).

$$t_h - t_1 = q/h_hA$$
$$t_n - t_c = q/h_cA$$

and the interface temperatures may be calculated by successively using Eq. (1). Since t_1 is now known, calculate t_2 from

$$t_1 - t_2 = (qx_A)/k_AA$$

Since t_2 is now known, calculate t_3 from

$$t_2 - t_3 = (qx_B)/k_BA$$

This process may be continued until all interface temperatures are known.

RADIATION

The net rate of heat transferred by radiation from one surface to another separated by a vacuum or a gas which is transparent to the radiation is calculated from the equation

$$q = A_1\mathfrak{F}_{12}\sigma(T_1^4 - T_2^4) \tag{10}$$

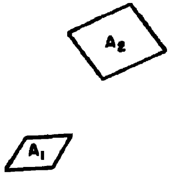


FIG 27-8. [See Eq. (10).]

where A_1 is the area of surface 1. The factor $\mathfrak{F}_{12}$ is calculated from the equation

$$\mathfrak{F}_{12} = \frac{1}{\dfrac{1}{F_{12}} + \left(\dfrac{1}{\epsilon_1} - 1\right) + \dfrac{A_1}{A_2}\left(\dfrac{1}{\epsilon_2} - 1\right)} \tag{11}$$

where ϵ_1 and ϵ_2 are emissivities of the two surfaces, and A_1 and A_2 are areas of the two surfaces. The emissivity ϵ of a surface depends upon the character of the surface and temperature of the surface. Values of emissivity for various surfaces are tabulated in the next subsection, Heat-transfer Properties of Materials. The angle factor F_{12} depends upon the size, shape, and arrangement of the two surfaces exchanging heat by radiation. The value of F_{12} is not known for all surface arrangements and shapes. It has been evaluated for only a few of the more common ones.

Angle Factor, F_{12}. Surfaces for which $F_{12} = 1.00$:

1. A small flat surface radiating to a wall which forms a large hemisphere
2. Concentric spheres or long concentric cylinders such as long pipes

3. Large, closely spaced rectangular or circular plates, where ratio of side or diameter to distance between planes is large (greater than 15)

Surfaces for which $F_{12} \neq 1.00$:

1. Figure 27-9 is for perpendicular rectangular planes.

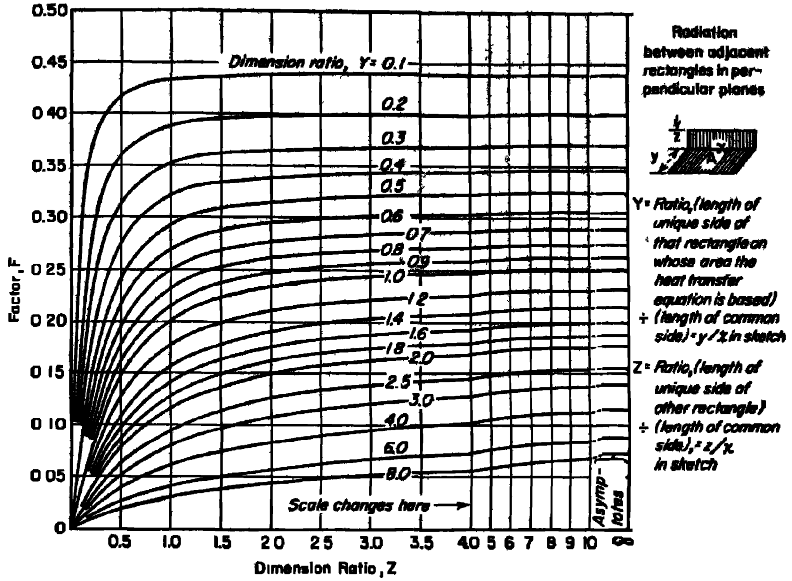


FIG. 27-9.

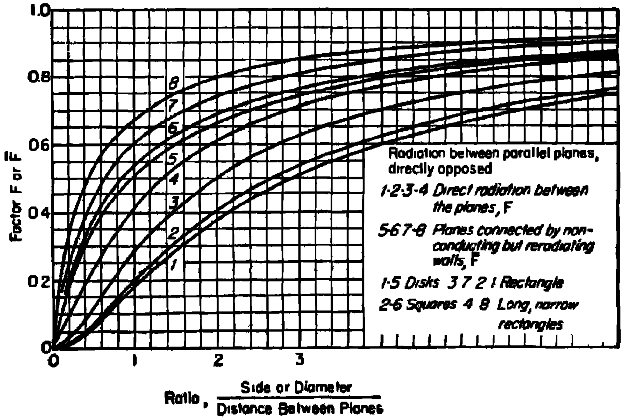


FIG. 27-10.

2. Figure 27-10 shows the angle factor for parallel planes (disks, squares, and rectangles) directly opposite each other. Lines 5, 6, 7, 8 are for planes separated by refractory (nonconducting but reradiating) walls.

3. Figure 27-11 is for radiation between a plane and one or two rows of tubes parallel to the plane.

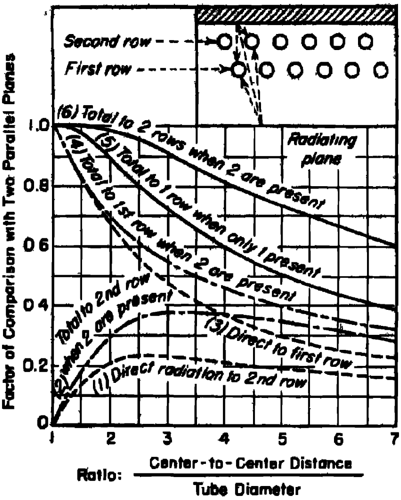


FIG. 27-11.

Problems of furnace design involve radiation of gases, flames, and clouds. A detailed study of these problems is found in McAdams.[1]

CONVECTION AND RADIATION

Heat transfer by convection and radiation occur at the same time from a given surface. Usually one has the greater effect, and the other may be neglected. In some cases, as in heat transfer from bare and insulated pipes, the convection and radiation are of the same order of magnitude. Then it is convenient to use a combined convection and radiation coefficient $(h_v + h_r)$. The rate of heat transfer becomes

$$q = (h_v + h_r)A(t_s - t_f) \tag{12}$$

where h_v is defined by Eq. (6) and is determined from experimental data given under the heading Film Coefficient, and h_r is determined by q_r calculated from Eqs. (10) and (11) from

$$h_r \quad A(t_s - t_f) \tag{13}$$

where t_s and t_f are temperatures of the surface and surrounding fluid, respectively.

For certain special cases, values of the combined coefficient $(h_v + h_r)$ are given also under Film Coefficients.

HEAT EXCHANGERS

The equation for calculating the rate of heat transfer in a heat exchanger is

$$YUA \frac{\theta_1 - \theta_2}{\ln (\theta_1/\theta_2)} \tag{14}$$

This equation assumes constant U, constant specific heats of the two fluids, constant rates of flow, no changes in phase, and negligible heat loss to the atmosphere. The product UA is evaluated as in Eqs. (8) or (9)

Factor Y: For a pure counterflow heat exchanger, $Y = 1$. For various multipass and crossflow heat exchangers, Y is determined from Figs. 27-12 and 27-13.

Steam Condenser. In a steam condenser one fluid is a condensing vapor, and the other is a liquid that does not change phase. The rate of heat transfer is calculated from

$$q = UA \frac{t_{w_2} - t_{w_1}}{\ln [(t_s - t_{w_1})/(t_s - t_{w_2})]} \tag{15a}$$

For condenser tubes with cooling water inside the tubes, the value for UA in the above equation is

$$UA = \frac{N}{\dfrac{1}{2\pi r_1 L h_c} + \dfrac{1}{2\pi r_1 L h_s} + \dfrac{\ln (r_2/r_1)}{2\pi L k_a} + \dfrac{1}{2\pi r_2 L h_h}} \tag{15b}$$

where h_s is the coefficient of heat transfer for the scale deposit in the tubes.

A valuable graphical method of interpreting the results of tests on surface condensers, proposed by E. E. Wilson,[2] consists in plotting $(1/AU)$ vs. $(1/V)^{0.8}$ and drawing the best straight line. When the condenser tubes are cleaned of scale deposit, the value of $1/AU$ at $1/V^{0.8} = 0$ is the sum of the heat-transfer resistance of the condensing vapor and the tube wall. For later tests, when scale deposit has formed, the straight line will be higher on the curve sheet. The difference between $1/AU$ for the "clean" and the "dirty" condition at $1/V^{0.8} = 0$ measures the amount of scale formed and may be used as an indicator for the necessity of cleaning the condenser tubes.

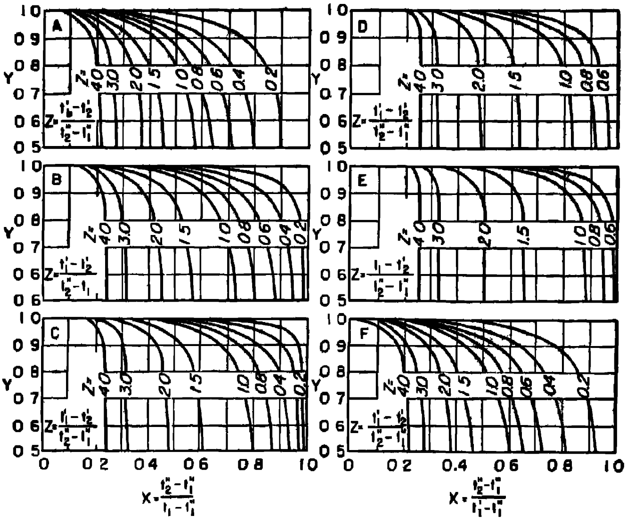
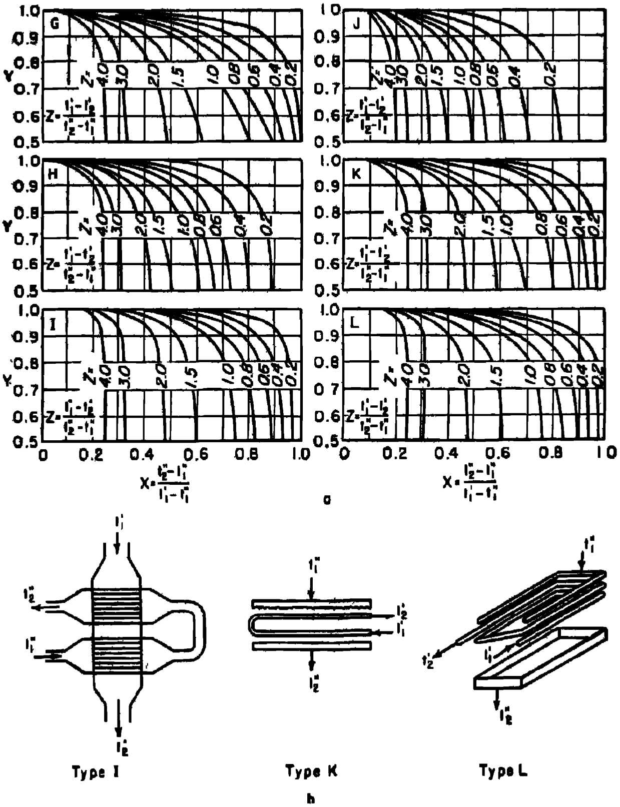


FIG 27-12. Factor Y in reversed-current heat exchangers (shell side well mixed at a given cross section). (A) 1 shell pass and 2, 4, 6, etc , tube passes, (B) 2 shell passes and 4, 8, 12, etc , tube passes, (C) 3 shell passes, and 6, 12, 18, etc , tube passes, (D) 4 shell passes, and 8, 16, 24, etc , tube passes; (E) 6 shell passes, and 12, 24, 36, etc , tube passes; (F) 1 shell pass, and 3, 6, 9, etc., tube passes. [See McAdams (1), page 146.]

FILM COEFFICIENTS

The value of the film coefficients depends upon two kinds of properties—physical properties of the materials, such as thermal conductivity, viscosity, specific heat, and density, and geometrical properties, such as shape and arrangement of heating surface and velocity of flow across the surface. The problem of prediction of film coefficients divides itself into three velocity ranges. In the high-velocity range *turbulent flow* exists as described in Fig. 27-5. As the velocity is decreased, the laminar layer increases until the entire tube stream flows in a laminar flow. A further reduction in velocity to near zero causes *natural convection* currents to govern the prediction of film coefficients.



a

b

Fig. 27-13. Factor Y in cross-flow heat exchangers: (G) cross flow, both fluids unmixed, 1 tube pass; (H) cross flow, shell fluid mixed, 1 tube pass; (I) cross flow, shell fluid mixed, 2 tube passes—shell flow flows across second and first passes in series; (J) cross flow, shell fluid mixed, 2 tube passes—shell fluid flows over first and second passes in series; (K) cross flow, (drip type), 2 horizontal passes with U-bend connection (trombone type); (L) cross flow, (drip type), helical coils with 2 turns. [See McAdams (1), page 147.]

Turbulent Flow. Most applications met in practice are forced flow of gases or liquids in turbulent flow. An empirical equation correlating the heating of gases and liquids inside horizontal pipes when Re > 4,000 is

$$\mathrm{Nu} = 0.023(\mathrm{Re})^{0.8}(\mathrm{Pr})^{0.4} \qquad (16)$$

where the dimensionless numbers are

Nu = Nusselt number = hD/k
Re = Reynolds number = DG/μ
Pr = Prandtl number = $c_p\mu/k$

The following simplified equations for specific cases are deduced from Eq. (16):

Turbulent flow of gases inside clean straight tubes:

$$h = 0.024[c_pG^{0.8}/(D')^{0.2}] \tag{16a}$$

Turbulent flow of water inside clean straight tubes:

$$h = 160(1 + 0.012t)[V^{0.8}/(D')^{0.2}] \tag{16b}$$

Turbulent flow in coiled pipes:
Multiply h for straight pipes by the factor

$$1 + 3.5(D/D_c)$$

Turbulent gas flow normal to a bank of staggered tubes:

$$DG/\mu > 1{,}000$$
$$h = 0.36[c_pG^{0.6}/(D')^{0.4}] \tag{16c}$$

Turbulent water flow normal to a bank of staggered tubes:

$$h = 370(1 + 0.0067t_f)[V^{0.6}/(D')^{0.4}] \tag{16d}$$

For tubes in line use 75 per cent of the value of h determined from Eqs. (16*c*) and (16*d*); for baffled exchanger use 60 per cent of the values above for unbaffled exchangers.

Streamline Flow. Streamline flow is encountered in the flow of viscous liquids, such as oils, at values of $DG/\mu < 2{,}100$.

The coefficient of heat transfer is calculated from the equation

$$hD/k = 2.0(wc_p/kL)^{1/3}(\mu/\mu_s)^{0.14} \tag{17}$$

Natural Convection. Coefficients of heat transfer by natural convection from surfaces to air are given by the following equations where Δt is the temperature difference between the surface and the surrounding air:

Horizontal plates facing upward:

$$h = 0.38(\Delta t)^{0.25} \tag{18a}$$

Horizontal plates facing downward:

$$h = 0.2(\Delta t)^{0.25} \tag{18b}$$

Vertical plates more than 1 ft high:

$$h = 0.27(\Delta t)^{0.25} \tag{18c}$$

Vertical plates less than 1 ft high:

$$h = 0.28(\Delta t/b)^{0.25} \tag{18d}$$

Vertical pipe more than 1 ft high and horizontal pipe:

$$h = 0.27(\Delta t/D)^{0.25} \tag{18e}$$

Condensing Vapors. When a pure vapor condenses on a tube, the condensate wets the tube resulting in "film-type" condensation. If the condensate flows in streamline motion ($4\Gamma/\mu < 2{,}100$), the following equations may be used:

Horizontal tubes:

$$h = 0.76(k^3\rho^2g/\mu_f\Gamma)^{1/3} \tag{19a}$$

Vertical tubes:

$$h = 0.93(k^3\rho^2g/\mu_f\Gamma)^{1/3} \tag{19b}$$

These values are for pure vapors. If noncondensable gases such as air are present, the values are lower than those calculated from Eqs. (19*a*) and (19*b*). Hence it is essential to vent vapor-heated apparatus.

When steam is condensing the addition of small traces of oleic acid or benzyl mercaptan prevents the formation of a film of the condensate on the surface and induces a "dropwise" condensation resulting in h values that are three to four times larger than those of "film-type" condensation.

Boiling Liquids. Liquids are usually evaporated by natural convection heat transfer from the surface of submerged tubes which are heated internally either by condensing vapors or by electricity. The coefficient of heat transfer depends upon the kind of liquid boiling, the temperature difference between the surface and boiling liquid (Δt), temperature of the boiling liquid, character and arrangement of heating surface, and agitation of boiling liquid. When a liquid is being evaporated at a given pressure, the coefficient of heat transfer rises to a maximum value as the temperature difference between heating surface and boiling liquid is increased; beyond this critical temperature difference its value decreases. This effect for boiling water is shown in Fig. 27-14. It is noted that the critical temperature difference is about 40°F for water boiling at atmospheric pressure.

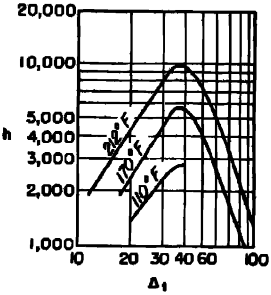


FIG. 27-14. Effect of Δt and t on h for boiling water.

Since $q/A = h(\Delta t)$, the maximum q/A occurs at a Δt greater than 40°F; for boiling water it occurs at about 45°F. If an electric heating element is used, it is essential that Δt be less than the critical value to prevent burning out the heating element or throwing the circuit breaker.

Combined Convection and Radiation. Values of $(h_v + h_r)$ given by Bailey and Lyell[4] for horizontal bare steel pipe in a room at 80°F are given in Table 27-1.

Table 27-1. Values of $(h_v + h_r)$
(For horizontal bare standard steel pipe in a room at 80°F)

Nominal pipe diam, in	Temperature difference, deg F, surface to room							
	50	100	200	300	400	500	600	700
1	2.26	2.50	3.00	3.60	4.34	5.16	6.05	6.98
3	2.05	2.25	2.73	3.31	4.03	4.85	5.71	6.66
5	1.95	2.15	2.61	3.20	3.90			
10	1.87	2.07	2.54	3.12	3.84			

For rotating cylinders in air Hinlein gives values as in Table 27-2.

Table 27-2. Values of $(h_v + h_r)$
(For rotating cylinders in air)

Surface	Velocity, V, fps					
	0	20	40	60	80	100
Smooth polished copper	0.47	1.97	2.79	3.14	3.30	3.38
Rough, dull, black-varnished	0.47	2.49	3.32	3.96	4.50	5.10

For flat surfaces contacting air, Lichty (*Univ. Illinois Bull.*, 1915) gives values as in Table 27-3.

For external surfaces multiply ($h_v + h_r$) by factors shown in Table 27-4.

Table 27-3

Surface in still air	$h_v + h_r$
Cement plaster finish	0.9
Concrete or corkboard	1.3
Brickwork, sheet asbestos, or wood	1.4
Plate glass or magnesia board	1.5

Table 27-4

Air velocity, mph	5	10	15	20
Factor for brickwork	2.4	3.0	3.8	4.2
Factor for wood	1.2	2.0	2.9	3.0

Scale Deposits. Heat-transfer coefficients h_s for scale deposits from water for use in Eq. (15*b*) are given in Table 27-5.

Table 27-5. Heat-transfer Coefficients h_s for Scale Deposits from Water*

Temperature of heating medium	Up to 240°F		240–400°F	
Temperature of water	125°F		Above 125°F	
Water velocity, fps	3 and less	Over 3	3 and less	Over 3
Distilled water	2,000	2,000	2,000	2,000
Sea water	2,000	2,000	1,000	1,000
Treated boiler feed water	1,000	2,000	500	1,000
Treated make-up for cooling tower	1,000	1,000	500	500
City, well, Great Lakes	1,000	1,000	500	500
Brackish, clean river water	500	1,000	330	500
River water, muddy, silty †	330	500	250	330
Hard (over 15 grains/gal)	330	330	200	200
Chicago Sanitary Canal	130	170	100	130

* From "Standard of Tubular Exchanger Manufacturers Association," 1941.

† Delaware, East River (New York), Mississippi, Schuylkill, and New York Bay

NOTE: Additional values and information concerning film coefficients may be found in McAdams (Reference 1) and Stoever (Reference 3).

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HEAT-TRANSFER PROPERTIES OF MATERIALS

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THERMAL CONDUCTIVITY

Before an engineer attempts to use tabulated data on heat-transfer properties, he must realize that "test" data and actual values in the field may vary to a great extent owing to changes within the material being used. The conditions of test and actual use must be compared and proper judgment used in selecting actual values of the various coefficients.

Conductivity k may vary with each of the following items (see Table 27-6).

Mean Temperature.

$$\text{Mean temp} = (t \text{ hot side} + t \text{ cold side})/2$$

The conductivity k of noncrystalline materials will increase with an increase of mean temperature (Fig. 27-15), while crystalline materials will show decreasing k within a certain temperature range, then reversing the curve and increasing conductivity k as temperature increases.

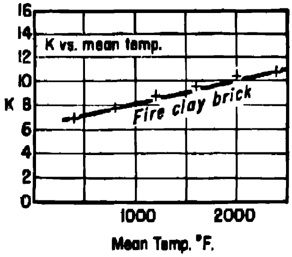

FIG. 27-15. Conductivity vs. mean temperature.

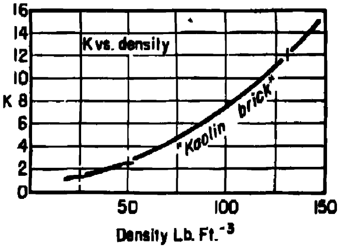

FIG. 27-16. Conductivity vs. density.

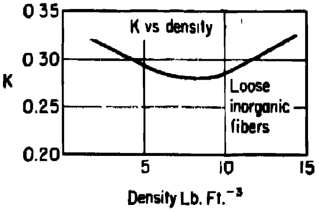

FIG. 27-17. Conductivity vs. density of loose inorganic fibers.

Density (lb-ft^{-3}). If the heat transfer through a given material is by conduction alone, the thermal conductivity will increase with increasing density (Fig. 27-16).

In loose fibrous materials where internal radiation and/or convection may occur, the apparent k may increase with lower densities as may be seen in Fig. 27-17.

Moisture. Not only will k increase with moisture content of the material, but also the heat-transfer coefficients will vary an undeterminable amount owing to moisture vapor migration through the specimen (see Fig. 27-18).

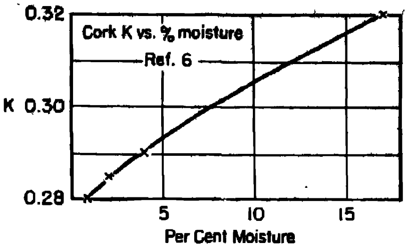

FIG. 27-18. Conductivity of cork vs. per cent moisture.

Pore Size. The conductivity of a material will decrease with a reduction of the size of closed pores; the limits of this reduction are not exactly known however. Experimental plastics with large closed pores and very low density have been found to have much higher k than the same material in a much smaller pore higher density form.

Table 27-6. Thermal Conductivity k of Miscellaneous Materials

Material, class and description	Density, lb-ft^{-3}	Mean temp, deg F	Conductivity, k*	Max safe temp, deg F	Authority†
Building boards, slabs, blocks:					
Portland cement and asbestos, pressed	123	86	0.23	600	1
Asbestos millboard	60.5	86	0.07	600	1
Gypsum, paper bound	63	70	0.11	225	3
Fiberboards	13–18	70	0.028–0.03	225	3, 4, 6
Corkboards	5–14	90	0.020–0.028	225	1, 6
Cork, asphalt bound	14–16	90	0.027	200	1
Wood shreds and cement	25–30	75	0.04–0.06	225	3, 4
Glass wool, asphalt-enclosed	11	80	0.021	200	7
Rock wool, asphalt binder	14–16	77	0.023–0.030	200	1, 7
Expanded glass block	11	80	0.035	300	7
Expanded rubber	5	80	0.018	200	7
Blankets and bats:					
Wood fiber, paper-bound	3.6	70	0.020	200	3
Eel grass, paper-bound	4	90	0.021	200	1
Hair felt	8–13	90	0.022 0.024	200	Various
Cotton fibers	0.5–1.0	72	0.020–0.030	200	Various
Kapok	1.0	90	0.020	200	Various
Glass-wool blankets	1.0–3.0	70	0.01	....	Various
Rock- or slag-wool blankets	15	60	0 027	400	6
Redwood bark	6.5	60	0.023	250	6, 7
Blowing materials:					
Mineral wool	14	90	0.027	500	12
	14	500	0.050	500	12
Miscellaneous loose materials:					
Lampblack	3–15	100	0.013–0.030	800	
Santocel‡	6	100	0.014	600	
Masonry:					
Brick, common	....	....	0.42	....	4
Concrete: sand and gravel	142	75	1.5	....	4
Cinder	97	75	0.41	....	4
Cellular	40–90	75	0.08–0.40	....	3, 4
Vermiculite	20–50	90	0.06–0.10	....	3
Plaster: cement	....	.....	0.66	....	2
Gypsum and vermiculite	40	75	0.08	....	3
Glass	....	.....	0.30–0.60	....	12

Table 27-6. Thermal Conductivity k of Miscellaneous Materials *(Cont.)*

Material, class and description	Density, lb-ft^{-3}	Mean temp, deg F	Conductivity, k*	Max safe temp, deg F	Authority†
Wood (all dry):					
Pine (various)	28–35	75–80	0.05–0.08	....	4
Hemlock	30	75	0.06	....	4
Fir	34	75	0.05	....	4
Oak	48	75	0.10	....	4
Maple	46	75	0.09	....	4
Refractories:					
Fire-clay brick	120–130	400	0.58–0.68	....	5
		800	0.65–0.73	....	5
		1200	0.73–0.84	....	5
		1600	0.78–0.90	....	5
		2000	0.84–0.97	....	5
		2400	0.88–1.00	2500	5
Kaolin brick	130	400	1.10	2900	5
	130	2400	1.64	2900	5
Silicon carbide brick	150	400	1.75	3200	5
	150	2400	6.65	3200	5
Magnesite brick	160	400	2.17	....	5
	160	2400	1.05	....	5
High-temperature insulators, "bricks":					
Kaolin brick	26	400	0.07	2500	5
	26	1600	0.13	2500	5
	55	800	0.24	2900	5
	55	2400	0.37	2900	5
Raw diatomaceous-earth brick	33	200	0.06	1200	5
	33	1200	0.08	1200	5
"Blocks":					
Vermiculite base	18.5	200	0.05	1200	5
	18.5	1200	0.07	1200	5
Mineral-wool base	22	200	0.035	1000	5
	22	800	0.07	1000	5
85 % magnesia	....	100–400	0.039–0.046	600	12
Metals:					
Aluminum	....	32	117	....	12
		572	133	....	12
		932	155	....	12
Brass	....	32	56	....	12
		752	67	. .	12
Cast iron	.. .	32	32		12
		752	25	..	12
Copper, pure	....	32	224	....	12
		1112	204	.	12
Lead	....	32	20	.	12
		572	18	.	12
Nickel	... ...	32	36	.	12
		572	32	.	12
Steel	....	212	26	..	12
		1112	21	. .	12
Admirality metal	... .	86	65		12
Bronze, commercial	.. ....	......	109	. .	12
Chrome alloys	.. . ..	625	14–19		12
Gases and liquids:					
Air	.... ..	−160	0.009	...	12
		32	0.014	.	12
		212	0.018	. ..	12
Ammonia	.....	−76	0.009	. ..	12
		32	0.013	.. .	12
		212	0.019	..	12
Oils: fuel and lubricating	......	70	0.08–0.09	....	12
Heat-transfer	... ...	86	0.077	...	12
		212	0.075	....	12
Water	....	32	0.032	....	12
		212	0.42	....	12

---

* k = Btu-ft/(hr) (sq ft) (°F).

† Authorities are listed at the end of this subsection under References.

‡ Trade-name product of Monsanto Chemical Company.

Mixture of Materials. The *k* of a material is not always proportional to the ratio of its components. A single example is given here, but each case would have to be worked out independently (Fig. 27-19)

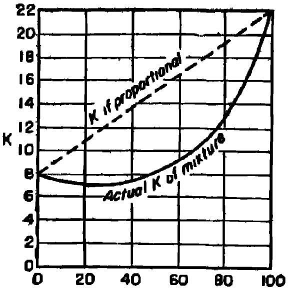

Fig. 27-19. Effect of mixtures of materials on conductivity.

EMISSIVITY

Total emissivity ε, the second property, which is shown in Table 27-7, is in general very difficult if not impossible to estimate by visual inspection of the surface This quality of a material may vary with the following items.

Nature of Surface. In general, bright clean metal surfaces will have low emissivity (0.03 to 0.10) at moderate temperatures, while nonmetallic materials, such as wood, paper, cloth, paints, etc, will have high emissivity (0.75 to 0.95) at the same temperature range

Surface Treatment. The mechanical treatment or condition of a surface is vital to its reflective properties. Polishing will decrease the ε of a surface, while any sort of roughening will increase emissivity.

Chemical treatment or painting a surface even with an extremely thin translucent coating may greatly increase the emissivity of a metal surface at moderate temperature ε values of treated surfaces are probably uncertain by 10 to 30 per cent.

Temperature. The emissivity ε of a surface is a function of the wavelength of the radiation being absorbed or reflected by the surface The wavelength of the radiation being absorbed or reflected is a function of temperature Therefore all data on emissivity should state the range of temperature at which the material was tested. For example, white paint is a poor reflector of low-temperature radiation but a good reflector of solar radiation

Table 27-7. Total Emissivity of Miscellaneous Materials, ε

Material	Temp of radiant source deg F								Reference*
	25	100	200	500	1000	2500	5000	Solar	
Metals									
Aluminum unoxidized		0.028		0.060					13
Bright		0.04		0.05	0.08	0.17	0.26		14
Oxidized		0.11		0.12	0.18				14
Rough-plate		0.055							12
Brass									
Polished		0.05							13
Unoxidized		0.035							13
Oxidized		0.61	0.61	0.59					13, 14
Emery rubbed		0.20							12
Carbon		0.81		0.79					13
Copper unoxidized		0.02							13
Liquid					0.15				13
Polished		0.04		0.05	0.08	0.17	0.26		14
Calorized		0.26	0.18		0.19				13
Oxidized black		0.67		0.83	0.77				14
Gold, polished		0.02		0.02	0.03	0.03			14
Iron pure polished		0.06		0.08	0.13	0.22	0.35	0.45	14
Smooth red oxidized		0.62							14
Oxidized		0.74							13
Rusted		0.65							13
Cast polished		0.21							13
Rough cast oxidized		0.98							14
Wrought		0.28							14
Wrought dull oxidized	0.94		0.94						13
Lead polished		0.06		0.08					14
Oxidized			0.63						13
		0.43							14
Magnesium, polished		0.07		0.13	0.18	0.23	0.26	0.30	14
Monel oxidized			0.43	0.43					13
Mercury, pure, clean	0.10	0.12							13
Nickel polished		0.04		0.06	0.10	0.16	0.28	0.40	14
Unoxidized	0.045	0.06		0.12	0.19				13
Oxidized		0.39		0.49	0.67				14

* Numbers in this column refer to specific references listed at the end of this subsection

Table 27-7. Total Emissivity of Miscellaneous Materials, ε (*Cont.*)

Material	Temp of radiant source, deg F								Reference*
	25	100	200	500	1000	2500	5000	Solar	
Platinum: unoxidized	0.037	0.047	...	0.096	0.152	0.191	...		13
Black	...	0.93	...	0.96	0.97	0.97	0.97	0.97	14
Silver: polished	...	0.01	...	0.02	0.03	0.03	0.04	0.11	14
Unoxidized	...	0.02	...	0.035	...	...	...		13
Steel: polished	...	0.07	...	0.10	0.14	0.23	0.37	0.45	14
Unoxidized	...	0.08	...	...	...	...	...		13
Clean liquid	...	...	...	...	...	0.29	...		13
Oxidized	0.80	0.79	0.79	0.79	...	...	...		13
Carbonized	...	0.52	...	...	...	...	...		14
Plate, rough	0.94	...	...	0.97	...	...	...		13
Tin, unoxidized	0.043	0.05	...	...	...	...	...		13
Zinc: polished	...	0.02	...	0.03	0.04	0.06	0.50	0.46	14
Unoxidized	...	...	0.05	...	...	...	...		13
Oxidized	...	...	...	0.11	...	...	...		14
Nonmetals:									
Paints									
Blacks and dark blues	...	0.94		0.98	...	...	...	0.99	14
Red (Fe_2O_3) Green (Cu_2O_3)	...	0.96	.	...	0.70	...	0.59	0.74	14
Whites	...	0.90–0.98	...	...	0.53–0.78	...	0.08–0.16	0.12–0.18	14
Aluminum paint (vary with binder used)	...	0.27–0.67	...	...	...	...	...		12
Ceramics:									
Building bricks and tiles	...	0.93	...	...	...	0.30–0.50	0.40–0.70	0.50–0.80	12
Glass, smooth	...	0.94	..	...	...	..	...		14
Firebrick: clay	...	...	..	...	0.75	...	...		12
Glazed	...	...	..	...	0.75	...	...		13
Silica brick	.	...	.	..	0.80	...	...		12
Porcelain, glazed	.	0.92	..	.	...	...	...		13
Quartz, fused	...	0.93	...	...	...	...	...		12
Miscellaneous:									
Asbestos boards and paper	...	0.93–0.96	...	...	...	...	...		12
Lampblack in water glass	..	0.92	Thin layers on steel						
	...	0.97	Thick layers on steel			...	...		12
Enamel white on steel	...	0.90	...	...	...	...	...		12
Oil on polished nickel	...	0.27	(0.001 in. oil)			...	...		12
	...	0.72	(0.005 in. oil)			...	...		12
	...	0.82	(∞ oil)			...	...		12
Ordinary paper, any color	.	0.92–0.94	.	...	.	.	.		12
Aluminum coated paper	...	0.07–0.30	...	...	...	...	...		
Plaster, rough	..	0.91	..	...	.		..		12
Roofing, paper	..	0.91	..	...	.	.	...		12
Rubber, hard glossy	...	0.94	...	.	...	...	...		12
Water	...	0.96	...		..	...	...		12

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SECTION 28

PHYSICS IN THE INDUSTRIES

By Lyman M. Dawes

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  - *C.* High-frequency induction furnace
  - *D.* Radio-frequency induction heating
  - *E.* Dielectric heating
  - *F.* Rectification of alternating current
  - *G.* Adjustable-speed electronic drive
  - *H.* Resistance welding control
  - *I.* Electronic timing
  - *J.* Temperature control
  - *K.* Regulating applications of electronics
  - *L.* Photoelectric cell
  - *M.* Photoelectric positioning control
  - *N.* Photoelectric control of registry
  - *O.* Lighting control by photoelectric means
  - *P.* Electrostatics
  - *Q.* Electrocoating

VII. Mechanical vibration
  - *A.* Vibration engineering
  - *B.* Commercial suspensions
  - *C.* Vibration instruments

VIII. Sonics and ultrasonics
  - *A.* Sound generators
  - *B.* Sonic vibrations
  - *C.* Ultrasonic vibrations

IX. Gravitation and centrifugal separation
  - *A.* Gravity for separation
  - *B.* Centrifuge and centrifugal separator

X. Radioactivity
  - *A.* Natural radioactivity
  - *B.* The structure of matter
  - *C.* Isotopes
  - *D.* Uranium pile
  - *E.* The curie unit
  - *F.* The Geiger-Mueller counter
  - *G.* Radioactive isotopes
  - *H.* Metallurgical processes
  - *I.* Safety of personnel
  - *J.* Nuclear fission

XI. Bibliography

PHYSICS IN THE INDUSTRIES

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X-RAYS

The discovery of X-rays dates from Nov. 8, 1895, when William Conrad Roentgen, while experimenting with cathode rays (streams of guided electrons), discovered that their impact on the walls of the tube to which they were confined produced a phenomenon, previously unknown, which because of its unknown character he called X-rays.

The general properties of X-rays were rather completely stated by Roentgen after he had experimented but a few weeks. In essence they were as follows: Many substances fluoresce when exposed to the rays; sensitized photographic materials respond to their action; they cannot be reflected or refracted by lenses; they travel in straight lines and are not deflected by magnetic fields; they discharge electrified bodies, whether the bodies are electrified positively or negatively; they are generated when cathode rays strike any solid body, and the heavier the element struck, the more efficient the generation of the X-rays.

Table 28-1. Wave Lengths and Frequency in the Electromagnetic Spectrum

Type of radiation	Wave length, cm	Frequency, cps	Some uses
Gamma rays caused by cosmic rays	Shorter than gamma rays from radioactivity	Higher than gamma rays from radioactivity	Scientific experimentation
Gamma rays	1.4×10^{-8}–2×10^{-11}	2.1×10^{18}–1.5×10^{21}	Radiography, therapeutics
X-rays	10^{-6}–6×10^{-10}	3×10^{16}–5×10^{19}	Radiography, fluoroscopy, therapeutics
Ultraviolet rays	4×10^{-5}–1.4×10^{-6}	7.5×10^{14}–2.1×10^{16}	Photography, phosphorescence, fluorescence, therapeutics, chemical changes, ionization of gases, photoelectric effects, sterilization
Visible rays	3.9×10^{-5}–7.7×10^{-5}	3.9×10–$^{14}7.7 \times 10^{14}$	Seeing, photography, photoelectric, and photochemical effects
Infrared rays	7.7×10^{-5}–4×10^{-2}	3.9×10^{14}–7.5×10^{11}	Heating, specialized photography, therapeutics
Short Hertzian waves	1×10^{-2}–1×10^{3}	3×10^{12}–3×10^{7}	
Radar	3×1–3×10	10^{10}–10^{9}	
Television	2.9×10^{2}–3.4×10^{2}	8.8×10^{7}–10^{8}	
Frequency modulation	6.8×10^{2}–1.4×10^{2}	4.4×10^{7}–2.16×10^{8}	
Amplitude modulation	1.88×10^{4}–5.46×10^{4}	5.5×10^{5}–1.6×10^{6}	
Short-wave telephone	....	4×10^{6}–10×10^{6}	
Radiotelegraph	....	10^{5}–10×10^{6}	
Radiotelephone	....	10^{5}–10×10^{6}	
Carrier telephone	....	4×10^{3}–10^{5}	
Carrier telegraph	....	425–10,750	
Program transmission	....	30– 8,000	
Ordinary telephone	....	200– 2,500	
Ordinary telegraph	....	10– 25	
Power work	....	25– 800	

The new discovery was immediately put to use in the medical and physical sciences, and other physicists discovered that X-rays can be polarized; are reflected from a smooth surface if they strike the surface within a fraction of a degree of parallelism; are slightly diffracted on passing through a narrow slit; are bent by refraction in passing through a substance; will either stimulate or destroy living tissue, depending upon the exposure; are scattered by the atoms of a substance, and when the substance is crystalline exhibit a pattern that identifies or establishes the substance's crystalline arrangement; can expel electrons from a substance, in which case the electrons expelled produce fluorescent rays; and the rays have been found to possess definite momentum and energy or quanta.

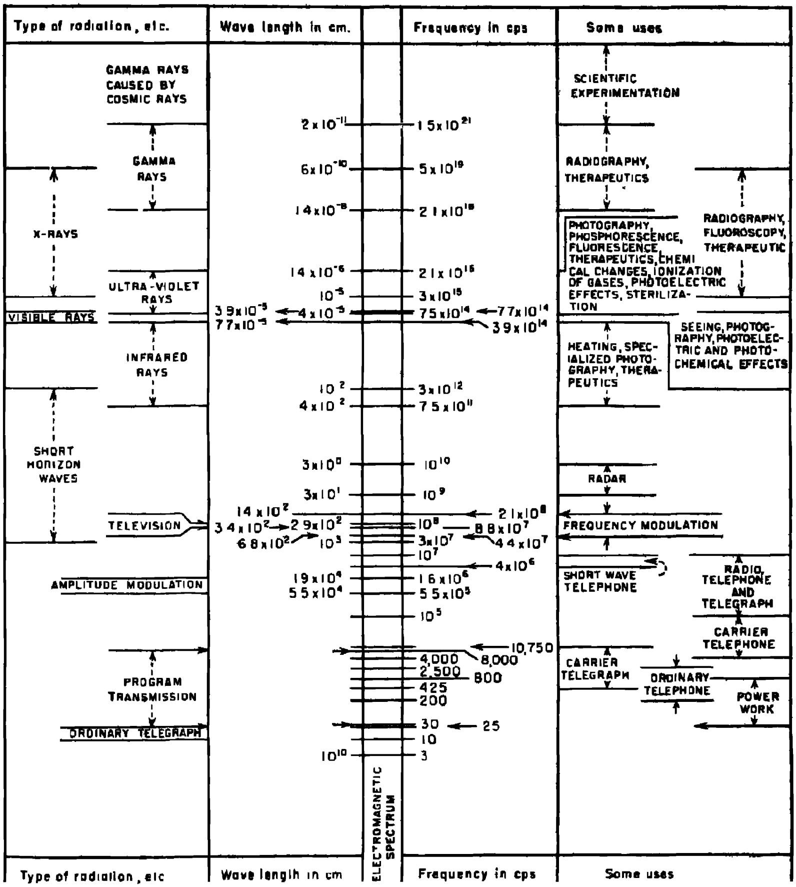

FIG. 28-1. Diagram of electromagnetic spectrum. A straightedge laid across the diagram will enable the user to identify the wave length and frequency bands by "Type of radiation" and "Some uses."

Electromagnetic waves as indicated in Table 28-1 and Fig. 28-1, include X-rays and gamma rays, closely associated radiations of very high frequency. Gamma rays are emitted during the radioactive disintegration of the heavier elements. It is now thought that they also result when cosmic particles impinge on matter. The impinging cosmic particles move under the influence of voltages more than a thousand times the high voltages used in the production of X-rays. The voltages for the production of X-rays range from a few thousands to a few millions of volts at present. As the voltage impelling a stream of electrons or cathode rays is increased, the velocity of the electrons is increased, and X-rays of shorter wave lengths are emitted on the impact of the electrons as their energy is increased.

The phenomena of electromagnetic waves are related by the equation

$$v = C/\mu\epsilon \tag{1}$$

where v = velocity of the wave train
C = ratio of electrostatic to electromagnetic charges
μ = permeability
ϵ = dielectric constant of the medium in which the waves are propagated.

These phenomena are also related by the equation

$$C = \lambda f \tag{2}$$

where C = velocity of light, usually taken as 3.0×10^{10} cm per sec [2.99796×10^{10} cm per sec (Michelson, 1926)]
λ = wave length of the radiation, cm
f = frequency, cps.

Within experimental error, C in Eq. (1) is equal to C in Eq. (2). In other words the velocity of light in empty space ($\mu = 1$, $\epsilon = 1$) is equal to the ratio C between the electrostatic and electromagnetic systems of units. Also

$$\lambda = hC/Ve = 12{,}350/V \quad \text{cm } 10^{-8} \text{ (minimum length)} \qquad f = Ve/h$$

where e = charge on the electron, esu
V = voltage between the electrodes of the tube
h = Planck's constant, equal to 6.55×10^{-27} erg-sec

The production of X-rays by the impact of the cathode rays on matter is associated with the phenomenon of photoelectric emission, and the kinetic energy of the photoelectrons is equated to light quanta and frequency by Einstein's photoelectric equation,

$$\tfrac{1}{2}mv^2 = hf - \omega_0$$

where m mass
v = velocity of the photoelectron
ω_0 = work required for the photoelectron to escape from the emitter.

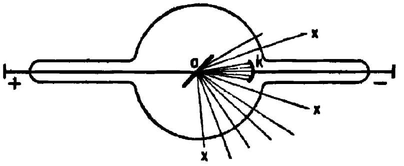

FIG. 28-1*a*. Early X-ray tube.

The generation of X-rays was at first produced in tubes such as that represented diagrammatically in Fig. 28-1*a*. It consisted of a spherical bulb tubulated at opposite ends of its diameter. Within the bulb was located a concave mirror of metal, usually aluminum, which served to bring the cathode rays to a focus on the anode placed near the center of the tube. The anode was made of a plate of platinum. Lead-in wires + and − extended to the anode and cathode respectively. When the terminals were connected, as marked in the figure, to a source of sufficiently high potential, X-rays were emitted from the surface of the platinum anode in all directions as indicated by x. In order that this tube may generate X-rays, it is necessary that it be filled with a gas at a low pressure (0.001 to 0.01 mm Hg). Under this condition and with a sufficient voltage on the electrodes, the gas becomes ionized, and a stream of electrons moves from the cathode to the anode as cathode rays. As the tube is used the pressure drops and ionization fails. To maintain this pressure relatively constant, many clever devices have been contrived. One scheme is shown in Fig. 28-2.

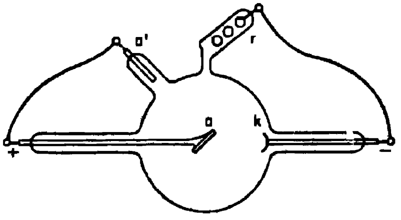

FIG. 28-2. Constant-pressure X-ray tube.

Material containing considerable adsorbed air, *e.g.*, mica, is introduced into an auxiliary tube r connected to the main chamber. When the pressure of the gas in the main bulb is lowered by the absorption of part of the gas, the resistance of the tube to the passage of the electric current is increased. When the resistance rises sufficiently, the current will be by-passed over the path from a′ to r. This results in heating of the gas-holding material, and gas is released to enter the main chamber. This corrects its partially exhausted condition, enabling it to continue to function. Tubes of this early type are still in use to a limited extent. It was the only type of X-ray tube in use until Dr. W. D. Coolidge developed the high-vacuum tube in 1913.

The high-vacuum or Coolidge tube is indicated diagrammatically in Fig. 28-3. In this tube the highest attainable vacuum is produced. The electrons are supplied by emission from the hot filament *f* which is at the same potential as the cathode *K*. The anode is often of copper with a tungsten plate inserted, thus providing a rugged target of high efficiency combining high melting point, high atomic number, and low vapor pressure. The copper has high thermal conductivity to facilitate heat transfer. The cathode stream is produced by evaporation of electrons from the hot filament of pure tungsten, and the high velocity of the particles in the stream is produced by the high voltage between the cathode and anode. The intensity and penetrating power of the X-rays may therefore be independently adjusted. The temperature of the filament, hence its emission, is adjusted by changing the voltage supplied by battery *B* or its equivalent, and the voltage applied to the electrodes is controlled by means of the generator.

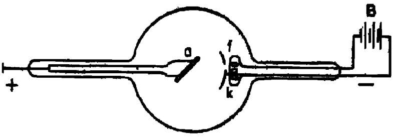

Fig. 28-3. High-vacuum or Coolidge X-ray tube.

Modern X-ray tubes are not so simple nor so uniform in construction as the foregoing paragraphs might imply. There are many problems in tube construction that years of development have not completely solved. Still the modern X-ray tubes are models of ingenuity and marvels of performance. They assume numerous forms and varied details in their construction, so that they may meet innumerable demands as to usage, precision, and economical life. Many of the required design features center about the focusing of the cathode rays to secure a suitable focal spot for the distribution of the X-rays without overheating the target or otherwise spoiling the tube. For many purposes an intense beam of X-rays, radiated from a focal spot of a few square millimeters, practically approximating a point source, is desirable. This condition is needed in radiography and fluoroscopy, where as close an approximation to a point source of X-rays as possible is desirable. For other purposes a focal spot several hundred square millimeters in area, the size of a dime or nickel, is warranted. This use would be in X-ray therapy, *e.g.*, in irradiation of tumors. Much of the difficulty stems from the fact that generally less than 1 per cent of the energy in the tube is converted into X-rays. For example, if a 100-kv tube (a medium-voltage tube) operates at a current of but 10 ma, the target receives 1,000 watts of power. A target of a few ounces receiving energy at such a rate will become white hot in a fraction of a minute unless some means of cooling it is provided.

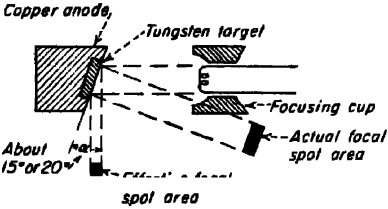

Fig. 28-4. Benson focus for X-ray tube.

The heating of tubes is controlled by many devices tending to limit, distribute, or dissipate the energy concentrated in the relatively small target. The destruction of the target may result from its melting or being so severely pitted that it can distribute only a small portion of the rays formed. The limitation of the heat is illustrated in the so-called universal tube, which has a target of solid tungsten and operates at an energy level below that at which the massive tungsten target will radiate. The target therefore operates at incandescence, and the tube is limited to an input which will prevent the tungsten from attaining a damaging temperature. But at this high temperature the tungsten emits electrons, and the tube is not self-rectifying and must be used on a d-c source. This tube may be used at a low intensity over long periods of time or at higher intensities for shorter periods of time. The ordinary universal X-ray tube may be used at 100 to 150 kv and 5 or 10 ma. Tubes similar to this, but with a specially constructed shield over the anode, having a rating about five or six times as great, are for use at 200 kv and a current of 25 or 30 ma.

One method of increasing the effective intensity of the X-rays from a tube while maintaining a lower intensity of electrons on the target is to focus the beam of electrons into a rectangular band and emit them in the form of a square having a side equal to the band width. This scheme is indicated in Fig. 28-4. The actual area of the electron beam striking the target is as indicated on the target as focal-spot area, but the resultant X-ray beam is about three times as intense as the normal beam would be when it leaves the target at the angle α instead of normally. This device is known as the Benson focus.

In addition to using the Benson focus, the tungsten electrode is frequently made in the form of a truncated cone with sides at an angle of 15 or 20 deg to its base. This target is mounted on one end of a shaft that has the rotor of an induction motor mounted on the other end. This assembly, together with a suitable bearing, is mounted within the tube. A stator is mounted outside the tube concentric with the rotor. A means is thus provided for rotating the target so that the focal spot is continuously moving over its surface. In this manner a much greater loading of the tube is possible with improved radiographic quality because of the higher intensity X-ray beam. The arrangement is indicated diagrammatically in Fig. 28-5. Many useful developments of rotating anode are included in the X-ray tubes of many manufacturers. To provide mechanical strength, tubes are made of Pyrex, or other heat-resisting glass, varying in thickness from the neighborhood of ⅛ to ¼ in. or more. Some tubes are made double walled.

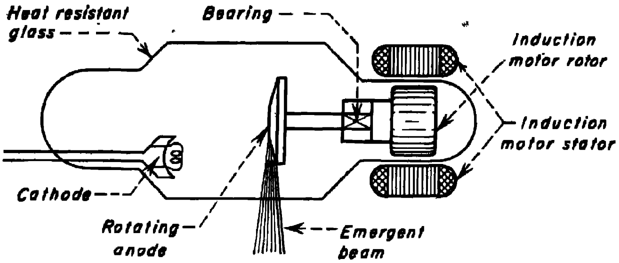

FIG. 28-5. Rotating anode for X-ray tube.

Cooling of X-ray tubes becomes more and more difficult as their capacity increases. A heavy copper support for the target can conduct considerable quantities of waste energy to the outside of the tube. A radiator is often placed on the outside end of the support to facilitate the dissipation of this heat. As the capacity of the tube is further increased, air may be blown across the radiator or liquid cooling embodied in the construction of the tube. The copper anode is often made hollow and oil or water circulated within it close to the target so that heat is carried rapidly to the outside of the tube. Water is about six times as effective as oil in carrying away the heat but presents electrical problems in its use not inherent in the oil.

Tubes are often completely immersed in a specially prepared mineral oil in an enclosing container. This permits connection by high-voltage cables to totally enclosed terminals and results in the "shockproof" tube for the greater protection of operating personnel.

Many other problems—thermal, electrical, and mechanical—are encountered in the making of modern X-ray tubes. With capacities hundreds of times as great as those of the early gas tubes, great quantities of heat must be dissipated. Voltages in the millions require the division of the tubes into numerous stages, each energized by the fraction of the total voltage that will meet the insulation problem and prevent flashover. Stray electrons must be recovered and drained away. Electrode spacing must be correct for the operating voltage—not close enough to allow cold-cathode emission or far enough to make focusing difficult. The envelope must be sufficiently rigid so that the electrodes may remain aligned even when one is rotating. The envelope must withstand heat without excessive change in shape or in resistivity when it is the insulator between electrodes. The necessary design conditions have been met by experienced and reputable manufacturers, and the tubes that they recommend for particular types of service may be used with confidence.

The voltage demand for X-ray tubes may vary from 5,000 to 50,000 volts for soft X-rays of low penetration, 20,000 to 200,000 volts for medium penetration, and 100,000 to several million volts for hardest rays with the highest penetration. In the early days of X-rays when the very high voltages and intensities were not required, static generators such as the Wimshurst machine were used. Many large induction coils were also used which furnished a pulsating direct current. Present-day practice usually embodies an a-c transformer with a high-voltage secondary, together with some form of rectifier to convert the alternating current to direct current. The rectifier may be a rotating spark gap driven by a synchronous motor which is fed from the primary circuit. The rotation of the commutator shown in Fig. 28-6 is in synchronism with the reversal of the alternating current. The commutator is so connected that, when the alternating current reverses, the connections through the brushes also reverse, thus causing the current to flow in one direction on the d-c side of the device. This contrivance is not suitable for reversal of large currents because at every reversal the brushes short-circuit the gap in the commutator, and severe sparking results.

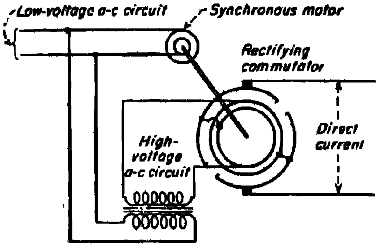

Fig. 28-6. Rotating commutator for voltage rectification.

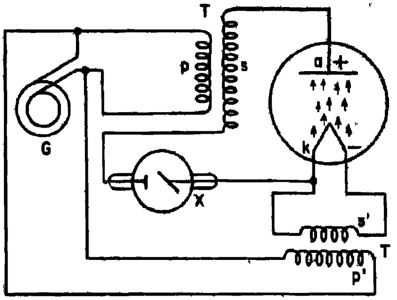

Fig. 28-7. Vacuum-tube rectifier, half-wave.

The vacuum-tube rectifier lends itself most readily to the rectification of high voltages with low currents. Figure 28-7 shows a circuit of a one-tube diode half-wave rectifier. Alternating current of customary commercial frequency is furnished by generator *G* to transformers *T* and *T'*. The low-voltage transformer *T'* has its secondary *S'* connected to the cathode or filament *K* for the sole purpose of heating it to cause the emission of electrons. At one-half cycle the conditions shown exist. Current may then flow by the electrons (negative charges of electricity) from the cathode, as shown by the small arrows, across the interelectrode space to the anode *a*, and through the transformer winding to the cathode of the X-ray tube. The electrons then cross the tube to the anode or target and back to the filament of the diode. During the next half cycle the anode becomes negative and repels the electrons, thus blocking the flow of current. This assumes that the tube will stand the inverse voltage without arcback, *i.e.*, that the inverse voltage rating of the tube should exceed the applied inverse voltage.

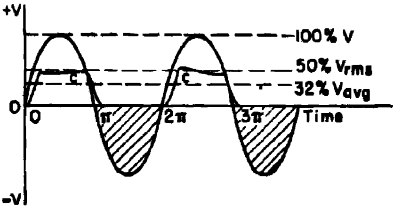

Fig. 28-8. Rectification of an a-c wave, half-wave rectification.

Figure 28-8 indicates an a-c wave assumed sinusoidal. A cycle extends from 0 to 2π and includes all positive and negative values of voltage. When a load is connected to the tube, a current will flow through the load somewhat as shown at *c*, its shape depending on the nature of the load. The negative half cycle, shown cross-hatched, is blocked out or not permitted to flow by the negative anode and positive cathode. The voltage shown as 100 per cent is the voltage across the electrodes of the tube, which is always somewhat less (from a few volts to some hundreds, depending on the characteristics and use of the tube) than the voltage impressed by the transformer. With half-wave rectification the rms voltage is 50 per cent of the net cathode-anode voltage, and the average voltage is 32 per cent of the cathode-anode voltage.

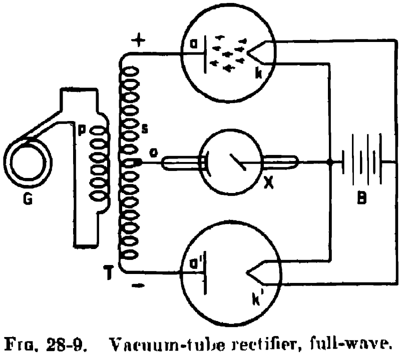

Fig. 28-9. Vacuum-tube rectifier, full-wave.

Full-wave vacuum-tube rectification is shown in Figs. 28-9 and 28-10. In Fig. 28-9 the upper tube is conducting from the cathode *K* to anode *a* through the X-ray tube X back to the cathode, since the anode is positive and the cathode negative. During this period the conditions in the lower tube prevent conduction, since the anode is negative and the cathode positive. During the next half cycle the conditions are reversed, and the upper tube is idle while the lower tube conducts. In this way the rectification of the voltage wave is accomplished, and current flows in only one direction through the X-ray tube *X*. The process is graphically indicated in Fig. 28-10. Two a-c cycles are shown with the negative halves dotted. Current flows always in a positive direction as indicated by the half waves *c*, *c*, *c*, *c*. The rms voltage is greater in full-wave rectification than in half-wave rectification by a factor $\sqrt{2}$; it is 70.7 per cent of the cathode-anode maximum voltage, and its average value is 63.9 per cent of this voltage. The full-wave rectifier is generally more efficient than the half-wave rectifier, depending on the characteristics of the load. It, however, requires exciting current for two cathodes rather than one. The ripple factor is lower in the full-wave rectifier. The ripple factor determines how closely the voltage approaches a direct or constant value. The lower the ripple factor, the more nearly the voltage approaches a constant voltage.

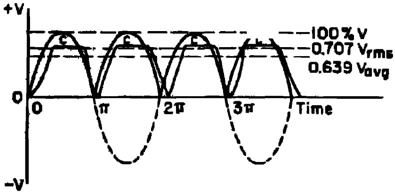

FIG. 28-10. Rectification of an a-c wave, full-wave rectification.

Full-wave bridge-type rectification is shown in Fig. 28-11 where four diodes are connected in a bridge circuit. Tubes *A* and *B* are conducting when the upper terminal of the transformer is positive, and tubes *C* and *D* are conducting during the time that the lower terminal is positive. This arrangement has the same tube efficiency, ripple factor, and percentage output voltage as the center-tap rectifier. However, full use is made of the transformer so that a smaller and more economical transformer may be used. This circuit also subjects the diodes to half the inverse voltage, which may result in economy in tube equipment even though twice as many tubes are required. It may be a disadvantage to have four exciting branches rather than two.

Other forms of rectifiers embody three-phase, six-phase, and other circuit combinations which decrease the ripple factor and provide a closer approximation to a direct current of constant value. It is not generally considered that the additional complication, expense, and lack of flexibility of multiphase circuits are warranted, since the resulting radiograph represents no improvement over those produced by single-phase X-ray generators. The multiphase systems possess the further disadvantage that polyphase supply is not so universally available as single phase is, particularly for portable use.

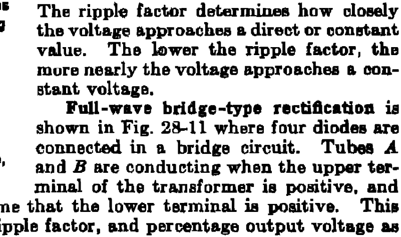

FIG. 28-11. Bridge-type rectification.

Self-rectification may take place in the X-ray tube itself, and this is a simple way of producing X-rays. Current results only by the movement of electrons from the cathode to the anode. In the high-vacuum X-ray tube, electrons are emitted from the filament; and these electrons, or negative charges of electricity, will move from the cathode to the anode whenever the anode is sufficiently positive with respect to the cathode. This occurs on alternate half cycles, as discussed under vacuum-tube rectification by means of diodes, and will occur during all that part of a cycle when the potential is high enough in a positive direction to impel them across the space from cathode to anode. This occurs during the greater part of each alternate half cycle. Thus, an intermittent direct current is provided, and X-rays of an intermittent character are produced by the recurrent positively directed cathode stream during nearly half the time when a high-potential alternating current is applied to the tube. Figure 28-12 is a diagram of the arrangement. The high-potential transformer supplies the voltage to the electrodes, and the low-potential transformer excites the filament. The primary windings of both transformers are connected to any convenient source of commercial electrical power.

The circuit of Fig. 28-12 does not provide for adjustment of voltage on the tube electrodes nor any control for the filament voltage. In most X-ray work such controls are necessary in order that the penetrating power and intensity of the X-ray beam may be adjusted to the requirements of the work. Autotransformers or reactors may be combined with the elements of the circuit shown in Fig. 28-12, as indicated in Fig. 28-13. Connections to the taps in the primary of either one or both transformers may be adjusted to regulate the voltage of the circuit in a manner similar to the method indicated at *AT*. An optional method of voltage adjustment is a variable reactor which may be inserted in the primary circuit of the transformer, as shown at *VR*. In either case the voltage of the primary, and hence the secondary, is adjusted.

A valuable improvement in X-ray practice is the use of the very-high-voltage d-c generators of the Van de Graaff type. These generators are now used with voltages of several million volts and with the X-ray tube built into the generator enclosure The physical size and weight of the equipment are not excessive, and very high efficiencies are attained (as high as 7 per cent at present). The target is made of gold, having a higher atomic number than tungsten, and thus higher X-ray emitting efficiency and ready cooling are attained because of its higher heat conductivity. The direct constant voltage is of great value, as it eliminates the self-rectification and intermittent operation present in the ultra-high-voltage a-c tubes. These elements combine to increase the intensity and penetrating power of the X-rays emitted. The intensity increases in a square-law relationship (doubling the voltage results in four times the intensity), an increase in the voltage causes a decrease in wave length, thus increasing the penetration, and for a fixed voltage the intensity is directly proportional to the tube current. The combination of these features results in very efficient equipment.

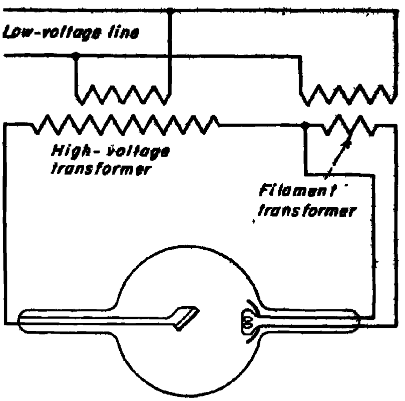

Fig. 28-12. Circuit of self-rectifying X-ray tube.

The elementary characteristics of X-radiation are required knowledge in the rational use of this powerful instrumentality. The essential characteristics of X-rays were summarized on pages 1624–1626. Before X-rays were discovered it was recognized that cathode rays, or high-speed electrons from a beam, when encountering solid matter penetrated it to the depth of a few millimeters. In fact, Roentgen was studying the behavior of these cathode particles when he discovered that in addition to penetrating the surface they caused the emission of a hitherto undiscovered form of electromagnetic radiation which he called X-rays. The place of this radiation in the electromagnetic spectrum has already been shown on page 1624. Quantitative expressions connecting wave length, velocity of light, Planck's constant, voltage, and charge on the electron are given on page 1624.

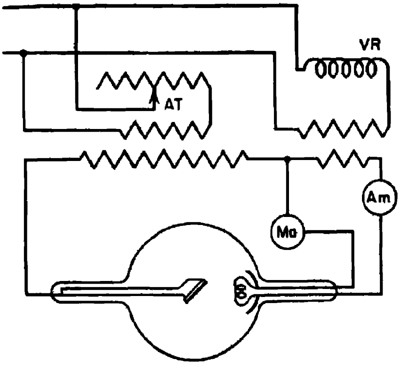

Fig 28-13. Circuit for regulating voltages and X-ray intensities by autotransformer and reactor.

The absorption of X-rays, a measure of what a technician might call "penetrating power," has been determined and tabulated for many substances. When a beam of X-rays encounters matter, it is absorbed in varying amounts dependent upon the coefficient of absorption of the particular material. The relationship is

$$\log I/I_0 = -\mu x$$

where I_0 = intensity of the incident beam
I = intensity of the emergent beam
x = thickness of the tablet of material
μ = coefficient of absorption

The coefficient of absorption may be determined by measuring I, I_0, and x. This proportionality factor has been published for numerous substances. If ρ represents the density of the material, we may introduce it into the foregoing equation to obtain

$$I = I_0 e^{-(\mu/\rho)\rho x}$$

Then ρx is the mass of a section of the tablet 1 cm square and x cm thick measured through the tablet; then the quantity μ/ρ is the mass absorption coefficient, a quantity often used. It is obvious that by making use of this relation, the intensity of the emergent beam of X-rays can be predicted if the intensity of the incident beam is known.

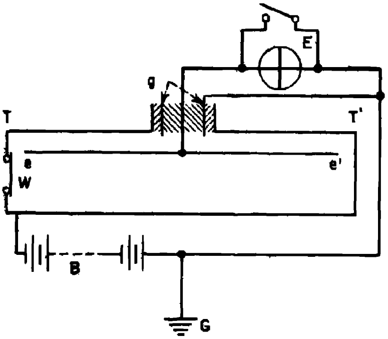

FIG. 28-14. Circuit for X-ray ionization chamber or roentgenometer.

Measurement of the ionization produced by X-rays may be used to determine their intensity, *i.e.*, X-rays ionize gases, or render them conducting, and a quantitative determination of the change in conductivity of the gas gives a measure of the X-ray intensity. It has been found that H, CO, CO_2, air, ether vapor, and CS_2 are increasingly active in the order given in their discharging action, *i.e.*, this activity increases with increase in gas density. The intensity of the beam may be conveniently measured by an ionization chamber arranged as indicated in Fig. 28-14. A metal tube T-T' has an electrode e-e' well insulated from it to prevent leakage from the electrode to the tube. The leakage is furthermore prevented by the grounded guard ring g. An electric field is maintained between the tube T-T' and the electrode e-e' by the battery B. X-rays enter the chamber through the thin aluminum window W, rendering the gas within the tube conducting, thus allowing the charge to move from the tube to the electrode. The rate of movement of the charge is determined by the electrometer E, which is therefore a measure of the rate of ionization and hence of the intensity of X-rays. Commercial instruments are offered for this purpose having the electrometer calibrated in roentgens[1] for a particular ionization chamber. This combination constitutes a roentgenometer and is often supplied with two or more scales on the electrometer calibrated to agree with an equal number of ionization chambers of different roentgen range. The instrument is a practical necessity to determine quantitatively the stray and secondary X-radiation for purposes of protecting personnel from injury.

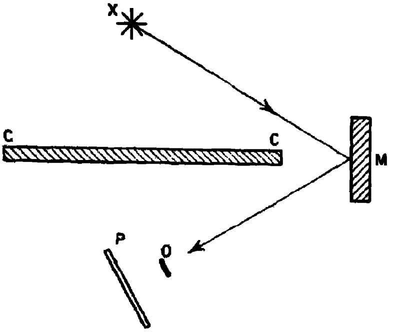

FIG. 28-15. Imbert and Bertin-Sans arrangement of X-ray screen.

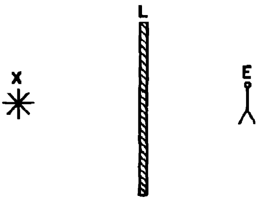

FIG. 28-16. Scattering of X-rays in air shown by discharge of shielded electroscope.

X-rays are scattered or diffused somewhat as light is scattered by fog. Imbert and Bertin-Sans arranged a screen of copper cc, a source of X-rays X, a mirror M, object O, and photographic plate P, as shown in Fig. 28-15. The position of the plane mirror was such that the beam of X-rays, if reflected, would cause a shadow of the object O to appear on the photographic plate P. A shadow was obtained regardless of the angular position of the mirror. It was also observed that a plate of paraffin was just as effective as the mirror. They reached the conclusion that the rays were scattered on striking the paraffin. At about the same time, Righi placed an electroscope behind a lead plate as shown in Fig. 28-16. X-rays from the source X discharged the electroscope E, although it was screened from direct rays. It was later determined that this phenomenon resulted from the scattering of the rays by the air.

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[1] Roentgen is the international unit of quantity of roentgen rays adopted by the Second International Congress of Radiology at Stockholm in 1928. It is the quantity of X-radiation which, when the secondary electrons are fully utilized and the wall effect of the chamber avoided, produces in 1 cc of atmospheric air at 0°C and 760 mm Hg pressure such a degree of conductivity that one electrostatic unit of charge is measured at saturation current.

The hardness or quality of X-rays is an attribute which determines the penetrating ability of the rays. The shorter the wave length, the harder the rays and the greater their penetrating ability. It may be expressed as the thickness of a given medium (*e.g.*, copper) which will absorb one-half the incident radiation. The quality of the X-rays is a function of the voltage. For a given minimum wave length there is a corresponding peak voltage given by

$$\lambda_0 = 12{,}350/V_0$$

where λ_0 is the minimum wave-length angstroms, and V_0 is the peak voltage in volts (see page 1626).

The medical use of X-rays embraces two main practices: radiography for diagnosis and surgery and therapy for the treatment of certain diseases, particularly some skin diseases and malignant or cancerous growths. For medical radiography, voltages range from 25 to 100 kv for usual work and currents from 5 to 100 ma. (Some special instantaneous radiographs may use 1,000 or more ma.) The optimum voltage and current in a particular case depend on the nature of the tissues as well as the depth that must be penetrated by the X-rays. Voltages beyond 100 kv are of little use in medical radiography. The decreased absorption under excessive voltages reduces contrast and detail in the resulting radiograph. The density of the radiograph is controlled by the intensity of the X-rays. Hence exposures are adjusted by the tube current, often called "milliamperage" by the technician; but the quality of the X-rays, which does not affect the exposure, is dependent on tube voltage which controls X-ray wave length.

A technique often employed to improve the contrast of certain transparent organs in the body is the introduction of diagnostic opaques. A common procedure in radiography of the gastrointestinal tract is to give the patient a suspension of barium sulfate in milk to drink. Other substances opaque to X-rays involve iodine salts, such as sodium iodide for the kidneys, sodium tetraiodophenolphthalein for the gall bladder. Thorium hydroxide and thorium dioxide are introduced. Some substances may be chosen for their selective absorption by the tissue to be radiographed; others may be introduced by injection at the site.

Another diagnostic technique is the photographing on small film, say, 35-mm motion-picture film, of the actinic light emitted by a large fluorescing screen exposed to X-rays. A full-sized chest plate of an adult requires a 14- by 17-in. film. A fluorescent screen may be interposed between a person's chest and a small camera and the film made to record the fluorescing chest plate. The small radiograph is then enlarged 10 or 15 diameters for study. The cost of radiography is thus reduced to a point where it does not militate against routine examinations of factory workers or school children so that tuberculosis or other diseases may be detected in their early stages.

It is very often an advantage to use intensifying screens in radiography. A very small percentage of the rays incident on the photographic emulsion— of the order of 1 per cent—is useful in changing the light-sensitive material. The remainder pass on through the film and are lost or, worse still, are scattered as they strike objects beyond the film—even air. This results in a fogging of the film and consequent obscuring of the image. This condition may be substantially improved by the use of intensifying screens. Several substances, such as barium platinocyanide, cadmium tungstate, calcium tungstate, zinc silicate, or zinc sulfide, have the property of fluorescing under the action of X-rays. A cardboard is coated with one of these fluorescent materials, usually calcium tungstate, to form a screen. In use, a photographic film is placed between two of these screens and the whole loaded in the cassette (film holder for X-ray use) in such a manner that both screens make good contact with every point of both surfaces of the film. When this assembly is exposed to X-rays, the image of the shadow is greatly improved with a reduction in exposure time by a factor of between 10 and 50. The grain size of the fluorescent material affects the exposure time. Large grains allow shorter exposure time with a decrease in detail, while smaller grains improve the detail but increase the time of exposure. Special film known as "screen film" is made for use with intensifying screens. "Nonscreen film" is made for direct use or use with lead screens.

Another method of decreasing the effect of scattering in making radiographs is by use of the Potter-Bucky diaphragm. This device consists of a grid of strips of lead alternating with a material virtually transparent to X-rays, such as wood, bakelite, or other plastic. The diaphragm has a radius of curvature about equal to the target-film distance. It is kept in motion throughout the exposure, and the timing is often interlocked with the exposure timer so that it is started in motion before the X-ray beam is started and stops after the exposure is terminated. The focal-film distance conforms to the range for which the diaphragm is made (often 30 to 48 in.) if fogging or grid shadows are to be avoided.

The geometry of X-ray shadows has been developed to a high state of perfection by medical roentgenologists as an aid to precise location of focal infection, foreign bodies, bone injuries, etc. Stereoscopic X-ray pictures have also been made and when viewed in a stereoscope yield a remarkably graphic device.

For dental use, small but powerful tubes are now mounted, together with high-tension transformer and filament transformer, in a vacuum-sealed lead case filled with oil. This construction renders the outfit shockproof and X-ray protective so that patient and operator are safe from electric shock or stray radiation. It also makes the use of dental radiography simple, convenient, accurate, and independent of climate, temperature, and humidity. Most dentists now possess an X-ray outfit as a part of their regular equipment, and diagnosis of the condition of teeth has become a precise matter.

Injury from X-rays, radium, and other radioactive materials was early found to result from exposure of living tissue to these rays. The cumulative effect of incidental irradiation received by early X-ray workers was not recognized until many of them had become true martyrs to science. Knowledge of the effects and control of radiation has now progressed to the stage where radiation in therapy and protection from stray radiation in industrial use are subject to precise measurement, and injury from either is inexcusable. It is fairly well established that interception of 1 roentgen of X-radiation (abbreviated r) per week will be tolerated by a normal person without ill effect. This corresponds to 10^{-5} r per sec for continuous exposure during a 7-hr working day. This of course applies to long-continued exposure of the body to weak X-rays. The human body may tolerate upwards of 500 r administered locally, provided a week intervenes before another dose is received. It may likewise tolerate two exposures of 400 r given 24 hrs apart. (The National Bureau of Standards Handbooks 20 on "X-Ray Protection" and 23 on "Radium Protection" afford information on this subject.) A simple method of providing knowledge of the exposure of personnel to X-rays is by means of a common paper clip fastened to the face of an ordinary dental film (not a slow one). After this film is carried for a working week by the worker, with the paper clip facing away from his body, the film should be developed and fixed. The absence of any darkening or image of the paper clip on the film indicates that the bodily exposure to radiation is safe. A slight shadow indicates borderline exposure, while a definite darkening indicates dangerous exposure. This should be regarded as a crude estimate which should be checked by actual measurement through the use of a dosemeter or mineometer, as suggested on page 1632. Furthermore, the foregoing procedure gives no knowledge of the injury that may result from exposure of parts of the body (*e.g.*, the hands) to localized radiation. Because of the greatly increased use of radioactive isotopes, weekly film-badge service is commercially available for monitoring of radiation dosage. Different-colored badges are sent to the customer each week, together with a report on the previous week's badges, which have been processed by the service laboratory after return by the customer.

Protection may be achieved by shielding the personnel with some material more or less opaque to radiation, such as sheet lead, of sufficient thickness to bring the radiation in the region to as low a value as required for safety. When sheet lead is used as a protective lining, its thickness will vary from $\frac{1}{16}$ in. for 100 kv to over 3 in. for 1,000 kv. For high voltages, from a few hundred kilovolts upward, concrete of sufficient thickness provides safety at less expense. For voltages in the neighborhood of 100 kv, the concrete of necessary thickness may be 50 to 100 times that of lead; while for voltages of 1,000 kv or so, ten times the thickness of lead may suffice. The thickness will depend on the aggregates and density of the concrete, and each case should be the subject of a separate study. Very complete protection is necessary in the industrial fluoroscopy of nuts, cereals, candy, oranges, etc., because the operators are exposed to stray radiation for hours every day, since such equipment operates continuously. Of course it is just as necessary to protect personnel against injury from gamma-ray emission from radium, radon, or other radioactive substances.

X-ray therapy soon followed the discovery that abnormal growths, whether warts or the more serious neoplasms, could be destroyed by X-rays with a dosage that the surrounding healthy tissues could tolerate. This treatment advanced to the deep-ray therapy where deep-seated neoplasms, even when adjacent to or in vital organs not susceptible to surgical interference, could be treated with much less hazard to the patient than would result from excision. It also developed that the X-ray was a valuable adjunct to surgical treatment. Irradiation of the circumadjacent organ or tissues is found to inhibit subsequent growth and spreading of excised malignancies.

Numerous workers in X-ray therapy have developed dosages for radiotherapy and geometries of application. Successive applications are often made from different angles terminating in the same spot at the seat of the malignancy, in order that this "cross firing" may traverse healthy tissue less frequently in successive applications. The dosages run from 2,000 to as high as 10,000 r and are spread over a sufficient period of time so that doses of 200 to 700 r may reach the accumulated total. The subject of X-ray therapy involves an extensive knowledge of doses and the technique of application and involves a very extended and specialized literature from which the experience and results of many workers may be learned.

Early in radiation therapy the gamma rays from radium and its decomposition products, especially radon, were used with varying degrees of success. There are many cases where such rays may still be used to advantage by locating the radiant material at the seat of the malady in what is often termed intercavity application. The radium salt is enclosed in a capsule or tube. This treatment is often combined with a more generalized radiation from X-ray beams.

Nearly all deep-ray therapy, as compared with superficial therapy, includes the use of filters composed of copper, aluminum, platinum, gold, or other material, singly or in combination, in order that the skin may be protected from the soft rays while the hard rays are projected to the deep-seated object.

The trend of deep-ray therapy for several years past has been toward higher and higher voltages. A million volts is not uncommon, and several million volts have been used. Two or three million volts approximate the quality of gamma rays. With the high-voltage Van de Graaff generators and betatrons operating at dekamillions of volts, the quality of gamma rays may easily be reproduced. Thus the quality and quantity of all natural radiation available may easily be exceeded. It is also probable that many new techniques will stem from the artificial radioactive isotopes made in the laboratories of atomic fission. One example of these possibilities is the use of radiophosphorus for the treatment of leukemia. Since phosphorus is selectively absorbed by the bones, an injection of the radioactive form will treat the blood and bone marrow at the seat of the malady. Other materials are being produced, and considerable development work in this field of therapy has been achieved.

Industrial applications of X-rays include examination and inspection of castings, welding, and food products, as well as X-ray diffraction in crystal and metallurgical analysis. The examination of castings and welds may be made either visually by means of a fluorescent screen or radiographically by photographic means. The examination of castings for production control gives the important information on molding practice in location of risers, gates, and chills, as well as temperature for pouring completeness, that cannot be achieved by sectioning. The fact that it is nondestructive gives it an advantage over the destructive methods but provides a production control that will aid in maintaining the original standards. With complex alloys it discloses shrinkage, tears, and cracks, as well as piping, gas cavities, inclusions, dross, and dendrites. This usefulness applies to steel and malleable iron, as well as nonferrous castings in aluminum, brass, bronze, or other alloys.

Since 1930, specifications of the Boiler Code Committee of the ASME have required the X-ray examination of welds of pressure vessels. Since fissures are usually in a plane perpendicular to the thickness of the weld, they may be detected in ½-in.-thick steel when they are only 0.5 per cent of the total thickness. In 3- to 5-in. steel they may be detected within 2 per cent of the total thickness. With the thicker plates, a diaphragm is necessary to prevent the scattered radiation from obscuring the trace of the defect. The destructive testing of a coupon gives no assurance that an adjacent section of the weld is without flaw; the X-ray testing of the completed weld gives assurance that the weld is as good as the test sample. Porosity, slag inclusions, and fissures are easily detected and when so located may be chipped out and rewelded. Of course, all the foregoing inspections may be carried out by radiographs which render a permanent record of the inspection.

Laws and the reputation of the manufacturer make positive and reliable inspection necessary in the food industry. It is often desirable that the completed package be examined. Here the X-ray enables the inspector to look through the object rather than at it. It is usual to have the product travel on a conveyor belt over the source of X-radiation and beneath a fluorescent screen, with the operator looking at the objects from a darkened booth or through a hood. A brilliant image of the object appears and usually with sufficient definition so that any lack of uniformity is immediately apparent. In this manner, imperfect filling of the package is recognized as well as defects or foreign bodies in the contents. Containers filled with candy, cookies, cereals, frozen fruits, or vegetables may be checked for completeness of packing or for foreign bodies such as pins, nails, stones, dirt, balls, etc., independently of magnetic or other physical properties except degree of transparency to X-rays. Bulk foods, such as nuts, beans, berries, and vegetables, may be examined for foreign materials and hollow hearts. Citrus fruits can be inspected for frost damage, crystallization, or pulpiness—a great improvement over the water-separation method, since much fruit discarded by that method would be saved. Meats are being inspected for foreign bodies and broken bones.

Another field of application assuming added importance is inspection by similar methods of fabricated articles, *e.g.*, shoes for proper nailing and lasting; rubber heels and plastics for positioning of metal inserts; wood products for knots or position of screws or other fastenings; electric fuses for loose connections; golf balls for concentricity of cores; tires for nails, wire, glass, breaks, or imperfections in the fabric. The list may also include storage-battery separators, insulating bricks, wire and cable, drugs in packages, small assemblies (for completeness), and many items of similar character. Small castings and other metal parts may be inspected on the conveyor belt in the same manner as the foregoing examples.

X-ray diffraction, which is now an essential tool of modern industry, was unknown four decades ago. Until 1912, the geometry of crystal lattices was based on knowledge revealed by visible light. The microscope, regardless of its power of magnification, cannot disclose any object smaller than the wave length of the light illuminating it. In 1912 the theoretical physicist Laue suggested that Friedrich and Knipping attempt diffraction of X-rays by means of a crystal. To this successful beginning, based on the assumed geometry of the crystal lattice, numerous experimenters have added much knowledge of crystal structure. Among these should be mentioned William Bragg and his son W. L. Bragg because they developed a method of crystal analysis which proved to be the basis of all further work. Simple structures, like rock salt and zinc blende, soon established principles of crystallography that could be extended to crystals of more complicated structure. Later the structure of metals, whose stability is based on the attraction between free electrons and positive ions, was studied. Pure metals have simple face-centered cubic structures, like copper, aluminum, and gamma iron; body-centered cubic structures, like radium, alpha iron, and delta iron; or closely packed hexagonal cubic structures, like zinc, tungsten, and magnesium. Some metals have phases of different structure, like the face-centered cubic structure of iron between 1100 and 1425°F, though it has a body-centered cubic structure outside this range. Alloys exist both as solid solutions and as compounds, and the structure is disclosed by their X-ray-diffraction patterns. In fact, our knowledge of crystals and metallic structures has nearly all been obtained by studies of X-ray diffraction.

In most cases the physical properties of a substance are determined by the arrangement of its submicroscopic particles or molecules. For example, heat-treating, cold-working, or a combination of the two will impart to a metal physical properties desirable for a specific use. By these methods their properties may often be changed through a range greater than that existing between two totally dissimilar metals. These differences are readily shown by X-ray-diffraction patterns. When one pattern is identical with another pattern obtained in the same way, the structures are identical.

Chemical properties of substances are also revealed by their X-ray-diffraction patterns. Thus it is possible to identify an element, distinguish whether it exists as a compound or a mixture, and determine whether it is amorphous or crystalline.

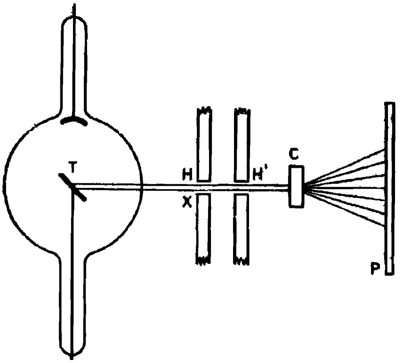

FIG. 28-17. Diagram of arrangement for X-ray diffraction.

From the foregoing discussion it is obvious that for research or production control in rubber, plastics, cements, paints, textiles, ceramics, metals, mineralogy, biology, physiology, pathology, and numberless other fields, it is possible to identify the constituents of a substance, determine their form, and predict their physical characteristics. In compounding, vulcanizing, or other processing of rubber, the structure and characteristics of the product can be determined and controlled. In rolling, drawing, or other working of metals and alloys, the structural changes attending these processes are at once determinable for production control. Precious stones and gems, natural and synthetic, can be differentiated and the best axis of orientation established for preparation for esthetic or mechanical purposes. The differentiation between normal and pathological tissues can be determined, as well as the effects of disease and medicine with their relations to bodily functions. These few illustrative cases indicate that a list of research and industrial uses of X-ray diffraction could be expanded indefinitely.

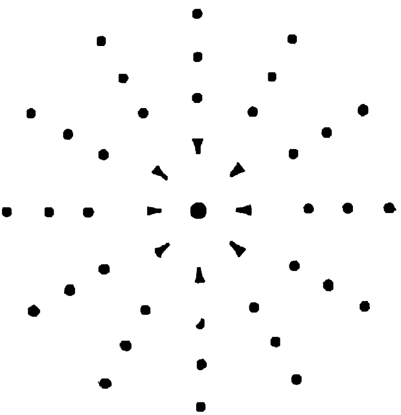

FIG. 28-18. Laue pattern.

Diffraction patterns are available only as photographs and are not suitable for reproduction in a book of this character. The interested reader can obtain numberless examples from manufacturers of X-ray-diffraction equipment. However, Figs. 28-17 to 28-20, which are diagrammatic in character, are introduced to explain the main features of this technical procedure. An X-ray source T, in Fig. 28-17, emits a beam of X-rays X through pinholes in two screens H and H' which traverse a crystal C and impinge on a sensitized surface P. The beam will be scattered as indicated, and in the case of a crystal the emergent rays will form a geometrical pattern, as shown in Fig. 28-18 or Fig. 28-19. The result shown in Fig. 28-18 was predicted by Laue and is called a Laue pattern. Figure 28-20 is a diagram of the atomic arrangement in a face-centered cubic crystal of a simple arrangement such as sodium chloride (NaCl). Figure 28-21 indicates the successive layers in this crystal. A ray of X-radiation will travel from *A* to *B*, and part of it will be reflected to *C*. Another ray travels over the path *A'B'C'*, a distance increased by the distance between the successive layers of crystals, multiplied by 2 sin θ, or formally,

$$n\lambda = 2d \sin \theta$$

where n = an integer
λ = wave length of the reflected X-ray
d = distance between the successive layers of crystalline structure

This is Bragg's equation and permits the correlation of X-ray wave length and crystal layers. By making use of X-rays having a known principal wave length, the dimensions of the space lattice in crystals can be determined. Conversely, when the dimensions of the crystal structure are known, the length of X-rays may be determined.

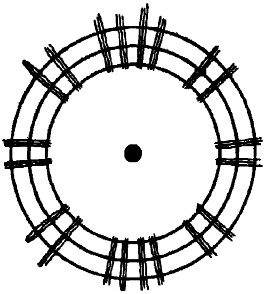

Fig. 28-19. An X-ray diffraction pattern.

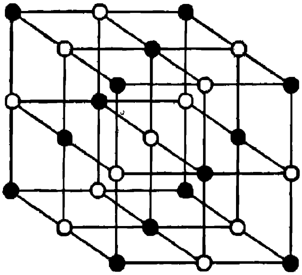

Fig. 28-20. Diagram of face-centered cubic crystal.

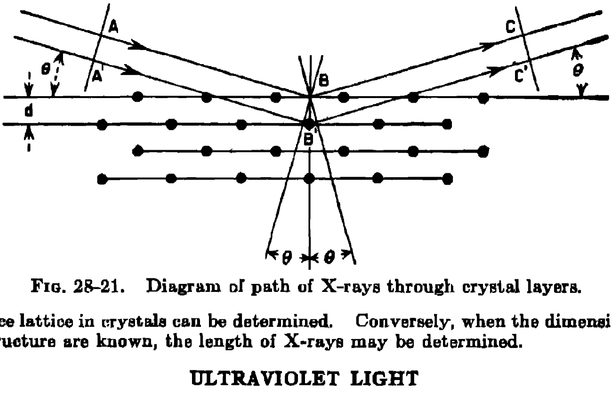

Fig. 28-21. Diagram of path of X-rays through crystal layers.

ULTRAVIOLET LIGHT

Ultraviolet light, being one of the constituents of solar radiation, has always contributed to life on earth. Foodstuffs, the fuels for all animal life, are produced by the action of sunlight on plants. Countless myriads of plant cells are vital laboratories for the synthesis of plant food. Billions of molecules in each cell are using solar energy, chlorophyll, and the cells' activities to produce food. Ultraviolet light is the chemically active constituent of solar energy. Its place in the electromagnetic spectrum is shown on pages 1624 and 1625. Light ceases to be visible at about 4000 Å, or 4×10^{-5} cm (1 angstrom abbreviated Å = 10^{-8} cm), and ultraviolet from the sun ends at about 2900 Å because of atmospheric absorption. At about 3400 Å, ordinary glass ceases to transmit any substantial quantity of ultraviolet. Quartz may be used until it, too, becomes too strongly absorbing, at about 1800 Å. Then fluorite may be used until a wave length of approximately 1200 Å is reached. Some workers have succeeded in measuring wave lengths as short as 50 Å, and the longest X-rays start in the neighborhood of half this value. The ultraviolet spectrum is ordinarily understood to extend from about 400 to 200 Å.

The emission of radiant energy results in three types of spectra: line, band, and continuous spectra. The physics of emission may be illustrated by the simplest case—line spectra from the single-electron atom of hydrogen. Three series of spectral lines were discovered, respectively, by Balmer, Paschen, and Lyman. Dark lines crossing the solar spectrum were seen by Wollaston in 1802; but Fraunhofer, who made the first measurements with a grating, recorded about 600 lines. The number has since been extended to more than 10,000. Under certain conditions the spectrum of hydrogen will exhibit a series of dark lines crossing it similar to Fig. 28-22. J. J. Balmer, in 1885, seeking to explain the regularity of the lines, found that the wave-length sequence could be described by a constant 3,645.6 which may be stated,

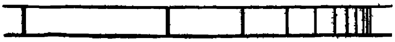

FIG. 28-22. General appearance of line arrangement in hydrogen spectrum.

$$\lambda = 3{,}645.6\,\frac{n^2}{n^2 - 2^2}$$

where n assumed the successive integral values 3, 4, 5, and 6. Rydberg later wrote the equation in the form,

$$\nu = \frac{1}{\lambda} = \frac{1}{3{,}645.6} \times \frac{m^2 - 2^2}{m^2}$$

When the numerator and denominator of the right-hand side of this equation are multiplied by 4 and λ expressed in centimeters, the equation becomes

$$1/\lambda = 109{,}720(1/2^2 - 1/m^2)$$

The factor 109,720 is known as Rydberg's constant and is designated by R. Later measurements have corrected it to 109,677.8 cm^{-1}.

Lyman, in 1906, discovered a series in the spectrum of hydrogen which may be represented by

$$1/\lambda = R(1/1^2) - (1/m^2)$$

and in 1908, Paschen discovered a series which may be represented by

$$1/\lambda = R(1/3^2) - (1/m^2)$$

Hence all the lines in the hydrogen spectrum may be represented by

$$1/\lambda = (R/n^2) - (R/m^2)$$

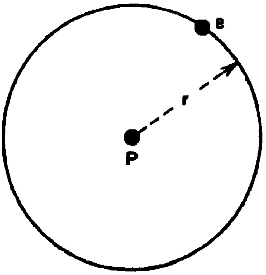

FIG. 28-23. Bohr's model of the hydrogen atom.

or the wave length is inversely proportional to the difference of its spectral terms. Bohr succeeded in devising an atomic model which gave a simple physical interpretation of these terms. Bohr's model of the hydrogen atom is shown in Fig. 28-23, where P is a central nucleus or proton with an electron e revolving in an orbit at radius r. It was assumed that

$$\Delta w = \nu h$$

i.e., that the change in energy Δw was equal to the product of the frequency of the radiation ν and Planck's constant h. Stated otherwise,

$$\nu = \frac{w_{n2} - w_{n1}}{h}$$

which gave the correct equation for the spectral series for hydrogen as well as Rydberg's constant.

Referring to Fig. 28-24, showing circular orbits for hydrogen, the normal state for the atom is at $n = 1$, or condition of minimum energy. No energy is emitted so long as the atom remains in this state. When the atom is bombarded by electrons or ions, the electron will be moved to an orbit further from the center. In a very short interval the electron returns to a lower energy level and emits energy in the form of light. In moving in to n_1, the Lyman series results; to n_2, the Balmer series; and to n_3, the Paschen series, as indicated in Fig. 28-24. The application of Bohr's theory to the more complex atoms becomes more complicated. The atomic number Z fixes the number of electrons in a given atom, and this determines its chemical behavior. If $Z = 11$, the element is sodium regardless of its atomic mass (see Fig. 28-25). Two electrons remain relatively close to the nucleus, and in a quantum state $n = 1$, or what is called the K shell. Eight electrons are in the quantum state $n = 2$, or the L shell. State $n = 3$ is the M shell. The electrons in any shell except the K shell do not have the same orbits, and the shell does not describe an energy level. The outermost electron, however, is responsible for the chemical and optical properties of the atom just illustrated. The optical spectrum of an element is a complex arrangement of many lines. The iron spectrum has thousands of closely spaced lines, and more than three thousand have been recorded for cerium.

The foregoing discussion has been applicable to line spectra resulting from single elements in the gaseous state. When elements are combined in compounds, the molecules emit more complicated groups of lines which have been called "band spectra." When energy is added to a solid so that it emits light covering a large spectral range, the lines will be suppressed or merged into a continuous spectrum. It should also be observed that, when light from a continuous spectrum is passed through an elemental gas, certain lines will be observed which agree with the lines which are emitted by the same gas. These are of the wave lengths to which its atoms are resonant and represent an absorption spectrum.

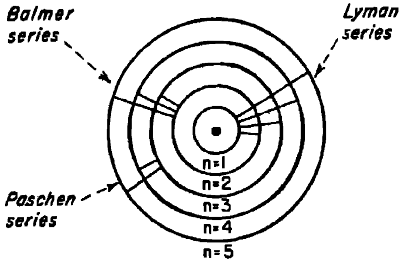

FIG. 28-24. Changes in energy state and spectral emission of the hydrogen atom.

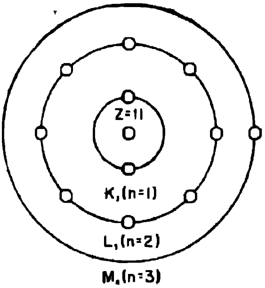

FIG. 28-25. Bohr model of sodium atom.

Sources of ultraviolet light commonly used are sun, arc, spark, and glow discharge. The sun provides an unlimited source of ultraviolet radiation, of which only the wave lengths above about 3070 Å reach the surface of the earth in the temperate zone in winter and 2970 Å in summer. The percentage of different wave lengths varies with the latitude and seasons, but a value of about 3010 Å can be taken as a fair average at a latitude of 40 deg. It has been found that ultraviolet rays shorter than 2950 Å are lethal to bacteria and destructive to living cells. In view of the fact that these conditions have existed during the development of life on the earth, they would appear to be in accordance.

Although an electric spark and an electric arc may appear to possess similarity, they are physically different. In a spark, electronic or ionic conduction takes place in the medium in which the spark exists. It is associated with ionization of the surrounding atmosphere under a high potential gradient. The spectrum partakes of the spectral characteristics of the ionized medium, and the ions may be singly, doubly, or even triply ionized. Spectral characteristics of the electrodes may be contributed as a contamination if the current is sufficiently high to cause their volatilization.

Under a sufficiently high voltage gradient with limited current, this spark between cold electrodes results in the current being carried by gaseous ions or electrons and becoming a glow discharge.

In arc conduction, on the other hand, the high-velocity arc stream is fed by vapor from the negative electrode or cathode moving at high velocity to the anode. The spectrum of the cathode material thus appears in the resultant light, and any characteristics of either the surrounding medium or the anode are incidental and contributed indirectly by heat luminescence.

Incandescent solids, such as the filament in an incandescent lamp operated at abnormally high temperature, may be used as the source of a continuous spectrum which includes the ultraviolet region. Soda-lime glass bulbs cut off the transmission at about 2900 Å, and lead-glass bulbs limit the use of this source to wave lengths above 3000 Å. High-temperature flames, excitation of gases by high-frequency electromagnetic fields, and some novel but cumbersome methods for the production of ultraviolet light have been used and found impractical.

Electric sparks are sometimes used as a source of ultraviolet radiation but mainly for the purpose of analyzing materials by spectrographic methods. A transformer with the primary wound for use on the usual power mains has a high-voltage secondary connected across the spark gap. A discharge will occur when 25,000 to 30,000 volts per cm (peak voltage) are impressed on the gap. An oscillatory or sustained discharge will occur if a suitable capacitor is also bridged across the gap, as indicated in Fig. 28-26. This condenser must, of course, have sufficient dielectric strength to withstand the maximum peak voltage. Iron or aluminum electrodes render valuable ultraviolet spectra with large numbers of reference lines. Equipment of this character is available from commercial manufacturers.

Carbon arcs date from their discovery by Sir Humphry Davy, at the beginning of the last century, who used a 2,000-plate battery and produced an arc 3 in. long between carbons. The carbon arc was an important illuminant for large areas during the last two decades of the past century where extensive areas were to be lighted. Other devices have superseded them for this use, and they are now employed where a high-intensity concentrated source is required. A highly actinic light is provided for such work as photoengraving and blue-printing where arcs are still frequently used. A d-c arc, because of the unidirectional high-velocity stream of vapor issuing from the negative electrode and impinging on the positive electrode, gives a very concentrated intense source of light. This intense spot is suitable for an application where it is to be placed at the focus of a lens or mirror. By placing the positive carbon in a horizontal position, use is made of its intense crater for such work as motion-picture projection and searchlights.

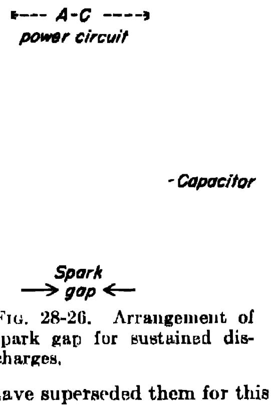

FIG. 28-26. Arrangement of spark gap for sustained discharges.

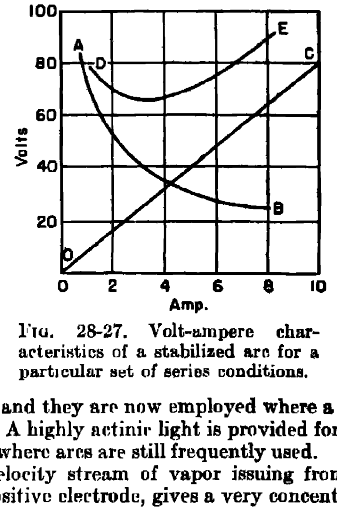

FIG. 28-27. Volt-ampere characteristics of a stabilized arc for a particular set of series conditions.

The volt-ampere characteristic of the arc should be mentioned relative to its operation. It will be observed by reference to curve *AB* (Fig. 28-27) that an increase in current accompanies a decrease in voltage. A constant voltage applied to the arc terminals results in instability over the entire range of current. The volt-ampere characteristic of a resistance (or reactance in the a-c arc) curve *OC*, as opposed to the arc characteristic, indicates a linearly increased current accompanying an increased voltage—a stable condition. The sum of the voltage at the terminals of the resistor and the voltage at the terminals of the arc, in series, indicated in curve *DE*, shows the arc voltage to decrease as the current increases until a current of 3.5 amp is reached. For any current beyond 3.5 amp, the current increases with an increase in voltage, and the circuit is stable. A greater resistance will stabilize the circuit at a lower current.

The plain carbon arc gives a spectral distribution ranging from 3000 to 15,000 Å or more, depending on current density in the arc, with the maximum energy output falling between a wave length of 5000 and 12,000 Å. Carbon arcs for industrial and therapeutic use are often formed by the use of so-called cored carbons. The carbons have a core of cerium, manganese, aluminum, iron, nickel, tungsten, chromium, titanium, or other metals or earths in some of their combinations or compounds. By this method the characteristics of the carbon arc may be made to meet specific requirements. By making use of some of the foregoing elements, a great part of the energy may be placed in the ultraviolet region, while the addition of certain other elements, *e.g.*, strontium, will move the maximum energy to the infrared region. Many carbons, covering a wide field of usefulness and various regions of the spectrum, are commercially available.

The mercury-vapor arc was brought out at the beginning of the present century by Peter-Cooper Hewitt, who applied it first to lighting. In this field its illumination produced a high visual acuity, but the color was objectionable for some uses. There are, however, still many applications, and extensive use is made of the mercury-vapor arc as an illuminant. It was soon realized that this device was a valuable source of ultraviolet light. To extend the spectrum beyond the cutoff point of glass, tubes of fused quartz were substituted. The mercury-arc spectrum extends into the extreme ultraviolet region with a line at 1849 Å. This wave length is below the point where quartz will transmit any considerable portion of the energy of the spectrum. Mercury-vapor lamps are therefore made by several manufacturers which use a quartz envelope which transmits a considerable quantity of energy in the far-to-extreme ultraviolet region. These lamps serve to meet the needs for chemical action in the ultraviolet region. Common glass transmits less and less of the spectrum in moving down toward 3100 Å where it is practically opaque to this wave length.

Therapeutic wave lengths are in the range 2950 to 3150 Å. Common glass transmits little energy between 4000 and 3100 Å, and the amount becomes less and less toward the shorter wave lengths until at 3100 Å the glass is practically opaque. The amount of energy transmitted at various wave lengths depends on the composition and thickness of the glass. Many special glasses have been developed which will transmit more or less of the spectrum in the therapeutic region. They are usually sold under names intended to connote their novel properties. Some are made in the form of windowpanes intended for use in solaria or situations where the sun's radiation may be applied to antirachitic therapy.

Many arc lamps are fitted with special therapeutic carbons for treatment of patients requiring ultraviolet therapy. An incandescent-arc combination lamp has also been developed for furnishing the therapeutic section of the spectrum—a so-called "sun lamp." This lamp is a combination of a tungsten incandescent filament and a mercury arc in argon, between tungsten electrodes, all enclosed in a special glass bulb. The special glass bulb will transmit some radiation at wave lengths as short as 2800 Å.

Photochemical processes cover a wide field in industry and are assuming increasing importance. The action of sunlight on oils in changing their physical and chemical properties has long been observed. This action can now be extended and accelerated by the use of ultraviolet light from artificial sources. Changes in physical and chemical character of various substances, in such a manner that some products will be improved and some injured, are produced by this agency. For example, olive oil is bleached by the photochemical action of light, while other oils, *e.g.*, cottonseed oil and soybean oil, may be darkened. Ultraviolet light has been used to remove objectionable taste and odor from cacao oil, and whale oil has been considerably deodorized. On the other hand, the rancidity of many products, such as potato chips, is favored by the accelerated oxidation caused by light. Artificial light may be used as a convenient means of testing for the determination of suitable packaging or coloration of wrappers that will render adequate protection against spoilage of products exposed to light.

In the textile industry it is necessary to determine the effect of light in reducing the strength and elasticity of fibers that may be exposed to this influence. It has been found that silk may lose 50 per cent of its strength on exposure to sunlight for 200 hr. Other fibers experienced decreased strength of the same order when exposed for increased times. The times increased through the series—jute, rayon, cotton, flax, hemp, wool—in the order given. The purer the atmosphere, the less the destruction.

It is general knowledge that the bleaching of textiles and paper by chlorine is accelerated by light (a fixed temperature assumed). In the textile and paper industries, tests for the rate of loss of color of dyed materials when exposed to light are necessary, *i.e.*, tests for light-fastness of dyes. Ultraviolet lamps are manufactured that are especially adapted for this use.

The testing of pigments in the paint industry falls in the same category as the testing of dyes. Accelerated weathering tests on oils, paints, and varnishes are possible by making use of carbon arcs developed for the purpose. These tests are also applied to synthetic varnishes and nitro- or ethylcellulose lacquers. Means of conducting accelerated tests on the oxidation of rubber and methods of protecting it from sunlight damage are developed by making use of artificial ultraviolet lighting sources. This is especially important in thin rubber coatings and membranes, such as those used for balloons. The spoilage of many pharmaceutical preparations is prevented by advance investigation of the effect of light and the application of suitable protection. The industrial applications of ultraviolet light for research and development are very extensive, and an extended discussion is not possible in a limited space; hence mention is made of only a few of the commonest examples.

Biological applications of photochemistry are very extensive and rapidly increasing as research in this field advances, although exact knowledge of the photochemical behavior of biological compounds is relatively limited. Work on enzymes, hormones, and products of pathological significance has been extensive. One of the important fields is the lethal action of ultraviolet rays in killing bacteria and molds. Prominent in this is the surface sterilization of foods and other expensive materials. The sterilization of air is extending from hospital operating rooms and corridors to schools and public buildings. Tests have shown that the region of bactericidal activity in the spectrum is between the wave lengths of 2960 and 2000 Å. The mechanism of the bactericidal action of the ultraviolet light (or, for that matter, any electromagnetic radiation) is not yet understood. Some interesting correlations for bactericidal action of different wave lengths have been published. There is some evidence that the absorption of a single high-velocity electron or a single X-ray quantum by a bacterium was sufficient to cause death. In the case of ultraviolet light the data seem to indicate a very low quantum yield when compared to the highly efficient X-rays. However, the ease of the use of ultraviolet light, together with its much lower deleterious effect on the higher animals, results in its extensive use as a bactericidal agent.

For a time the use of ultraviolet light for the sterilizing of drinking water seemed most desirable. The water itself would undergo no physical or chemical change, an overtreatment was impossible, the cost was not prohibitive, and the solution of air and other desirable materials was retained to preserve its pleasant taste. But since the objectionable features of chlorination have been largely removed by improved methods, the ultraviolet method has declined in favor. The extension of this method to the sterilization of milk has proved difficult, and the process is said to interfere with the production of activated ergosterol, or vitamin D_2. The process is not considered commercially feasible. Mercury-arc irradiation has been successfully used for bleaching flour. The growth of molds has been inhibited by the action of ultraviolet radiation on bread dough, and it does not interfere with the growth of the yeast cells. It is also effective in the bread-wrapping room to sterilize the baked bread and thus inhibit the growth of mold in the humid atmosphere of the package.

A considerable use of ultraviolet light rays between 2000 and 2900 Å has been made in the meat industry. Meat is benefited by aging in storage until the connective tissue within the muscle fibers is changed to gelatin by enzyme action, thus rendering it tender. But a relative humidity of 85 to 90 per cent to prevent excessive weight loss and storage above usual refrigeration temperatures favor the growth of surface mold. When mercury-vapor tubes, developed for the purpose, are included in the equipment of the tenderizing room, mold is prevented both by the direct action of the light on the surface of the meat and by the sterilization of the circulating air. The installation of the equipment in the sales refrigerator is said to reduce losses also from the deterioration of the surface of the meat.

Contamination in the operating room of a hospital is affected by the ventilation, the cleanliness of the room and the incoming air, and the number of occupants and the efficacy of their masks, as well as the contamination of their respiratory passages.

The danger in thoracic surgery is so great that operations in this region are postponed, when possible, until summer months. It has been observed that air-borne pathogenic bacteria are about ten times as susceptible to abiotic rays as the same bacteria when growing in a nutrient medium. This circumstance has made the use of abiotic or bactericidal rays effective in the protection of the patient. A battery of suitable mercury-arc lamps arranged above the operating table affords a very great measure of protection to the patient and because of its low concentration and inability to penetrate the skin will produce no known physiological effects. It has also been found that postoperative infection is avoided, wound healing is more rapid, and mortality is decreased more than 50 per cent. Moreover, the light is of a pleasant blue color but is of a negligible illuminating value. The development of the tantalum photoelectric cell, which responds to the abiotic wave lengths almost in accordance with their lethal effects, has made possible the application of this source of light with exactitude. This source of radiation is also applied to hospital corridors to prevent cross infection from transportation of infection across this barrier.

Therapeutic applications of ultraviolet light are treated in an extensive literature and will be only briefly mentioned here. Much of the literature is contradictory, which may be in part due to lack of proper limitation of the wave band concerned, either in its application or in the description of results. Ultraviolet radiation is often divided into four bands, with reasonable agreement on the division between scientific workers in the field. The four bands are fluorescent or near ultraviolet, 4000 to 3300 Å; biological or erythemal region, 3300 to 2800 Å; abiotic or sterilization region, 2900 to 2000 Å; ozone-producing region, 2000 Å to the limit of air transmission. The cutoff limit of quartz is near 1700 Å; that of air, somewhat below this wave length. The erythemal band of 3300 to 2800 Å is of therapeutic significance. There is general agreement on the broad aspects of the use of ultraviolet radiation for the treatment of such ills as rickets and surgical tuberculosis and for the improvement of the bodily vitamin D. But in other applications, such as altered circulation, changed respiration, modified composition of blood, or growth in the body, the literature is often controversial or contradictory. Proper control of wave lengths from the various sources and measurement of biological dosages may result in better agreement in the literature. Light of erythemal wave length is absorbed in very thin layers of the skin and under certain exposures results in erythema and pigmentation. Studies of the effect of light in this band have contributed to the development of many lotions and ointments for the prevention of sunburn. Most of them owe their efficacy to the incorporation of fluorescent material which at the surface of the skin changes ultraviolet rays to light of longer wave length. It is interesting to note that the lens of the eye filters out ultraviolet rays and thus protects the retina from their action. The use of heliotherapy in the correction of tuberculosis of bones and joints seems to be established, but there appears to be no unanimity as to the wave length of the curative ray. Even in antirachitic therapy one can only gather from the literature that an average of many opinions places the wave-length active in this sphere at an average value of 3080 to 3130 Å. In poultry husbandry it has been found that the hatchability of eggs and bone growth in chicks are improved by even short-time daily exposures of the flock to ultraviolet light. This assumes increased importance as the congested and artificial conditions of poultry raising increase.

INFRARED RAYS

The nature of infrared light will be evident by reference to the table on page 1624 and the previous treatments on X-rays and ultraviolet light. It is seen as electromagnetic radiation, to be classed with cosmic radiation, gamma radiation, X-radiation, ultraviolet radiation, visible radiation, short-wave radiation, and radiation in the broadcast band. It fills in the blank indicated in the series—the blank between the longest visible waves and the short Hertzian waves. The infrared starts where the long waves in the visible spectrum leave off, at 7600 Å, and extends to 500,000 or even 1 million Å. When the solar spectrum is considered, the wave lengths are 7600 to 30,000 Å, but terrestrial sources have yielded much longer wave lengths. The region of the spectrum in which we are concerned has the approximate energy distribution indicated in Fig. 28-28 for the five sources shown. Wein's displacement law shows that the maximum radiation intensity is inversely proportional to the absolute temperature of the black body, or

$$\gamma_{max} =$$

where γ_{max} is the wave length in microns at which the intensity of radiation is a maximum, and θ is the temperature in degrees Kelvin. These curves are approximate for the sun with a temperature of 5800 K; tungsten lamp, 2500 to 3000 K; carbon lamp, 2000 to 2200 K; flatiron, 525 to 600 K; boiling water, 373 K.

The industrial source of infrared rays is commonly incandescent lamps operated at a temperature that gives an economical production of heat energy without excessive lamp cost. Gas-operated infrared generators are used to a limited extent, mainly where a high heat concentration within a limited area is required. The incandescent lamp especially made for the purpose may be either tungsten or carbon filament. They respond to a need first brought to the attention of industry by a large application of this source of energy to the drying of automobile finishes by the Ford Motor Company in 1938. The shorter the wave length, the greaer the energy output, but in the case of incandescent lamps this is at the expense of lamp life. The filament, either tungsten or carbon, sublimates and is deposited on the inner surface of the bulb, and this process continues until the filament parts or "burns out." Photoflood lamps have an average life of 2 to 3 hr; projection lamps, 10 to 50 hr; general-purpose lamps, 750 to 1,000 hr; and street-lighting lamps, 3,000 hr when operating under rated conditions. They operate at successively lower temperatures in the order mentioned. Since the infrared lamp is designed to operate with its maximum output at as long a wave length as will result in good economy, its operating temperature is a few hundred degrees below that of the lamp designed for illumination, and the life consequently is extended to a 5,000-hr rating. Vibration and shock reduce the life of infrared lamps as they do that of illuminating lamps. Carbon filaments, for the same voltage, are shorter than tungsten filaments; hence the configuration of the finished filament may be made to approach more nearly a point source. When the filament is placed at the focus of a suitable reflector, a more uniform distribution of the light will result. The carbon filament, however, is more fragile, its life is shorter, and the energy output during its life is less than for a tungsten filament when referred to a given input.

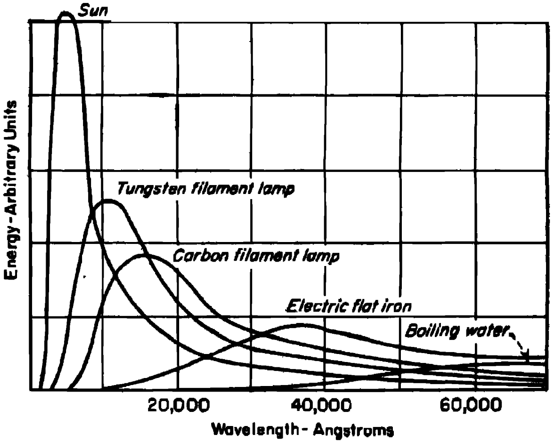

FIG. 28-28.

There are two general types of infrared lamps: those requiring a separate reflector and those with the reflector self-contained and sealed in. Those with the self-contained reflector are made in 250- and 375-watt sizes, both in an R-40 bulb. Common sizes without self-contained reflectors are 100-watt lamps in an A-23 bulb, 250-watt in a PS-30 bulb, 250- and 375-watt in G-30 bulbs, and 500- and 1,000-watt in a T-40 bulb. All but the 500- and 1,000-watt lamps may be used in the medium-screw porcelain receptacles, while the 500- and 1,000-watt lamps require medium bipost receptacles or sockets. Variation in applied voltage has the same general effects on output, life, efficiency, current, and power as it does on general-purpose illuminating lamps. The cheapest and best lamp for use, from the point of view of lamp cost and operating cost, is the 120-volt lamp. For use on 240- or 480-volt circuits it is generally cheaper to connect two or four 120-volt lamps of the same rating in series. Groups thus made up for series-multiple operation will have a somewhat shorter life than when burned in multiple on a circuit of their rated voltage. The life and operating efficiency, however, would exceed that of lamps specially made for the higher voltage and furthermore would have a lower first cost and lower operating cost. This is because of the fragility and lower efficiency of filaments made for the higher voltage. The approximate shape and size of the R-40 bulb with sealed-in reflector are shown in Fig. 28-29. Other bulbs, except the tubular with bipost base, are like those used for general-purpose illuminating lamps.

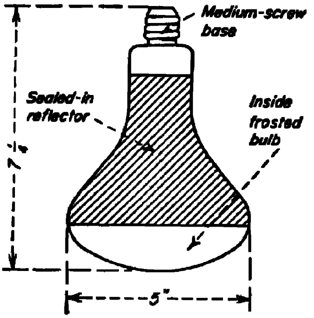

FIG. 28-29. R-40 infrared lamp with sealed-in reflector.

Gas-fired infrared generators usually consist of a refractory bowl or cup made of a ceramic or silica brick. In the center of the cup is placed a gas burner operated with air-pressure flames which leave the burner radially and scrub the refractory bowl, heating it to incandescence. They will release between 3000 and 5000 Btu/hr/sq in. of projected area. This is from 50 to 80 times as intense a heat as the 375-watt lamp will give. It fills a need differing from that met by the incandescent lamp. The gas-fired infrared generator is suitable for such work as brazing, annealing, hardening, forging, or melting of small volumes of metals. It is developed to concentrate a large amount of energy in a small space. It would also be suitable for use where the object to be heated moved so fast that electric heaters could not provide the necessary intensity of heat.

The equipment for the industrial use of infrared rays is in many respects similar to lighting equipment. It includes lamp holders, reflectors, current supply, means of getting the work to the radiant heat, and sometimes means of removing noxious fumes. The lamp holders must stand continuous high temperature. To have a useful length of life, the sockets must withstand continuous elevated temperatures. The insulation and resiliency of contacts must be maintained. Porcelain receptacles meet these requirements when lamps smaller than 500 watts are to be used. For the 500- and 1,000-watt sizes, which have medium bipost terminals, no suitable sockets have yet been developed. It is better at present to braze leads to the posts, using for the purpose asbestos-covered or other heat-resistant wire. These leads may then be connected to suitable terminals somewhat removed from the heat.

Coefficients of reflection vs. wave length for five of the most suitable reflector materials are plotted in Fig. 28-30. A good visible-light reflector is not necessarily a good infrared reflector. Not only has electrolytic gold, as plated on a metal surface, a good coefficient of reflection for infrared rays, but the surface retains its reflecting value well at temperatures that attend the use of infrared. Vaporized aluminum or silver, deposited on the reflector surface, is secondary in value and is used in the R-40 bulb. This surface operates in an inert atmosphere and retains its efficiency for the life of the lamp except as it is deteriorated by the deposits of the sublimated filament. Alzak stands next to gold for permanence and high reflectivity when exposed to the oxidizing influence of the atmosphere at high temperatures. The surface of the aluminum reflector is oxidized when manufactured. This protects the surface, which maintains a high reflecting coefficient after having been oxidized electrolytically. The specular finish has from 80 to 85 per cent coefficient of reflection, while the diffuse finish has a coefficient of 75 to 80 per cent. The specially prepared glasslike surface permits easy cleaning without damage.

The high production rates achieved by the use of infrared heating derive largely from the use of conveyors for moving the material, unless of course the material is in strip form, as paper, textiles, etc. For this rapid rate of handling, the work is carried through a tunnel of concentrated heat on a conveyor or overhead carrier. The lamps are preferably arranged so that uniform heat is delivered to the faces of the work by proper spacing of the lamps in the banks, and by making adjustment of the distance from the lamps to the work. For many kinds of work the conveyor is a belt of the mesh type, with the size of the meshes of course suited to the work. Where the work must be rotated in traversing the oven or tunnel in order that all faces of the work may be presented to direct rays from the lamps, the overhead type of conveyor is especially desirable. Where the speed of the heating process is subject to change, the conveyor drive must permit the adjustment of its speed to meet this requirement.

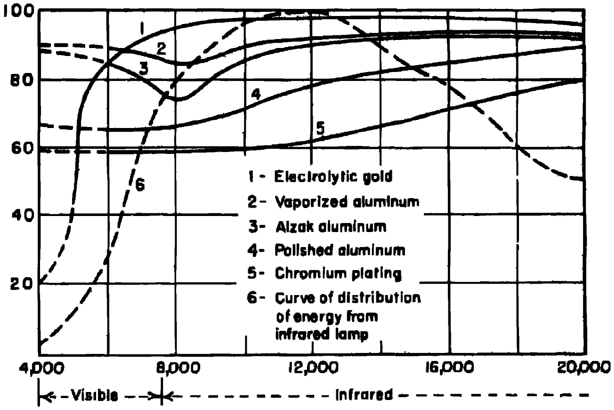

Fig. 28-30. Coefficients of reflection for some useful reflector materials.

The majority of installations are of the open type, but where noxious or flammable fumes are encountered, forced ventilation may be necessary. However, a satisfactory method of ventilating a great many installations may be to provide a hood above the lamp banks with a properly located stack and natural convection to discharge the vapors to the out-of-doors. Commercial infrared equipment is available which extends from a simple row of lamps, properly mounted and spaced, to a complete system of banks, tunnels, or ovens, complete with conveyors and adjustable-speed drives.

In planning an infrared installation, use is made of fundamental principles in heat calculation. For example:

A toy factory requires that steel wheels weighing 5.4 oz each be heated from room temperature to 150°F before tires are applied at the rate of 4,000 wheels per hour.

Specific heat of steel = 0.107
Room temperature = 68°F
1 kwhr = 3412 Btu
Weight of work = (5.4 × 4,000)/16 = 1,350 lb per hr
Energy required = 1,350 × (150 − 68) × 0.107 = 11,850 Btu per hr
Power required = 11,850/3,412 = 3.47 kwhr

Another example is as follows:

A textile plant requires that 100 yd per min of cotton cloth weighing 8 oz per yd have 2.3 oz of water evaporated from it when going into the drier from a room at 77°F. It is considered desirable to superheat the cloth 10°F.

Specific heat of cotton = 0.32 (approx)
Specific heat of water = 1.00

Heat of vaporization of water = 970 Btu per lb

Weight of cloth per hour: $(60 \times 100 \times 8)/16 = 3{,}000$ lb per hr

Weight of moisture per hour: $(60 \times 100 \times 2.3)/16 = 865$ lb per hr

For evaporation of moisture:

Raising temperature of water:

$$865 \times (212 - 77) \times 1 = 116{,}700 \text{ Btu per hr}$$

Evaporating the water:

$$865 \times 970 = 839{,}000 \text{ Btu per hr}$$

For heating the cloth: $3{,}000(222 - 77) \times 0.32 = 139{,}000$ Btu per hr

If the efficiency of vaporizing the water is 30 per cent,

Heat input for water $= (116{,}700 + 839{,}000)/0.3 = 3{,}185{,}000$ Btu per hr

If the efficiency of heating the cloth is 50 per cent,

Heat input for cloth $= 139{,}000/0.5 = 278{,}000$ Btu per hr

Energy input required $= 3{,}185{,}000 + 278{,}000 = 3{,}463{,}000$ Btu per hr

Power input required $= 3{,}460{,}000/3{,}412 = 1{,}015$ kw

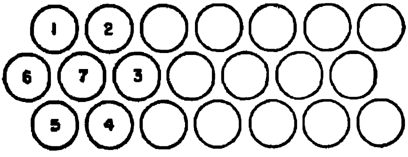

Fig. 28-31. Equilateral triangular lamp spacing.

In the foregoing examples the actual heat needed to accomplish the work was stated. Then efficiency was assumed and input to the lamp bank computed. In the first problem a certain mass was to be raised to a certain temperature in a certain time. The input to the lamps would depend on the efficiency of the lamps in converting electrical energy into infrared energy, on the efficiency of the surface used as a reflector, on the transmission losses between the lamps and the work, and on how well the work was arranged to intercept the heat rays. Approximately 10 per cent of the output of the lamp is in the visible spectrum; about 10 per cent of the input is converted from the bulb by the surrounding air; in the neighborhood of 2 per cent of the input is conducted to the base of the lamp and dissipated through the socket. Hence it may be expected that between 75 and 80 per cent of the input to the lamp, depending on the size, will be turned out as useful heat, and this is true. Reference to Fig. 28-30 will show that if an Alzak reflector is used, an average of some 80 per cent of the infrared will be reflected. This, then, will indicate that a maximum of $0.75 \times 0.80 = 0.60$, or 60 per cent of the input to the lamps, will be projected toward the work. When heating a mass, an efficiency of 50 per cent may be considered good performance, although in a closed tunnel or oven this over-all efficiency may be exceeded somewhat because use is made of the incidental heat convected by the atmosphere. When moisture is evaporated, there is a decreased efficiency, and 35 per cent may be considered good performance. In the example of the wheels, $3.5/0.5 = 7$ kw input would be a minimum requirement. Likewise in drying the cloth,

$$370/0.35 = 1{,}050 \text{ kw}$$

input would be the least required.

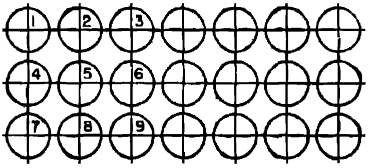

Fig. 28-32. Rectangular or square lamp spacing.

Infrared lamps should be arranged in uniform spacing, as shown in Figs. 28-31 and 28-32. The spacing in Fig. 28-31 is somewhat to be preferred to that of Fig. 28-32. Wiring of the lamps in the rectangular arrangement is easier than in the equilateral triangular arrangement. Essentially uniform heating may be had with either arrangement if the lamps are placed at the proper distance from the work, but with the equilateral arrangement uniform illumination will be secured at a shorter distance. A test bank may be readily assembled, with either arrangement, using lamps 1 to 7 in Fig. 28-31 and 1 to 9 in Fig. 28-32. Since infrared radiation obeys the laws of visible radiation, since both are generated at the same source, are projected together, and impinge on the same surface, the distribution of the light may be readily seen. In irregular work it is necessary to eliminate the shadows, which are easily seen. For a more exact survey, a meter similar to a foot-candle meter used in photography, but sensitive to the infrared range, is available. It is scaled to read in gram-calories per centimeter per minute. These units are of course easily converted into any desirable units of energy per unit surface per unit time, such as Btu per square foot per hour or kilowatts per square foot. The necessary degree of uniformity depends upon the application. To dry the surface of a thermally nonconducting material would demand much more uniformity than that of a metal where heat distribution would be aided by its conductivity. For test purposes, in connection with design, the R-40 self-reflecting infrared lamp will be found to be convenient and adequate. Arrangement and rating of larger lamps can easily be deduced from the test results found with the bank of R-40 lamps.

Surface finishing constitutes the most extended use of infrared heat because it is possible to achieve more rapid heating and an increased temperature of the work. This accelerates the three processes involved in curing a finish, *viz.*, evaporation, oxidation, or polymerization. Some finishes require only evaporation of the solvent, as lacquers, shellac, and other solution coatings. Others require oxidation in addition to evaporation of the solvent, and most oxidations are accompanied by polymerization. This is true particularly of synthetic or modified synthetic finishes. Many of these finishes are heat-hardening or polymerizing and are completely processed when they reach a final temperature. The rate of processing a finish is affected by the rate of heat absorption, which is at the highest rate with black, at the lowest with white, and with colors at an intermediate rate.

Industrial applications of infrared were first used in connection with the heating of surface finishes. The speed that can be attained by its use often pays for the increased thermal cost over other competing methods. It is not uncommon to have the time of a process cut to 10 or even 5 per cent of that required in convection ovens. Its use in textile plants up to the present has been principally as a booster or as a supplementary operation. It is convenient in this use to add or take away lamps as variations in the process demand. In the paper industry, much the same state exists. In ceramic industries, it is often replacing the biscuit oven for drying ware before it is glazed, as well as having utility in the decorating shop. The infrared driers provide a convenient and time-saving device for skin drying of green molds in the foundry. The heat can easily be taken directly to the mold, the risk of damage by movement of the green mold to the source of heat being thus avoided. There are doubtless many uses in the food industry and in other industries where the extension of its use would result in convenience and savings.

The therapeutic use of infrared rays extends back to the days when carbon-filament lamps were used for lighting. Physicians found the penetrating power of this radiation superior to various forms of conduction heat, such as that from hot-water bottles or heating pads. They have for some time past used the larger sizes of lamps, 1,000 watts or so, in clinics and hospitals. Many special forms of infrared generator for therapeutic use have been manufactured for years. The nonluminous heater for the production of wave lengths of 15,000 to 150,000 Å usually consists of a resistor operated at 750 to 1500°C, mounted in a suitable reflector or arranged in an ovenlike structure which may be used to surround the extremities or the torso as required. These oven devices are intended for superficial heating, since the long wave lengths supplied by them penetrate the tissues by no more than 0.1 to 1.0 mm. For deeper heating, the incandescent lamp is used in sizes from 60 watts to 1,000 or more. The maximum penetration of human tissue is at about 11,000 Å. By Wein's displacement law it may be computed that maximum emission of energy at this wave length is from a source operating at a temperature of 3550°C. This is not far below the filament temperature of the infrared incandescent lamps, which therefore furnish an excellent source of radiation for deeper heating in thermal therapy. This source, however, emits a considerable portion of its energy in the wave-length band beyond 15,000 Å, which is in the range of superficial heating. For the deeper heating the longer wave lengths are not wanted, since they cause discomfort. They may be readily eliminated by interposing a few-millimeter screen of water between the lamp and the tissues. This allows the lamp to be brought almost into contact with the body surface without discomfort, and this increases severalfold the penetration of the tissues by the shorter wave lengths.

The most evident physiological effect of infrared radiation on the body is vasomotor. When this heat is used, the blood vessels are dilated, and the circulation in the area increased. Much of the beneficial effect doubtless dervies from this circumstance. In addition there are chemical changes in the blood, the formation of tissue fluid which may leave the area as lymph, and changes in tissue metabolism. Some of these effects may be of such a nature in a particular inflammatory condition—redness, heat, pain, swelling—that the use of infrared is contraindicated. The estimation of its value requires a consideration of the possibility of capillary stasis and venous congestion. Any lesion of the sensory nerves requires that the infrared be used with the utmost caution. In every case the intensity of the radiation should be well within the tolerance and the sensation of heat comfortable.

POLARIZED LIGHT

The nature of polarized light should be reviewed before its uses are discussed. At the beginning of the last century it was considered that light consisted of longitudinal wave motion traveling in straight lines from its source through a transparent medium that pervaded all space. A quarter century later, the discovery of polarization of light acted to deny the theory of its longitudinal wave motion, but further study made use of this very phenomenon of polarization to establish the wave properties of light. Its early practical use was limited to mineralogy and crystallography. Its use as a technical tool is extending as more of its nature is revealed and equipment to make further use of it is developed.

At the risk of oversimplification of the actual facts, consider the illustration that will apply to Fig. 28-33. As the man's arm is waved up and down, sidewise, or at any angle, a wave moves along the rope from the man toward the post. When the motion is vertical, the vertical pickets do not limit the wave motion, but the horizontal ones do. When the two sets of pickets are at right angles, no wave will pass through both. Obviously if the vertical barriers were turned at any angle, the wave traveling at the same angle would pass through and be entirely stopped only if the direction of the pickets in the third barrier were athwart the motion of the wave. The first fence acts as a polarizer, the action of the second one is nil, and the third fence acts as an analyzer.

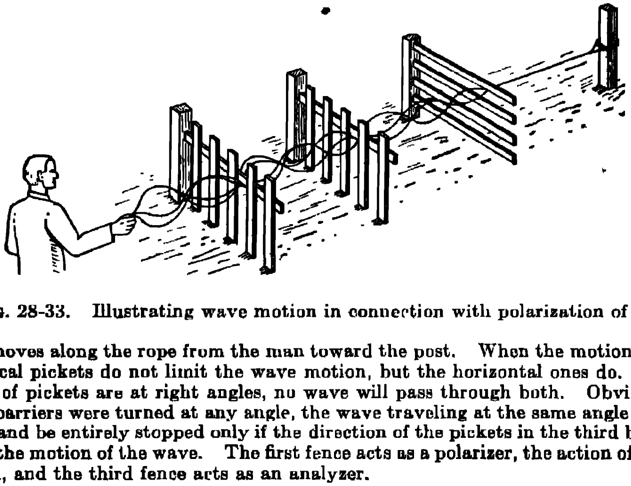

FIG. 28-33. Illustrating wave motion in connection with polarization of light.

The mechanism of light and its polarization is much more complicated than the simplified illustration offered in the foregoing paragraph. When light originates at an incandescent source, all wave lengths of visible light (as well as some of infrared and ultraviolet) are propagated at all possible angles and at many amplitudes. Vertical components of this light will pass through a polarizer with its polarizing axis placed vertically, and horizontal components will be cut off. If a second polarizer is placed beyond the first, with its polarizing axis located vertically, light will be passed as through the first one. If the second polarizer is placed with its axis horizontal, all the light will be extinguished by it. If the second polarizer is turned at any angle between vertical and horizontal, a partial quantity of the light will be transmitted depending on the angle. The second screen is called an analyzer. These arrangements are shown in Fig. 28-34. It will be observed that monochromatic light—light of a single wave length—is much less complicated in its physical aspects than the heterogeneous white light. Also, that polarized white light still possesses the heterochromatism of white light.

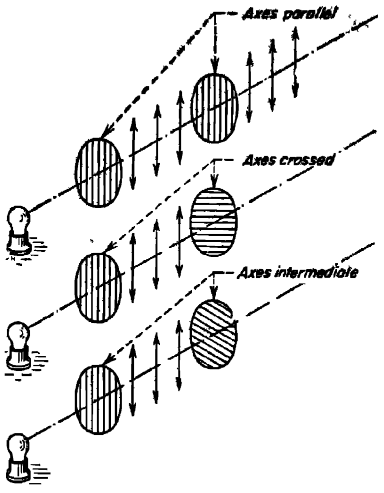

Fig. 28-34. Polarizing and analyzing screens.

In Fig. 28-35, *V* represents a vector diagram of light waves of a single amplitude but random angles in a ray of light. The block *AB* represents a polarizer, such as the anisotropic crystal tourmaline, which possesses the property of double refraction and hence will polarize light. The unpolarized light enters the crystal at *i* and emerges at *O*. All vertical vibrations are transmitted with some absorption; the vertical components of the inclined vibrations are also transmitted in the same manner; the horizontal vibrations are absorbed; and a beam of plane-polarized light emerges at *O*.

Nicol prisms were originated in 1828 by the Scotch physicist, W. Nicol. The prism consists of an Iceland-spar crystal cut and cemented with Canada balsam in such a way as to deflect the ordinary ray and transmit the extraordinary ray. This was the most common polarizer until a few years ago and is still used where the best precision is required. In Fig. 28-36 the plane faces are on the plane *abcd*, and this figure also shows the paths of the rays. In 1852 an English physician, W. B. Herapath, discovered that crystals of sulfate of iodoquinine, later called herapath, would polarize light very effectually, but the crystal would disintegrate into useless dust at the slightest shock. About seventy-five years later, Edwin H. Land, while an undergraduate at Harvard, devised a scheme for preserving innumerable tiny crystals of herapathite in a plastic-film matrix which would also maintain their proper orientation and alignment. This film is usually mounted between glass plates for protection and ease of handling and is the Polaroid now commonly used in most applications where the use of polarized light is embraced.

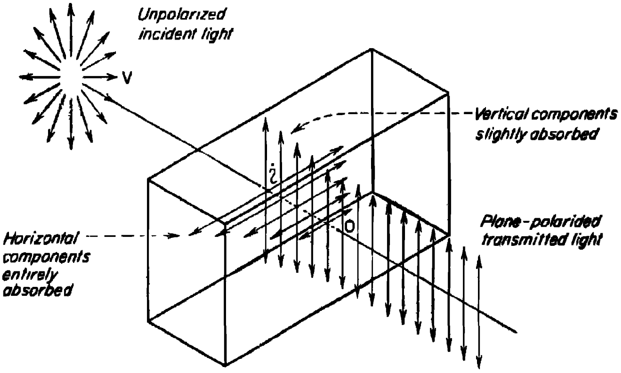
Unpolarized incident light
V
Vertical components slightly absorbed
i
O
Plane-polarized transmitted light
Horizontal components entirely absorbed

Fig. 28-35. Polarization by a crystal polarizer.

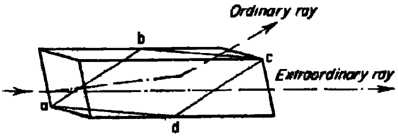

Fig. 28-36. Nicol prism.

There are many natural occurrences of the polarization of light. Light is polarized by its impact on a glass surface at an angle of 57 deg from normal. The index of refraction of ordinary glass is 1.54. The polarizing angle then becomes 57 deg when the simple equation

$$n = \tan \alpha$$

is considered, where n is the index of refraction, and α is the polarizing angle. Polarization by reflection is indicated in Fig. 28-37. This polarization takes place also when light strikes the top of a desk, drawing board, book, linoleum, concrete pavement, roadway, and other nonconducting surfaces at an inclined angle. The extent of polarization is greatest when the angle is near the polarizing angle for glass. This type of polarization is not usually recognized; for to be observed, it must be seen through an analyzer or polarizing screen properly oriented. Most of the surfaces named are usually seen in a horizontal position; hence, for extinction, the polarized rays must be viewed through a polarizing screen with its plane normal to the angle of reflection and its polarizing axis vertical. This phenomenon is the reason for the use of polarizing screens arranged as spectacles to eliminate or reduce glare on streets and roads. It is also the reason for controlled polarized illumination for inspecting benches, drafting tables, desks, and similar surfaces. The simple, relatively rugged screens of Polaroid of large area now available render such applications practical.

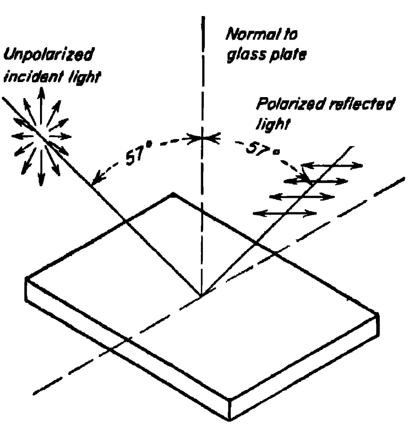

FIG. 28-37. Polarization by a plane glass surface.

Many years ago it was proposed that the discomfort resulting from glare in night driving be eliminated by the use of polarized light. At that time no device for bringing it about was available. The invention of Polaroid a few years ago makes this improvement in night driving feasible. In its adoption, polarizing screens would be applied to head lamps and windshields of cars, with the axis of the screen at 45 deg from the horizontal and inclined, say, from the operator's lower left to his upper right. This would place the polarizing screens of any oncoming car at right angles to the operator's viewing screens. The operator would see objects lighted by his own lights, but the polarized light from the headlights of the oncoming car would be stopped by his own polarizing windshield. Automobile-headlight glare could be eliminated if all cars were equipped with polarizing equipment. The combination of Polaroid and sealed-beam headlights makes this desideratum possible, but automobile manufacturers must provide the equipment and double the size of the present generators in order to initiate the change.

Small particles in a liquid or gas will "scatter" light waves and cause polarization. Short waves are scattered more than long ones; hence the sky appears blue to the observer. If a bright blue sky be observed through a polarizing screen while the analyzer, or viewing screen, is slowly rotated, the light will be found to be noticeably polarized.

Applications of polarized light fall into two categories: one where the initial polarizing process is natural, the other where the polarizing process is artificial. In both cases another polarization process, by means of an analyzer, is required to identify the polarization.

Sunglasses are really analyzers with the polarizing axis vertical and serve to reduce the glare by eliminating the natural horizontally polarized component.

When the light causing glare is from an artificial source, as in an office or drafting room, the glare from horizontal polarization may be eliminated by using only vertically polarized light for illumination.

Stereoscopic pictures may now be made in pairs, with light polarized at 90 deg for the making of each pair. A process has now been developed, called the vectograph, in which the picture with one direction of polarization is on the front of the film, the other on the back. When projected on a nonpolarizing screen and viewed through polarizing spectacles (really analyzers), the observer's eyes fuse the two images into a single three-dimensional impression.

Polarizing filters that may be rotated, when applied to binoculars, enable the user to eliminate the glare reflected from water or sky, as well as to adjust the density of the transmitted light to the brilliance of the scene being viewed.

Much of the glare from the sea, or other bodies of water, is highly polarized. Windows equipped with polarizing filters will permit the vision of a water surface with greater clarity and comfort.

Railway-coach and airplane-cabin windows have been fitted with vertically polarizing disks fixed in position to stop the glare reflected from horizontal surfaces outside. An inner disk, controlled by a knob enabling the passenger to rotate it, was added, and the combination provided a variable-density window giving him any amount of brightness control desired. Variable-density sunglasses have been made with double lenses operated on the same principle.

Polarizing screens are now available for use over camera lenses, and by their use sky brightness and glare may be controlled in the manner described for similar processes in the foregoing paragraphs.

Several applications in the ocular field have been made. One is an improvement that eliminates disturbing reflections in the ophthalmoscope; another device permits independent images to be formed in the patient's two eyes for the purpose of detecting an unusual eye defect; and a device has been made to test the sight of malingerers who claim that the sight of an eye is destroyed when it is really in good condition.

A gunsight has been developed which utilizes the interference pattern created by a section of calcite or sodium nitrate between two crossed polarizers. When the sight is used, the object is made to appear in the center of a series of rainbow-colored concentric rings if the gun is trained on the object seen through the center of sight.

Polarimeters for determining densities of many chemical solutions by their optical activity—particularly sugars, when the instrument is called a saccharimeter—have been in use many years. The polarizing microscope has likewise been an important tool for the mineralogist and crystallographer, and much of the useful knowledge of minerals, gems, and crystals has been developed by the use of this precise instrument of measurement.

Photoelastic stress analysis is an important use of polarized light. An isotropic material becomes doubly refracting when submitted to unsymmetrical strain. A common example of unsymmetrical strain is produced when glass cools spontaneously. Strain sufficient to cause it to fly apart is often produced when the object is spontaneously cooled. Annealing relieves the strain, and the glass is then isotropic. Imperfect annealing is readily detected when the glass object is viewed in a polariscope. The polarized light is supplied by a suitable illuminant and a polarizing screen. The glass object is placed so that the polarized light passes through it, and it is viewed through an analyzer which may be another polarizing screen or a pair of polarizing spectacles. The appearance of light and dark bands indicates a condition of stress. These stresses are set up in the glass as it contracts on cooling and can be prevented by cooling the article slowly. If stresses exist in a finished product, they may be detected by a polariscope and removed by annealing.

Photoelastic stress analysis or photoelasticity represents an extension of the strain testing of glass. By this method, stress computations may be checked or stresses in an object analyzed when computation is too lengthy or cumbersome. The method is to make a model of the article from some transparent isotropic material such as bakelite BT-61-893, catalin, or celluloid. This model is then loaded and viewed between crossed polarizers. In a simple flat object like a beam or ring, where two-dimensional analysis will suffice, the stress bands are readily viewed or recorded photographically and studied. If a dissymmetrical object is to be studied in three dimensions, it is possible to load a suitable model made from a thermosetting plastic while in a heated condition, cool it while under load, and thus freeze the stress patterns into the model. Thin sections may then be made, polished, and studied. This method would also be applicable to the study of dynamic stresses due to rotation. The model, while hot, would be rotated and cooled to room temperature while still stressed. In most cases, however, sufficient knowledge may be gained from the study of representative sections in two dimensions. The actual thickness of the sections is immaterial, but a common thickness employed is ¼ in. The faces of the model should be polished for good resolution, although satisfactory results have been obtained by other methods, such as coating the surfaces with oil or lacquer. Sheets of plastic with polished surfaces may be obtained from manufacturers or dealers. The edges of the model must be carefully machined perpendicular to the face. It is good practice to rough the piece out to within 1/32 in. or so of the boundaries and then complete shaping of the remainder by light cuts, avoiding heating. For the most accurate stress determinations, the model should be analyzed a few hours after completion before the edge effect may appear.

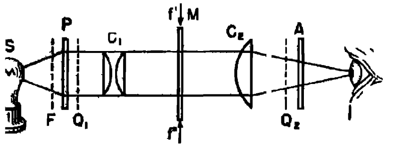

Fig. 28-38. Stress analysis in a plane model by means of polarized light.

When the loaded model is viewed in a polariscope, bands will be observed following the lines of constant principal stress. They result from the material becoming birefringent when stressed. Plane polarized light will be split into two components, each parallel to a principal stress: one, tension or compression; the other, shear. These two components travel through the stressed material at different velocities. If on emergence they are recombined by an analyzer, they produce isochromatic lines because they are out of phase. The amount that the components are out of phase depends on the magnitude of the stress in the model at that point. Where the plane of the entering polarized light coincides with the direction of a stress, that ray passes through the model without change in direction and

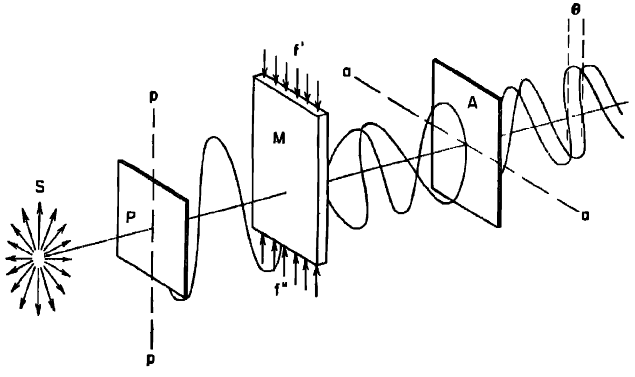

Fig. 28-39. Path of light waves in polarized light stress analysis.

is cut out by the analyzer, thus showing a black line. The black line formed is isoclinic. The other component shows bands of color when white light is used, owing to the merging of the successive isochromatic lines. When monochromatic light is used, the isochromatic lines appear black, and resolution of stresses to higher orders is made possible. With the use of either white or monochromatic light, bands appear along the lines of constant stress. If a given load produces one band in the object, double the load produces two bands, and each equal increment produces an additional band moving outward from the point of origin. The first line that appears is said to be of the first order; the second, of the second order; etc. When heterochromatic light is used, the colored bands tend to become confused at about the sixth order. By making use of monochromatic light, lines of an order as great as the twelfth or fifteenth may be resolved. This nears the elastic limit of most photoelastic materials.

In Fig. 28-38, a source of light S projects light through a polarizer P. The condensing lens C_1 directs the light in parallel rays through the model M, loaded with a force f'-f'', to the condensing lens C_2 which converges the rays passing through the analyzer A. An eye or camera at E will receive the rays. A filter may be interposed at F to introduce a band of monochromatic light, or a suitable monochromatic source may be introduced at S. At Q_1 and Q_2, quarter-wave-length plates or wedges may be used to remove the isoclinic lines when only isochromatic lines are wanted. When only isoclinic lines are wanted, a rapid relatively fine-grained panchromatic plate or sheet film capable of responding to the color of the monochromatic source may be used in the camera at E. For records of the colored lines, it is of course necessary to use a color-sensitive material. The three-color mosaic screens were formerly used, but now one of the many color films may be used. The diagram in Fig. 28-39 supplements Fig. 28-38 in illustrating the essentials of the process.

MAGNETISM

The lifting magnet appeared in industry about the year 1900. Three types have evolved: the circular or annular type, the rectangular type, and the bipolar type. Their construction is indicated in Figs. 28-40, 28-41, and 28-42. A coil surrounds a central core, and when energized by direct current, a high flux density is created in the poles. The highest feasible flux density is desirable because the pull of a magnet, for a fixed area of poles, is proportional to the square of the flux density,

$$F = B^2A/8\pi$$

where F = force exerted, dynes
B = flux density, gausses
A = area, sq cm

In engineering units,

$$P = B_i^2A_i/(72.1 \times 10^6)$$

where P = pull, lb
B_i = flux density lines per sq in.
A_i = area, sq in.

In Figs. 28-40 and 28-42, the steel casting—of high permeability—furnishes a housing of substantial character, as well as pole and yoke. The housing is provided with wide fins to add strength and cooling surface to the structure. The cover plate is made from manganese steel, which withstands the pounding and abrasive action of loads. It is also nonmagnetic and does not short-circuit the flux path, which would destroy the efficacy of the magnet. The use of lifting magnets includes the handling of miscellaneous scrap, pig iron, borings, and turnings, and almost all ferrous materials, even to recovering steel lading from sunken barges. A common use is the lifting of a skull-cracker ball, 25 tons or so, with the larger magnets. After raising to the proper height, the magnet is deenergized and the ball dropped to break the scrap beneath it. The same magnet then picks up the scrap and loads it on cars. The bipolar magnet is used for handling orderly piles of rails, blooms, billets, structural shapes, pipes, or other steel products in stockyards and is used singly or in pairs. The rectangular magnet shown in Fig. 28-41 is not subjected to such severe service as the steel-clad magnets of Figs. 28-40 and 28-42; hence its coils do not require the heavy steel housing of the ordinary lifting magnet. It is used for handling flat-surfaced materials such as billets, slabs, and blooms and is not subject to weather.

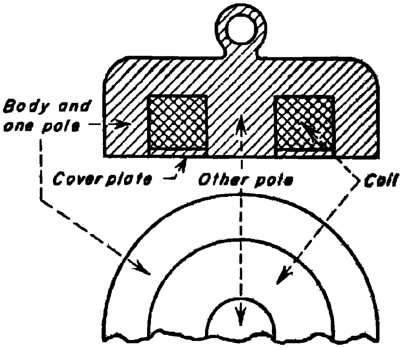

Fig. 28-40. Lifting magnet, annular type.

When a magnet is demagnetized to discharge its burden, the breaking of the highly inductive electric circuit is accompanied by a high-voltage inductive discharge. To prevent this and the consequent damage to contacts and insulation, a discharge resistance of suitable value is connected across the terminals of the coil before the circuit is opened. An instant later, the current is reversed to demagnetize the magnet and clear it entirely of its load. The controller is often equipped, especially on large magnets, with a series resistor which reduces the current in the magnet to a much lower value, sufficient to retain the load after it is once lifted. These controllers may be either hand-operated or of the automatic magnetic-contactor type.

Magnetic chucks are extensively employed when machine work is to be held for a grinding or light tooling operation. They offer a refined method of holding nice work that is magnetic without marring or distorting it. They cannot be used on nonmagnetic material. Two types of magnets are used: permanent magnets and electromagnets. They are of two different forms: circular for lathe and rotary-grinder service and rectangular platen form for plain or surface grinders and milling machines. The development of aluminum-nickel-cobalt magnet steels has introduced the feasibility of making excellent magnetic chucks by employing permanent magnets. This makes it unnecessary to provide direct current, or any current for that matter, and yet have a chuck with a holding force approximating that of electromagnetic chucks. The working of the permanent-magnet chuck will be understood by reference to Fig. 28-43. Strong permanent magnets are arranged to alternate with steel strips of high permeability, as shown in the figure. In Fig. 28-43*a*, the magnetism furnished by the magnets leaves them at the top and passes through the steel top plate, then

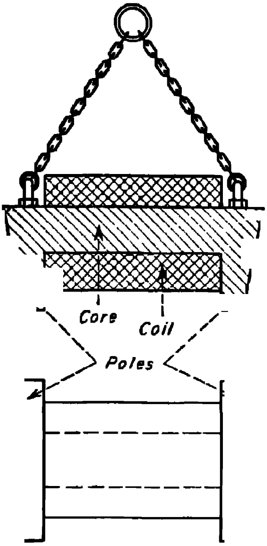

FIG. 28-41. Lifting magnet, rectangular type.

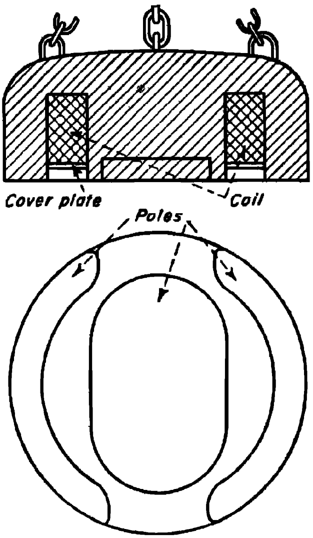

FIG. 28-42. Lifting magnet, bipolar type.

Table 28-2. Some Approximate Data Relating to Magnets

	Nominal diameter, in. (load, lb)						
Diameter	30	36	42	48	54	60	66
Billet or slab	15,000	22,000	30,000	40,000	50,000	60,000	75,000
Pig iron or heavy scrap	630	800	1,200	1,500	2,000	2,500	3,000
Skull-cracker ball	5,000	6,500	7,500	12,000	15,000	18,000	20,000
Light scrap	300	400	600	750	1,000	1,250	1,500
Weight of magnet, lb	1,200	1,800	3,000	4,000	5,700	7,800	9,500
Current, amp, at 220 volts	1.7	4.4	5.6	7.7	8.8	11.0	13.2

through the work, returns down through the top plate, the soft steel strip to the sole plate, and back to the opposite end of the magnet. The flux is confined to the desired path by means of the nonmagnetic separators. The magnetism thus induced in the ferrous work results in a strong attractive force between it and the magnet poles, thus holding it firmly to the surface of the magnetic chuck. To detach the work, the magnetic structure is shifted to the position shown in Fig. 28-43b. When the magnets are in this position, the magnetic flux passes through the top plate without entering the work, and the holding power is withdrawn. The shifting of the magnets is accomplished by a lever which actuates a rack and pinion, or other mechanism, to move the magnets between the ON and OFF positions. An arrangement similar in operation is employed with the circular chucks, except that a key is inserted in a suitable opening and withdrawn when the shift is accomplished. All the chucks are made with precision, and the plane faces of the top and bottom parallel. Setting up the work is greatly simplified, especially where a number of thin strips or washerlike objects are to be ground to a uniform thickness at one time.

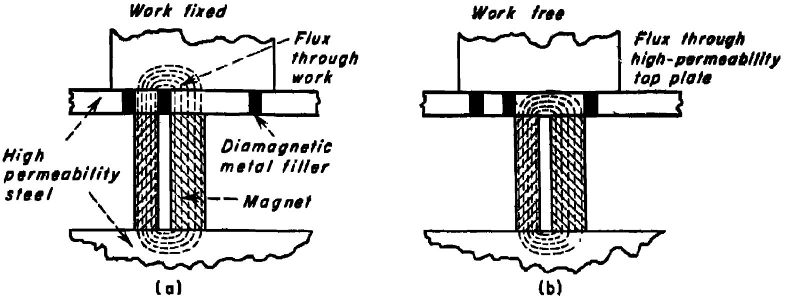

Fig. 28-43. Magnetic chuck. (a) Work fixed. (b) Work free.

The electromagnetic type of chuck has a structure that is represented in Fig. 28-44. All the steel or iron parts are of high permeability so that as much as possible of the magnetomotive force may be made effective on the work. The magnet cores, situated between the top plate and the sole plate, are wound with magnetizing coils, as shown.

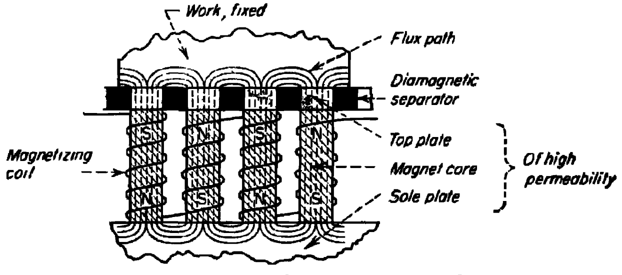

Fig. 28-44. Electromagnetic chuck.

The windings are connected in such an order that the poles at a given end alternate between north and south, and flux is caused to traverse the workpiece somewhat as shown in the figure. The work is thus held firmly. To detach the work, a reversing switch is used to open the circuit and reverse the direction of the current in one quick movement. If properly timed as to length of dwell in the reversed position, the flux reversal kills the remanent magnetism, and the work may be removed from the chuck in a demagnetized condition.

It is not always possible to demagnetize a material of high retentivity entirely when removing it from an electromagnetic chuck. Since no reversal occurs when objects are removed from a permanent-magnet chuck, some magnetism is retained.

This is troublesome in the case of tools, particularly cutting tools, since the swarf will interfere with vision of the working of the tool, as well as cause it to jam, and result in spoilage of work. In the days when only d-c dynamos were used, attendants and others soon learned that the stray flux from the strong bipolar fields destroyed the timekeeping qualities of their watches by magnetizing them. They also soon learned that the same field which did the damage could be employed to undo it. This was accomplished by suspending the watch from a string passed through the bow. As it hung vertically, the string was twisted. The watch was then suspended in the stray field and rotated in the field as the string unwound. This subjected the magnetized parts of the watch to a strong alternating flux; and if the watch was withdrawn while rotating, it would be found to be demagnetized. This same method may be applied to any other small parts, but it also illustrates the principle involved in demagnetization. The magnetized material is subjected to an intense rapidly alternating flux (25 or 60 cps is useful), and the field is decreased in value to zero while it is still acting. This may be simply accomplished by inserting the object in a solenoid energized with an alternating current and effecting a reduction in current until it reaches substantially zero. A convenient device for use in connection with magnetic chucks is a surface consisting of a divided flat plate connected to the two limbs of a laminated silicon-steel core energized with alternating current of commercial frequency. Pieces larger in area than the surface of the plates may be moved about while in contact with them, so that the alternating flux will act on all portions of the magnetized surfaces.

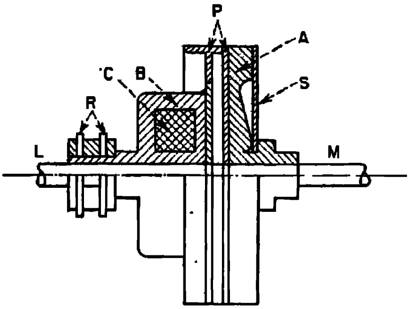

Fig. 28-45. Elementary form of magnetic clutch.

Magnetic clutches represent an important industrial use of magnetism. One type of single-plate, dry-type, magnetic clutch is shown in the elementary sketch, Fig. 28-45. Current is supplied to the magnet coil *C* by means of the collector rings *R* which revolve with the body *B* and collect current from stationary brushes not shown. To engage the clutch, current is supplied to the coil *C*, which attracts the movable armature *A* against the restraining action of the spring *S*. The spring *S* normally holds the armature from contact with the friction-plate assembly, thus allowing the clutch to run free. On energizing the magnet, a substantial pressure is applied to the assembly *P* which consists of a friction plate and pressure plates faced with treated woven-asbestos brake lining. Multiple-disk clutches are also made which greatly increase the effective area of the driving surfaces for a given body diameter and thus maintain a low ratio of moment of inertia to driving torque. The magnetic clutch possesses distinct advantages for coupling many machinery drives. It may be operated by a push button or small remote-control switch at any desired distance from the clutch; there are no levers or other mechanical linkages necessary to engage the clutch; the only wearing parts are the friction plate and rugged facings which are easily renewed; a smooth gradual engagement of the clutch is ensured by the exponential increase in current in the inductive circuit, even though full voltage is applied when the clutch is energized. The current increase depends on the familiar equation governing the increase of current in an inductive circuit,

$$i = E/R(1 - \epsilon^{-(Rt/L)})$$

where i = current at any instant of time t
E = applied voltage
R = resistance
L = inductance of the circuit
ϵ = base of the hyperbolic logarithms

The magnetic clutch can also be used as a torque-limiting device. The driving torque of the clutch depends on the pressure applied to the friction disks, which in turn depends on the current. If a limit is placed on the current, a corresponding limit is placed on the torque. Any torque exceeding this limit causes the clutch to slip. When slip occurs, the circuit may be opened by mechanical action or by thermal action from the heating of the pressure-plate system. Very little energy is necessary for the operation of the clutch; it may vary from something in the neighborhood of 10 watts per hp for small or low-speed clutches to as little as 2 or 3 watts per hp for large or high-speed clutches. An effective arrangement for reversing service consists of an armature on the driven member and clutch bodies revolving in opposite directions for each of the two driving members. Either clutch may be engaged by energizing the proper coil, and electrical interlocking is easily and effectively accomplished.

The magnetic brake in one form (multiple-disk) is not unlike the magnetic clutch described in the foregoing paragraph. The fundamental difference is that the spring is normally acting to set the brake; *i.e.*, the brake is normally on. The coil of this brake, when energized with direct current, is usually in series with the armature of the motor and is released whenever the armature is energized. (It cannot be used in this manner for a-c service because of the necessarily high inductance that would be inserted in series with the armature.) This brake finds its principal use on the motor shafts of electric cranes and hoists operated by direct current. The motor has an extended shaft on both ends to accommodate the driven machine and the brake stops the motor within a few revolutions when the current is cut off. The brakes are made in the dry type, with treated-asbestos friction disks rubbing on steel plates; and in the oil-immersed type, with fiber friction disks rubbing on brass plates. This type of brake is particularly adapted to severe service conditions because all wearing surfaces and moving parts are entirely enclosed. Arrangement of the operating coil in series with the armature is particularly desirable in hoisting service, since this requires that the armature be energized to hold the load before the brake is released. To obtain superior braking and magnetic-contactor service, the use of direct current is sometimes dictated. Steel-mill machinery, hoists, cranes, lift bridges, turntables, conveyors, and other applications where it is necessary to stop a load quickly and hold it belong in the foregoing class.

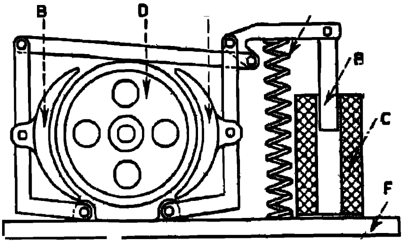

FIG. 28-46. Drum-type magnetically operated brake.

The drum type of brake with shoes is shown in Fig. 28-46. This type of brake is of very general application and suitable for elevators, cranes, bridges, door and window operators, printing presses, rotating machines, etc. It is solenoid-operated and designed to stop a load quickly and hold it but is not intended for lowering loads. The drum *D* is mounted on the shaft to be stopped, usually on one end of the doubly extended shaft of the driving motor. The brake shoe *B* should be so constructed as to eliminate chatter and so placed as to eliminate drag. It is usually spring-set by a spring *S*, electrically released by a solenoid *C*, and normally in the applied position. The system of levers is operated by the plunger *P*. In small sizes, gravity sometimes takes the place of a spring for setting the brake. It has the advantage of a constant ever-present force but the disadvantage of requiring more cumbersome devices for adjusting the torque. In small to moderate sizes, say from a few pound-feet to a hundred or so, this brake is solenoid-operated from full-line voltage for either d-c or a-c service (25 hp at 1,150 rpm = 114 lb-ft). Larger sizes, up to many thousands of pound-feet of torque, especially for alternating current, are thrustor-operated. The thrustor is a hydraulic cylinder and piston, the pressure being applied by a small vertical motor and centrifugal impeller.

Band brakes are operated by a mechanism similar to that described in the foregoing paragraph and are satisfactory and desirable for many types of service if properly constructed.

For quick starting and stopping, the brake drum should be of small diameter and low inertia. The brake shoes should lift clear of the drum when the brake is released to prevent dragging. The adjustments should be few and should embody adjustment of spring pressure for braking torque, shoe clearance to compensate for wear of the lining, and an adjustment to equalize clearance of the shoes. The brake rating and setting should usually be equal to or greater (say 2.5 per cent greater) than the full-load torque of the motor. If not on the name plate, it may be computed by torque in pound-feet = 5,250 × hp (rated)/rpm (full load).

The eddy-current loading brake shown in Fig. 28-47 is a very useful magnetic device for conducting output-load tests on small machines (5 hp or less), particularly electric motors. The load may be varied over any range within the capacity of the brake in any increments desired by variation of the current in the loading coils. The aluminum disk *D* is keyed to the shaft and the rest of the brake carried on bearings *E* mounted on the shaft and supporting the torque arms *A*. The disk cuts the adjustable flux from the magnetic yokes *F*, which are energized by the loading coils *C*. The rate of cutting the flux determines the magnitude of the eddy currents formed in the disk and hence the torque released by the shaft. Power is dissipated by natural convection to the air surrounding the rotating disk. Torque in pound-feet is determined by the product of the lever arm *L* in feet and the balance reading in pounds. The rating of this brake is practically limited by the rate at which energy can be dissipated naturally from the rotating aluminum disk. Since there is no friction involved in fixing the input to the device, it cannot grab or chatter, and the operation is steady and smooth.

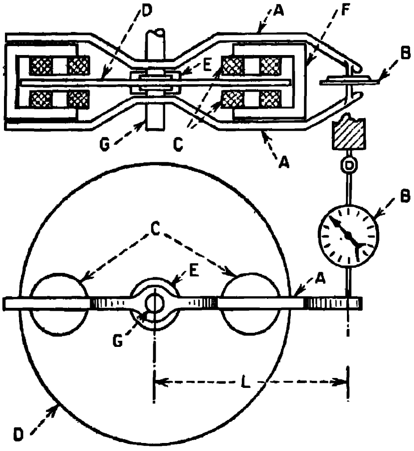

FIG. 28-47. Eddy-current loading brake.

Magnetically operated valves serve a very useful purpose and are extensively used for remote or electric control of fluid flow. They are usually of the full-on to full-off type and are seldom used for direct regulation. One example of the use in regulation would be the application to a gas-fired furnace. In this case a hand-operated throttling valve would be used to supply sufficient fuel to keep the furnace somewhat below the required operating temperature. An electrically operated auxiliary valve would supply additional fuel sufficient to exceed the operating temperature. This valve would be controlled by a temperature-actuated device and operated in an off-on manner to maintain a constant temperature in the furnace. Many manufacturers furnish valves of this character, reliable and simple in operation but with various actuating mechanisms. For large valves, or for continuous variation in throttling, motor- or thrustor-actuated valves would be used.

Magnets and solenoids are a very necessary complement to industrial plant equipment and are often a requirement for special devices. They constitute a very reliable and important device if properly constructed and applied. Their design should be developed from treatises on this subject. It should be generally understood that direct current is most desirable for the operation of tractive magnets, plunger magnets, or solenoids. This objective can often be met, when only an a-c supply is available, by means of a boundary-type rectifier such as copper oxide or selenium. A magnet should not be relied on to hold any device in a safe position. Since current supply may fail, the normal position or a latched position should be the safe one, and the magnet should be used for tripping or release.

Permanent magnets have reached such a stage in development and manufacture that the designer of modern industrial instruments and control devices would be seriously handicapped if deprived of their use. Even small motors and generators can often be constructed with a cheapness and simplicity not possible with electromagnets. Previous to 1934, the designer was dependent on carbon, tungsten, chromium, or cobalt magnet steels. The introduction of Alnico steels, composed of aluminum (Al), nickel (Ni), cobalt (Co), and iron, has given to industry a new series of permanent-magnet materials vastly superior in permanent magnetic properties to any previously existing. One in the series, Alnico 5, has an energy value twenty-five times that of 1 per cent carbon steel. At present seven steels in the Alnico series, as well as other magnetic materials, some without any iron, are available for making high-energy permanent magnets. Most of these modern magnetic materials require special casting or sintering and finishing methods. It is often better to buy the fabricated, and in many cases magnetized and finished, magnets. So many permanent magnets in various sizes and shapes have been developed that it is often possible and advantageous to adapt a stock magnet to the designer's use from the large numbers the manufacturers have available.

Any permanent magnet is subject to demagnetization from temperature changes, vibration, impact, stray magnetic fields, or contact with other magnetized materials. Modern magnets, stabilized against such influences, can be furnished by the manufacturers. The stabilized magnets are first carried to a residual induction higher than the flux density needed in the intended service and subjected to cycles of the particular demagnetizing force somewhat greater than the service will demand. This ensures a high degree of stability when the magnet is used.

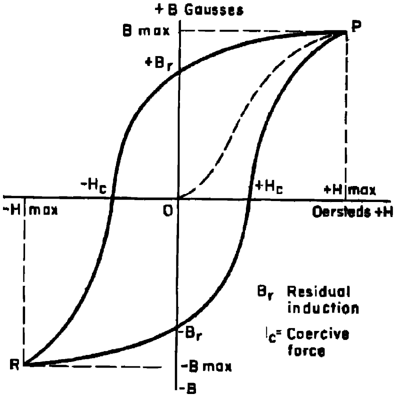

Fig. 28-48. Hysteresis curve for a permanent magnet.

Permanent magnetic properties may be discussed by the aid of Figs. 28-48 and 28-49. If any demagnetized magnetic material be magnetized by applying to it successively increased magnetizing forces H in oersteds, the flux density B in gausses will increase along such a curve as the dotted line OP, Fig. 28-48. If now the magnetizing force is decreased, the flux density will decrease, but not so fast as it previously increased, along the line PB_r. It will be observed that at 0 oersteds there still remains a flux density of B_r gausses. This is called residual induction. To reduce the flux density to zero, the magnetizing force must be reversed. As the magnetizing force is increased in a negative direction, the flux density will decrease along a line as shown, from B_r at $H = 0$ to 0 at $H = -H_c$. The value of negative magnetizing force required to reduce the flux density to zero is in magnitude H_c and is called the coercive force. Further increase of $-H$ will produce a flux density in a negative direction at point R equal in magnitude to the positive value at point P. The cycle of magnetizing forces and corresponding flux densities may be carried through the other remaining values to the point P and thus complete the hysteresis loop.

The important points to note are B_r, the residual induction, and $-H_c$, the coercive force. The corresponding values of B_r and $-H_c$ measure the external energy of the magnet, *i.e.*, represent the variation in potential energy available for external use. The section of the curve between B_r and $-H_c$ is replotted to an enlarged scale in Fig. 28-49. The products of flux density B_d at any point and the corresponding value of the demagnetizing force $-H_d$ are plotted in the curve at the right of the axis OB. This is the external energy curve. The curve signifies the flux density at which a magnet will give the maximum

energy output. To obtain a minimum volume in a magnet, the design should be such that it will operate at the flux density that makes use of the maximum external energy.

Magnetic separation for protection of equipment and foods, as well as segregation and recovery of magnetic materials, is an important application of magnetism. In many industries it is necessary to protect from damage by tramp iron (manganese steel is nonmagnetic and an exception) and foods in process from parts of machinery and tools. In mineral and mining industries, considerable segregation and concentration of unlike materials may be accomplished by taking advantage of their differences in permeability. In iron foundries, materials such as chaplets, sprues, shot iron, etc., may be recovered from molding sand. In the sugar industry, for example, the shredder or extraction rolls would be badly damaged if steel from the cane-harvesting equipment were carried along with the cane to the shredder or crushing rolls. It is also desirable to remove particles of scale or rust introduced into the syrup from the inside surfaces with which the liquid comes in contact. This material is not strongly magnetic and requires a field of high intensity for its removal. The intensity of field required for a particular application will vary with the permeability of the material that is to be separated and the depth of the load on the belt or carrier. It may often be necessary to spread the load very thinly as it passes over the magnetic separator.

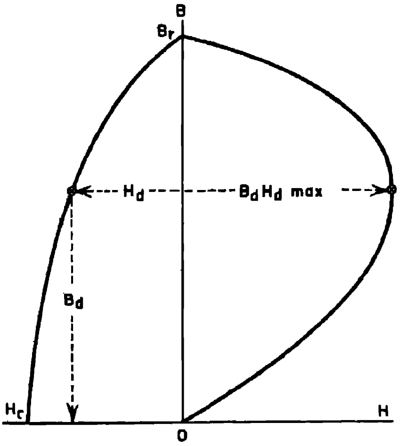

FIG. 28-49. External energy curve.

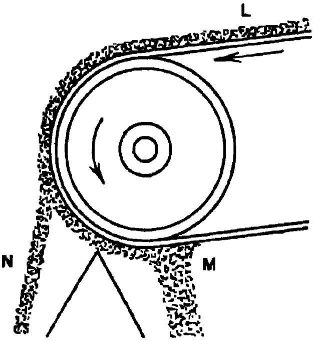

FIG. 28-50. Magnetic pulley, end view.

A device commonly used in magnetic separation is the magnetic pulley shown in end elevation in Fig. 28-50 and in longitudinal cross section in Fig. 28-51. The pulley shown in Figs. 28-50 and 28-51 is magnetic over its entire surface. The magnetic material in the conveyor load approaching the pulley at *L* is held against the pulley through the one-half circumference or so until it is removed from the magnetic influence in the neighborhood of *M*. The nonmagnetic material leaves the pulley under the action of gravity when it reaches a point where the surface of the pulley is vertical. The pulley consists of a number of magnetic poles as indicated in Fig. 28-51. The poles are created by the exciting coils *E* surrounding the core *A*, causing the flux to flow through the disks which separate the coils and form the poles. A strong field is created external to the nonmagnetic shell *D* and takes the general path indicated in *F*. Direct current for excitation of the coils is supplied through the collector rings *C*. Where only alternating current is available, direct current may be conveniently obtained by means of electronic rectifiers. It will be seen that for continuous service this pulley presents a problem in dissipating the heat generated in the coils. It is a more or less solid mass of material, often a few feet in diameter and several feet long. The cooling can be accomplished only by convection from the outer surfaces, and even this is not effective because of the usual low speed. Elimination of the cooling problem may be accomplished by making use of the newer high-energy permanent magnetic materials. At the same time the necessity for moving contact and d-c supply is removed.

The magnetic separator drum shown in Fig. 28-52 is sometimes used. The drum *D* is of nonmagnetic material and surrounds the stationary magnet *M*, which has its poles created in a manner similar to the magnetic pulley. The drum revolves constantly around the magnet, the unsorted material is delivered at *L*, the nonmagnetic material leaves at *N*, and the magnetic material is carried around on the drum to *M* where the magnetic field becomes too weak to hold the material, and it is discharged.

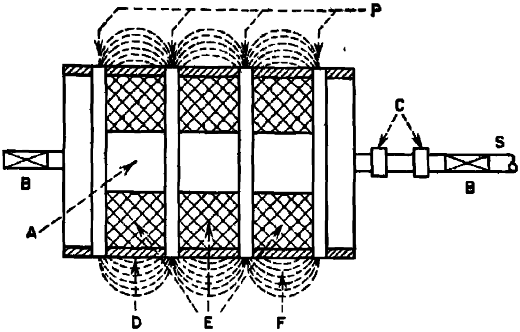

FIG. 28-51. Magnetic pulley, longitudinal section.

Another very effective magnetic separator, used for the rudimentary forms of separation, such as removing tramp iron from coal, is made by suspending a magnet similar to a lifting magnet over the conveyor belt. Since the highest flux density is at the center of this magnet, the total diameter of the magnet should be great enough for this high flux density to cover the belt width. This form of separator for iron pieces of troublesome shape entrained with large lumps gives good performance. It is also possible with this device to operate the conveyor belt at a much higher speed, 500 fpm or more, and still maintain effective separation.

A more refined type of separator is constructed by running a horizontal conveyor belt between the poles of very strong magnets with their fields vertical through the belt. The poles below the conveyor belt are broad and flat, and those above the belt are brought to a narrow edge in order that a very high-density field may be obtained. Small transverse belts move over the main belt and its load but under the concentrated pole where the high flux density exists. Magnetic material rises from the conveyor belt through the high flux density toward the narrow poles but is intercepted by the moving transverse belt and carried off to suitable chutes or collecting bins. This machine may be provided with as many pairs of poles as desired so that a mixture of materials of many permeabilities, varying from strongly magnetic to feebly magnetic, may be separated. A series of materials of decreasing susceptibility, such as magnetite, franklinite, ilmenite, cerium, manganese, pyrrhotite, palladium, strontium, biotite, and chromium, could be separated in the order given. The first-named mineral has the highest susceptibility and is removed first under a comparatively weak field. The others are removed in order as the conveyor belt carries them to stronger and stronger fields under successive pairs of poles.

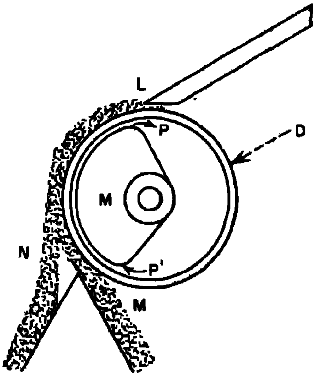

FIG. 28-52. Magnetic separator drum.

Nondestructive testing by magnetic methods has become of great importance in the manufacture and use of steel products within the past two decades. It is based on the very high reluctance of gas or other occluded nonmagnetic bodies compared with the low reluctance of the steel body in which it appears as a flaw. Of course, surface cracks, whether resulting from improper methods of heat-treating or grinding, or from other cause, fall into the same physical category. The interruption in the magnetic continuity is most evident when the flaw is at the surface, but in any case it will distort otherwise parallel flux lines as represented in Fig. 28-53. It is desirable that the flux be perpendicular to the long dimension of the defect, and this is accomplished by properly orienting the magnetizing force with respect to the defect. For example, in a steel bar or shaft where the defects are suspected of being transverse, the bar may be magnetized longitudinally by means of a solenoid that encircles it. However, defects may be longitudinal, resulting from long seams, splits, slag inclusions, or other defects rolled into the piece. For this work a heavy current through the bar from end to end will magnetize it circumferentially. On the other hand, a hollow shaft that would be likely to have longitudinal defects could be conveniently magnetized by a heavy current in a conductor threaded through the shaft. This would create a flux around the periphery of the shaft normal to the general direction of the defects. It is desirable, but not essential, that the flux be normal to the long axis of the defect; for no matter what the respective general directions of flux and flaws are, there is usually some directional component that will reveal the defect.

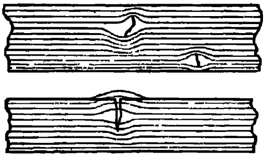

FIG. 28-53. The distortion of magnetic flux by flaws.

Methods of detecting this deviation of flux are numerous. A modified form of fluxmeter, with an exploring coil which may be moved about over the surface of the piece undergoing test, is very effective on some types of work. The presence of a fault may be indicated by a scale on an instrument scale or dial or by audible means. The audible detection is particularly useful in the form of a headphone when an a-c excitation is used in connection with an amplifier. In general, a higher flux density will be produced by direct current and more obscure faults revealed; hence its use is more general. When an alternating current is used, the necessity of subsequent demagnetization is eliminated. Detection of faults by observation of the orientation of finely divided magnetic particles was the first method employed and has been highly developed. The article under test is often immersed in an oil suspension of the magnetic particles and their orientation readily observed. The iron filings first employed were found to be most faulty as detectors. They often contained carbon and retained permanent magnetism, were not finely enough divided for exact work, and settled too rapidly for use in an oil suspension. Magnetic oxides are now available in gray, red, and black, so finely divided as to orient over minute flaws, remain in suspension in the lightest oil, as well as offer a high degree of visibility on account of the possible contrast of color that may be selected for a particular surface. The magnetic powder may be dusted on a surface from a shaker or applied in an air stream from a suitable blower. This method is often used in preference to oil immersion or when oil immersion is impracticable. A convenient detector is made by enclosing a thin layer of oil-suspended particles in a very thin, circular, highly polished, nonmagnetic tray or dish with a clear-glass or plastic cover. When this detector is moved over a magnetized surface, the particles may be seen to orient themselves about the distorted magnetic lines at a fault. This method, as well as the fluxmeter method, avoids the necessity of cleaning the article following examination.

The magnetization and demagnetization may be accomplished in numberless ways, and many special equipments have been developed for this express use. Direct current is most frequently used but is not so easy to obtain in high values as alternating current because of the ease of transformation of the latter. Direct current may be obtained from a large storage battery with cells connected for a high rate of discharge, from special high-current low-voltage generators, or from the discharge of large capacitors. Capacitors are also useful in demagnetization if connected in parallel with a magnetizing coil. When the magnetizing current is removed by opening the switch, the decaying oscillatory discharge which results will demagnetize any material threaded by the coil flux. Where the size and shape of work will permit, it may be introduced into an a-c solenoid and the current slowly decreased from its maximum value to zero. The demagnetizing coil is improved by having a laminated outer casing to limit the current inrush when its load of material to be demagnetized is small.

A fluorescent method of detecting surface cracks or flaws in nonmagnetic material, including plastics and ceramics, possesses considerable merit and will be noted here, although it is not magnetic in character. In its ultimate concepts and development it is a relatively new method of nondestructive testing. It may have been suggested by the old method of permeating surface cracks by wiping the surface with whiting and kerosene. The whiting was often carried into cracks by the kerosene and thus revealed otherwise invisible flaws. The newer method is a distinct refinement of this somewhat crude procedure. A very finely divided powder is combined with a penetrating oil to go into flaws too small to be found by the unaided eye. When the surface is illuminated by "black" or ultraviolet light, the fluorescent glow emanating from the permeated flaws is plainly visible. The object is preferably viewed in a subdued light under ultraviolet excitation of 3200 to 4000 Å. Total darkness is unnecessary, as the fluorescence emits a blue-green glow from the flaws and is readily visible except in bright light. In addition to its use on nonmagnetic materials, this method has been used to advantage on highly polished magnetic materials where the highlights interfered with high visual acuity.

ELECTRONICS

Electronics in industry has been increasing in importance since about 1932, but its applications were greatly accelerated under the influence of the demands of the Second World War. There are innumerable ways in which electronics can serve the industrial plant, and new ways are being found and used daily, but they may be grouped in five categories:

1. Tool in production processes
2. Testing and inspection
3. Instrumentation
4. Measurement
5. Protection

Electric theory and electronic principles will not be developed here because numerous texts are available dealing with these extensive subjects. Neither will a catalogue of the innumerable circuits and applications be introduced here, rather, some examples suggesting typical applications will be cited. The electron tube, or photoelectric cell, furnishes the brain of an application; and the electric circuit that it controls, the body of the creature. The electrons within the electronic tube or photocell have so little inertia that the time involved in their movement, or energy for their control, is negligible. The electron tube may be looked upon as a switch that can be completely closed in a microsecond or so and opened in a millisecond. The lever that operates this switch may be a few microamperes or less from a photoelectric cell or other detecting device. These very small amounts of energy, when properly directed and amplified, can easily control with exactitude thousands of kilowatts. In a true sense, an electronic function involves the control of free electrons. Among common examples is the action in a thermionic tube, photoelectric cell, X-ray tube, cathode-ray tube, electron microscope, gas and vapor rectifiers, and the common fluorescent lamp. High-frequency induction furnaces, dielectric heating equipment, and many other similar devices are also classed as electronic, since, though not strictly electronic, they are closely associated with the electronic domain.

Induction heating was recognized as practicable by many scientists and engineers as early as 1890, but it was 25 years later before extensive use was made of the process. The melting of metals for the manufacture of ingots of brass, bronze, nickel-silver, jeweler's metal, etc., in this country is nearly all done by the low-frequency or core-type induction furnace. This furnace is essentially a core-type transformer where the primary coil is connected to a source of commercial frequency, and the secondary consists of a ring or single turn formed of the metal that is to be melted. In the early furnaces this ring was horizontal, but this form of furnace—largely because of the troublesome pinch effect—was abandoned in favor of the vertical-ring type represented by the Ajax-Wyatt furnace. A diagram of this furnace without the structural details, supporting frame, etc., is given in Fig. 28-54. The ring of metal in the V-shaped channel *E* is completed by the reservoir *A*. The laminated core *D* is magnetized by the primary winding *C*. The body of the furnace *F* is constructed of refractory insulating and supporting materials. The molten metal is rapidly stirred by the electrodynamic forces in the V-shaped channel *E*, and this rapid movement carries on into the reservoir *A*, as indicated by the dotted line near *B*. The most common size of furnace has a capacity of some 900 to 1,000 lb of metal, and from 600 to 750 lb is tapped at each heat, the balance being retained in the furnace to preserve the closed ring in the channel. When started, the furnace is filled with molten metal and must be entirely emptied or kept running at reduced energy input when production is interrupted. Heats can be run off quickly, the furnace has a low energy consumption under continuous production, and the power factor approximates 75 per cent.

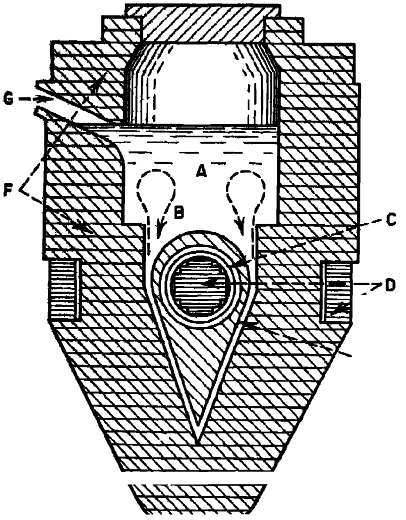

Fig. 28-54. Vertical-ring low-frequency induction furnace.

The high-frequency induction furnace has been developed since the low-frequency one proved so successful. The charge in this furnace is also the secondary of a transformer which operates without a core, and this furnace is frequently called the coreless induction furnace. In Fig. 28-55, a charge of conducting material *A* is contained in a refractory lining or crucible *B*. The charge is cut by a high-frequency electromagnetic flux supplied by the single-layer exciting coil *C*. This coil is usually made from round copper tubing that has been partially flattened to the approximate cross section of an ellipse, through which water flows to carry away the heat and reduce its resistance losses. It is commonly made with the height about equal to the diameter, but it may vary more or less from this proportion. The heating of the charge is by induction from the action of eddy currents, and hysteresis in the case of magnetic materials when below the recalescence point in temperature. (Nonconducting materials are often melted by placing them in a conducting crucible, such as graphite.)

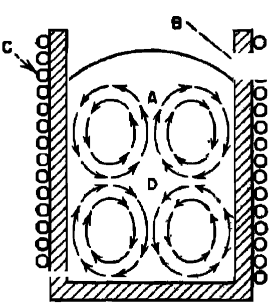

Fig. 28-55. Coreless or high-frequency induction furnace, elementary arrangement.

The depth of penetration is governed by the resistivity of the material and frequency of the flux. It is expressed by the formula $\delta = 5{,}030\sqrt{\rho/f}$, where δ is the equivalent depth of penetration in centimeters when uniform current over the shell of δ-cm depth is assumed; ρ, the resistivity of the material in ohms per centimeter cube; and f, the frequency in cycles per second. The effective resistance R_e equals $\delta 2\pi(a - \delta/2)/\delta l$ (ohms), where a is the average outside radius of the charge and l its axial length. The effective reactance X_e equals $2\pi a\rho N^2/\delta l + (8\pi^3 f N^2/l)(b^2 - a^2)10^{-9}$ (ohms), where b is the average radius of the primary coil and N is the number of turns in the primary. From these formulas the electrical characteristics of a coreless induction melting furnace may readily be computed.

The equivalent circuit of the coreless induction furnace is indicated in Fig. 28-56. As is usual, the primary and secondary parameters may be combined to form equivalent resistance, reactance, and impedance referred to the primary side. For commercial furnaces with capacities from a few hundred pounds to a few tons, power at frequencies of 960 or 480 cps is used and is obtained from specially built synchronous generators. In order that a good power output of the generator may be obtained without excessive heating, power-factor-correcting capacitors are applied to the furnace circuit as indicated in Fig. 28-57, where *A-A* represents the primary coil of the furnace with taps for adjusting the voltage. It may thus be used as an autotransformer for adjusting the voltage on the capacitors, thereby adding flexibility to the arrangement.

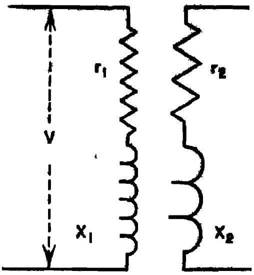

Fig. 28-56. Equivalent circuit of coreless induction furnace.

Smaller high-frequency induction furnaces for laboratory use or for commercial melting of a few ounces or pounds of precious or high-melting-point metals use frequencies of 10,000 or 20,000 or more cps. The generator for this use consists of a steel-core transformer with a relatively high leakage reactance. The primary winding *P-P* (Fig. 28-58) is connected to a supply line of commercial frequency. A high-voltage secondary *S-S* is connected to the inductive coil of the furnace *F-F* through the capacitors *C′* and *C″*. This will be recognized as an oscillatory circuit. The hydrogen-quenched mercury spark gap *G* is connected across the high-voltage terminals of the transformer. When the voltage across the transformer reaches a critical value, a breakdown and spark discharge occurs, and the ensuing oscillation furnishes the necessary frequency of several thousand cycles per second. Each alternation of the primary voltage initiates a new oscillatory discharge, thus maintaining the high-frequency supply. The same principles discussed in connection with the synchronous-generator-operated furnace apply to the oscillator-operated furnaces, except that the frequency and wave form are neither fixed nor constant.

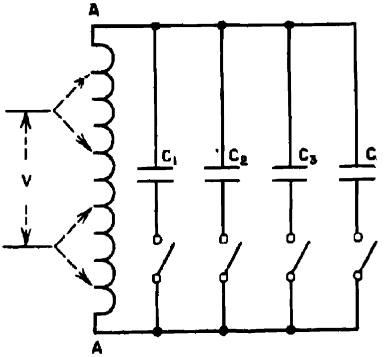

Fig. 28-57. Circuit arrangement for power-factor correction of coreless induction furnace.

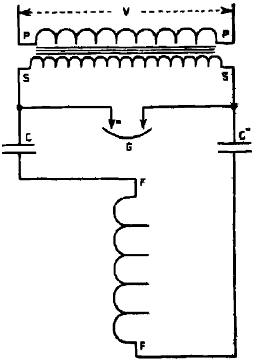

Fig. 28-58. Circuit arrangement for small high-frequency induction furnace.

Radio-frequency induction heating started about 1925. The automobile industry needed parts with a hard surface to resist wear and a tough core with better mechanical properties than could be had by casehardening mild steel. This was accomplished by subjecting the parts to an alternating flux of so high a frequency that there was very little depth to its penetration and then quenching the part to produce the hard surface. When cold ferrous metals are subjected to radio frequency, the permeability

—which is of a high value since the metal is far below the point of recalescence—must be considered. This requires that permeability be introduced into the denominator of the radical in the penetration formula. It will be evident, then, that the depth of heating may be kept as close to the surface as desired or increased by proper control of frequency and time. A short-time heating does not allow heat penetration by heat conduction.

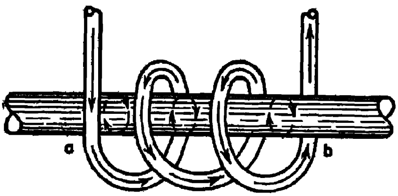

FIG. 28-59. Heating the middle of a bar by high-frequency induction.

If a gear, for example, is to have a tough strong center which will resist shock, yet have the surface of the teeth hard enough to resist wear, the result may be achieved by quickly heating the object in an induction heating coil and quenching it instantly *in situ*. This is readily accomplished by doing the heating in a basin that may be instantly flooded with quenching fluid and as quickly emptied. The basin is ready for the next workpiece as soon as it is emptied, which results in very rapid heat-treating with the rapidity limited mainly by the very short heating period. Very rapid brazing of parts may also be accomplished by high-frequency induction heating. If a flat surface is to be heated, this is effectively accomplished by means of a pancake coil adjacent to the surface. Three representative conditions are indicated in Figs. 28-59, 28-60, and 28-61. In Fig. 28-59, a bar is introduced into a coil of a few turns and heated only in the region *ab*, which is cut by the flux of the coil. The remainder of the bar will receive heat only by conduction from the heated region, even though this region is at a bright orange heat. The bar may be heated in a few seconds, and the short time allowed for conduction of heat along the bar is one of the strong advantages of this method of heating. In Fig. 28-60, the end of a chisel is introduced into a coil, and only the end within the coil for heating to a hardening temperature will be acted on by the coil flux. A flat block is adjacent to a pancake coil in Fig. 28-61, and only the face of the block nearest the coil will have any great amount of heat developed in it. Such examples, given as illustrations of the method of selective heating, could be repeated indefinitely. The configurations of the fields and eddy-current paths are so obscure and complicated that it is difficult to predict by computation just what heating results will be achieved with a given coil, frequency, and object. The variation of frequency and construction of a coil are so simple, however, that a trial can be made in a fraction of the time that would be used in an attempted calculation. Once the frequency and coil construction are established, however, the heating process can be repeated with exactitude.

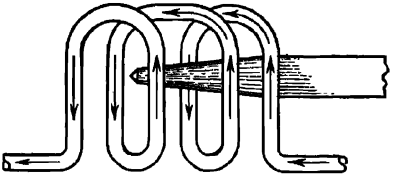

FIG. 28-60. Heating the end of a bar by high-frequency induction.

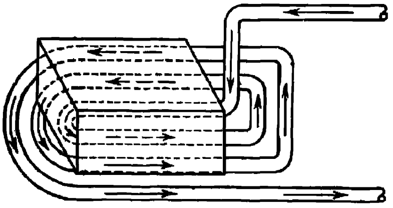

FIG. 28-61. Heating the flat surface of a block by high-frequency induction.

Dielectric heating is sometimes called "electrostatic" heating. This is an error. Static electricity is stationary and, of course, can produce no heating effect. A capacitor, such as is used for power-factor improvement, consists essentially of two conducting plates separated by a dielectric. There is a small amount of heating in the dielectric of the condenser when the plates are energized by an alternating current of low frequency.

The heating is considered to be due to dielectric hysteresis, which has a characteristic similar to magnetic hysteresis, and also to the Joule effect of the current flowing through it. The quotient of this loss expressed in watts and the volt-amperes delivered to the terminals is the power factor of the capacitor. As the frequency of the applied voltage is increased, the impedance of the capacitor decreases in the inverse sense, or with a fixed voltage, the current increases directly with the frequency. The heating is also dependent on the dielectric constant of the material.

If a dielectric fills the space between two flat plates and the edge effect is neglected, the frequency used will depend upon the characteristics of the material, *i.e.*, the power factor, dielectric constant, and the loss factor at the frequency at which the material is to be heated. The power absorbed is approximately

$$P = 1.41 \times 10^{-12} f V^2 \lambda K A / d \qquad \text{watts}$$

where f = frequency, cps
V = rms voltage
λ = power factor
K = dielectric constant
A = area, sq in.
d = thickness of the work, in.

The heating energy in watthours is

$$W = wc\Delta T 10^3/0.95$$

where w = weight of dielectric heated, lb
c = specific heat
ΔT = temperature rise, deg F

By making use of the foregoing formulas, the heating of a dielectric may be closely approximated. However, the following points should be noted: (1) the power factor of the material at the operating temperature and frequency should be known; (2) the dielectric constant does not change greatly with temperature, and if the value at room temperature is used, the results will be as precise as the remainder of the computation warrants; (3) no allowance is made for edge effect, and it is also assumed that the thickness of the dielectric is small compared with its width; (4) for most materials, voltages should not exceed 3 kv per in.; (5) a total voltage in excess of 20 or 25 kv will result in troublesome corona and arcing. At present the available equipment limits the frequency to about 200 megacycles for power output up to 100 watts; to 30 megacycles for power to 40 kw; and to 4 megacycles for greater power outputs. To avoid nonuniform heating from standing waves, the maximum dimensions of an electrode should not exceed one-eighth the wave length used. The wave length in meters is $l = 3 \times 10^8/f$ in a vacuum and substantially the same in air. Air gaps between the electrodes and the material introduce a series capacitance and will introduce large errors in the estimates. Solid dielectrics usually have dielectric constants between 2 and 8.

Rectification of alternating current is by far the most extensive use of electronics when total energy controlled is the criterion. Transformers to change the voltage and tube or steel-tank rectifiers to convert the current to direct provide flexible and efficient apparatus with no moving parts with relatively low investment costs. More than 10 per cent of the total energy used in the United States at the present time flows through rectifiers before use. The magnitude of current conversion extends from a few microamperes to supply a positive potential to the plate of an amplifier tube to many thousands of amperes—furnished by mercury-pool steel-tank rectifiers—to equipment for the electrolytic reduction of aluminum or magnesium. The mechanism of rectification is the same in either case. It results from the physical circumstance that electrons or negative ions will flow freely from a cathode which is maintained negative to an anode which is maintained positive and will not flow in the opposite direction so long as the inverse voltage is kept below a certain critical value, the inverse-voltage rating of the device in use. In the tube, conduction is established by electrons emitted from the hot filament (thermionic emission); in the mercury-arc rectifier, conduction is established by the release of negative ions from the mercury surface to the arc stream by a very high voltage gradient at the surface of the mercury. The electronic or ionic conduction is maintained on the same basis: movement of the negatively charged bodies in a particular direction, *i.e.*, positive, in an electric field. The rectifier thus supplied is necessary for a variety of d-c applications where process control, electrical, mechanical, or optical measurements, or alarms and safeguards are required. The most extensive use is in the millions of kilowatts of installed capacity for aluminum and magnesium reduction. Much additional use is for d-c railways; d-c power for adjustable- and variable-speed machine drives; auxiliary drives for hoists, cranes, ore bridges, coke ovens, and blast furnaces; as well as extensive use on main drives in steel mills. The mining industry makes use of this equipment to supply direct current to mine haulage systems, coal-cutting and ore-loading equipment. Battery-charging equipment for any battery can be secured.

The adjustable-speed electronic drive for d-c motors provides, in one device energized from an a-c supply line, starting and accelerating with control of speed and torque. Figure 28-62 is a schematic diagram of the arrangement. The electron tubes convert alternating current from the transformer into direct current which is supplied and controlled independently for armature and field. The instantaneous response of the tubes provides constant and precise control of speed and torque by control of both armature and field current. Compensation for armature *IR* drop aids in maintaining constant speed independent of load, and armature control in conjunction with field control provides for a very wide speed adjustment. Constant-current acceleration is also provided, and the current and torque peaks that would result from step-by-step acceleration are eliminated.

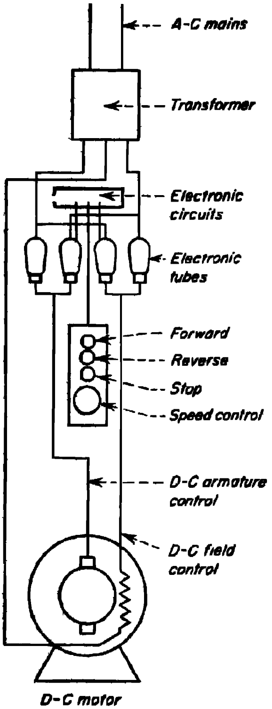

FIG. 28-62. General arrangement of electronic operation of adjustable-speed drive.

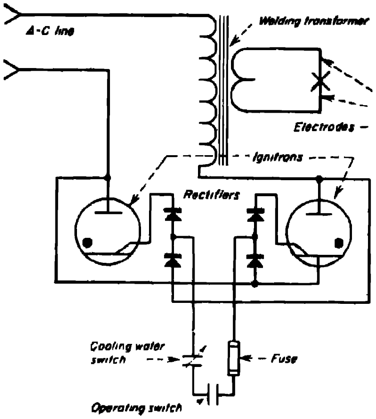

FIG. 28-63. Electronic control of resistance welding.

Resistance welding control is one of the most important contributions of electronic control to industrial applications. Electric resistance welding is the only method of metal fabrication that adds no weight or additional parts to the structure, and it provides consistent quality in the junctions with fewer handling operations It is adaptable to sheets from a few millionths of an inch to ½ in. or more in thickness. In the control of this work, ignitrons controlled by thyratrons provide instantaneous switching with practically perfect timing or spacing of welds without noise or vibration. Control of the magnitude of the current provides adjustment of heat without tap changing. Synchronous switching by thyratron control allows adjustment of time and current control for each weld, with increased speed of welding, improved quality of weld, and greatly increased life of the welding machine. The very narrow range of temperature for the plastic state of most nonferrous metals made the development of the energy-storage system of welding desirable. In this system, a-c energy is converted to d-c energy and intermittently stored in capacitors or inductors, then quickly discharged through the welding transformer. The slow storage and quick discharge of the energy, together with the distribution of the load over three phases, greatly reduce the kva demand on welder supply systems. Electronic control is highly flexible and readily adapted to local needs and plant conditions. Figure 28-63 is a typical circuit arrangement for a resistance welder.

Electronic timing, aside from the timing of resistance welding previously noted, is very widely used in industry; it is accurate, reliable, and readily adjustable. If a potential E is applied to a capacitance C and resistance R connected in series, the value of the current i at a time t sec after the voltage is applied is fixed by the following relation:

$$i = (E/R)\epsilon^{-t/RC}$$

where $\epsilon = 2.718+$ is the base of the natural logarithms. The time constant is RC in seconds with E in volts, i in amperes, C in farads, and R in ohms. If the resistor and capacitor are connected in parallel, the relation is

$$i = -(E/R)\epsilon^{-t/RC}$$

An inductor of L henries could also be used, and if connected in series with a resistor the relation of the quantities is

$$i = (E/R)(1 - \epsilon^{-Rt/L})$$

If the inductance and resistance are connected in parallel, the relation is

$$i = -(E/R)\epsilon^{-Rt/L}$$

The time constant for the inductive cases is L/R. The foregoing relations form the basis of electronic timing. It will be observed that with a fixed voltage and inductance, or capacitance, the resistor can be varied in value to cause the current to reach any selected value in a predetermined time. By proper choice of the circuit and its parameters, the time taken for the voltage across capacitor, resistor, or inductor to reach a desired value is subject to definite adjustment.

The timing device described may be connected so that the voltage across any element of it may be applied to the grid of an electronic tube, and the voltage so selected that the tube is rendered conducting or nonconducting at a predetermined time. This combination represents a timing relay and may be combined with more powerful devices to meet almost any industrial timing need, whether it be milliseconds, seconds, or even several minutes. It is used for timing plastic molding—the various divisions of press-operating cycles in injection moulding, for example—and is generally used for any machine-operating sequences that can be controlled by electrical or electromechanical means.

Temperature control is also readily accomplished by electronic means. This may be particularly simple and direct in the case of electric resistance heating. One useful method consists in supplying the heating load through a main winding of a two-winding iron-core reactor. This winding has a low resistance and, when the core is unsaturated, a considerable reactance. Of course the reactance will be greatly reduced if the core is saturated by means of a direct current in the auxiliary winding. This suggests the electronic control of a-c resistance heating. The furnace or oven windings are fed with a saturable-core reactor in series. A control panel embodies a rectifier to supply direct current for the reactor auxiliary winding and an electronic means of increasing or decreasing this current to adjust the furnace temperature in response to a thermocouple or other temperature detector. The elements embodied in this method of regulation are indicated in Fig. 28-64. This regulating method is suited to a great variety of processes suggested by heat-treating of all kinds, vulcanizing, strip annealing, glass lehrs, air heaters, etc.

Regulating applications of electronics are numberless, particularly speed regulating and voltage control. Most controls of this character involve the elementary system indicated in Fig. 28-65. In this system a piece of equipment, often very large, is to be controlled to meet a definite condition set by a reference standard. This standard and some small detecting device actuated by the main equipment feed a signal into a balance detector. This signal generally represents the difference between the standard quantity and a quantity represented by the operating condition of the main device. The balance detector determines the unbalance between the detecting device and the standard and supplies a corresponding signal to the controller. The controller then acts on the main equipment to reduce the variation between its operating condition and the standard condition represented by the reference standard. It thus effects the operation of the main equipment under preselected conditions.

An example of the foregoing control is the paper-machine drive. In this machine several machines must be coordinated, each performing a separate function on a continuous web of paper which changes in length as it passes through the machines, and must have their various speeds adjusted to accommodate them to the change in length. Furthermore, the web starts as a mixture of ½ per cent cellulose suspended in a matrix of water, travels at a speed between 300 and 1,500 fpm, and must result in a finished roll of paper of the required size between 0.00025 and 0.040 in. thick. While traveling at its normal speed the water is removed and the paper dried, sized, calendered, and reeled. Each separate machine has its own pilot generator driven from the driving motor of the machine. The voltage of the pilot generator is matched against a reference voltage and the difference applied to the grid of an electron tube. This signal acts to control the motor field—and hence its speed—through an electronic amplifier and an amplidyne exciter. This is but one example of numerous controls, extending from the lightest to the heaviest machinery, that are becoming increasingly common but precise in their action.

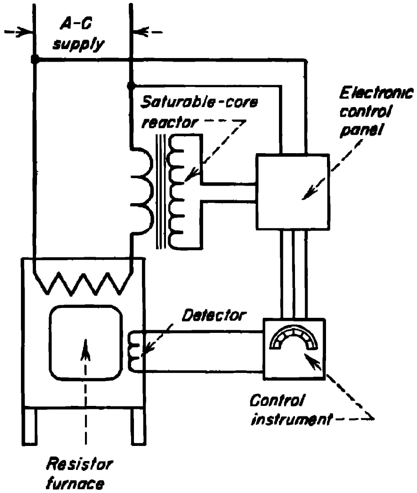

FIG. 28-64. Automatic temperature control in resistance furnace by saturable-core reactor.

The photoelectric cell, from its beginning as a control for the familiar door-opening device, has become one of the most useful electric tools of the century. It receives its excitation from a light source by direct radiation, by reflected light, by transmitted light, or by refracted light. This light impinging on the photocell sets free, by photoemission, electrons which become a current—usually of a few microamperes—which initiates the desired control, sounds a warning, opens or closes, counts, diverts, discards, limits, protects, selects, matches, or does any of thousands of things that it can do better and with vastly greater speed than can the human eye. A very few applications will be mentioned here as representative. Any large manufacturer of photoelectric-control equipment can supply lists of hundreds of successful applications of this tool so extensively used that it has long since ceased to be new.

One of the early applications of the photocell was to initiate movement by the obstruction of a light beam. As long as the beam of light shone on the photocell, its stream of electrons through a resistor would prevent the grid of an electron tube from becoming negative. An interception of the beam would remove the positive voltage, allow the grid to swing to negative, the tube to conduct, and, for example, the door to open. Another early representative use was for burglar protection. A labyrinth of beams of invisible ultraviolet or infrared light could be reflected between mirrors and ultimately reach a photocell. With any interruption in this multiple beam, the conduction of the controlled tubes would fail, and an audible or visual signal would be given. Innumerable assemblies, both standard and special, for sabotage fences were used during the war, and many were arranged to photograph the intruder automatically. When smoke intercepts the beam, a measurement of the density of the smoke may be made and recorded as an aid in smoke control. The same principles apply to turbidity control for giving a warning when a water supply is contaminated or automatically diverting the supply that has entrained sediment. It is also useful in the automatic supervision of chlorination, in filtering, determining hydrogen-ion concentrations, titration, indication of sludge level, and in innumerable applications where the partial or total interception of a beam of light will convey the needed information.

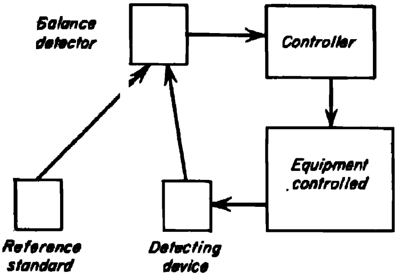

FIG. 28-65. Scheme for electronically matching performance of equipment with a reference standard.

Pinholes in metal or other opaque sheets are automatically detected by photoelectric cells and the defective sheet marked or diverted. Many plants also use machines that sort the sheets for thickness, count, and pile them, all by electronic means. Tons of beans are sorted and any discolored bean rejected into the chute for discards. Traffic-control equipment sorts traffic as readily as the bean sorter sorts beans. A vehicle of excessive height, or one carrying a load that projects too high, signals its condition to the traffic officer on duty, and if it is not immediately halted, the traffic light is changed to red. Traffic is often regulated in tunnels by devices that count the number of vehicles entering the tunnel and the number leaving. The difference represents the number of vehicles in the tunnel. Any excess over a preset number causes a red traffic light to show. At the same time, more ventilating fans are started. Of course this same scheme is applied to the movement of smaller and less cumbersome articles. The number of cars passing over a highway, as well as their speed, is often recorded by photoelectric equipment and the information thus secured used as a basis for traffic regulation and highway development.

The human eye, at best, can detect perhaps 10,000 colors and shades. The spectrophotometer, an instrument combining suitable filters and photoelectric cells, can differentiate between more than 2 million colors, hues, and shades and record them as certain curves or represent them as wave-length numbers in the visible spectrum. The information represents the reflectance of each different wave length by the sample as measured by the instrument. By this means two pieces of cloth manufactured in widely differing localities can be made to match perfectly. A banana ripened in Boston could, by proper control of temperature and humidity, be ripened to match a similar one in New Orleans. This instrument is invaluable in many applications in the medical, pharmaceutical, metallurgical, and chemical fields where delicate color changes are to be studied quantitatively.

The photoelectric cell may be used as an important device for aiding in temperature determination and control when the range is of the order where radiation pyrometry is useful. The improvement in uniformity and quality and decrease in spoilage of product of many steel mills have been marked by its use. A photoelectric pyrometer located to "look at" the center of the section of a bloom or bar in process may be interlocked with controllers in the pulpit, ensuring a finishing at the best rolling temperature. The photoelectric cell may watch the flame of a Bessemer converter through specially selected filters, record the progress of the blow, and catch the end point of the blow at exactly the preselected instant. In centrifugal casting the photocell watches the inside of the spinning pipe or tube as it cools and causes the driving motor to stop at the critical temperature. This enables the maximum use to be made of the casting machine, with a corresponding increase in output. In portland-cement manufacture, control of clinker temperature has increased the output per kiln, lowered fuel cost per barrel of cement, and produced cement of better and more uniform quality. The control is complete and automatic. A suitable optical system focuses the hottest part of the clinker bed on the photoelectric cell, and the speed of rotation is changed to maintain a constant clinker temperature. Temperature and kiln speed are also simultaneously recorded. A legion of photoelectric controls is suggested by these diverse applications, which are intended to suggest applicability rather than to be guides to practice.

Photoelectric positioning control brings scientific assurance that a piece of printed paper, a strip of steel, a web of cloth, a packaging container, or a power shovel weighing more than 1,000 tons will be in the proper position at precisely the right time. Weft straighteners, for squaring the weft with the warp in cloth, operate by light transmitted through the cloth to impinge on phototubes. When threads are square, the frequencies of light impulses to the tubes are equal; when skewed, are unequal. Electronic controls operate straightening mechanisms in the tentering machine to make proper corrections. Cloth speeds may range from 20 to 150 yd per min and from 20 to 100 picks per in. Side controls on steel strip reels furnish a control that produces smooth ends on the coil.

Another important device is represented by the automatic loop control applicable to all kinds of moving webs, whether light materials such as paper or cloth or heavy materials such as steel or brass strips. Figure 28-66 indicates the elements of this important control. The motor-driven rolls move the strip under control of the photoelectric cells through the amplifier and thyratron. When the loop of the strip intercepts about one-half the light from the source, the strip is in equilibrium. Any increase or decrease in the amount of light reaching the photoelectric cells acts through the amplifier and the thyratrons to change the field of the motor in such a direction as to restore equilibrium in speed.

Photoelectric control of registry is an extensive field of application and embraces such tasks as registry of postage stamps for perforating and cutting into sheets. The registry of four or more colors in turning out millions of copies of colored pictures, such as magazine covers, is accomplished by means of photoelectric register control. Each color must be printed directly on top of another within an accuracy of a few thousandths of an inch to avoid spoilage of the picture. An early application of photoelectric register is the proper location of the wrapper on candy bars or bread, for example, with respect to the object to be wrapped. This is done by making use of the pattern printed on the wrapper or by means of special registry marks printed on the wrapper. In many elevators the leveling of the car at the landings is done by photoelectric tubes mounted on the car and light sources mounted in the shaftway.

Elevator safety is often promoted by making it impossible to move a car when any part of a passenger's body extends into the doorway. This is easily accomplished by means of photoelectric relay control reacting to any obstruction of a light beam passing across the doorway. Injury to operators of dangerous machines is prevented by making it impossible for the machine to move when any part of an operator's person is in a dangerous position with respect to the machine.

Lighting control by photoelectric means is extensively practiced. When the daylight falls below a selected level in factories, offices, or schools, owing to the sun going down or to storm clouds passing over, the controls automatically turn on the lights. When the daylight returns to the proper level, the artificial lights are turned off. In installations of this character the method just described protects the worker's eye muscles from serious strain and fatigue. In stage and theater lighting where colors of light are presented in various combinations of contrast, harmony, and intensity, electronic control introduces features of regulation that are impossible by other means. The saturable reactor for changing light intensity and thyratrons or other highly conducting tubes are combined to be controlled by small buttons or keys or even by photoelectric cells with amplifiers. This permits the increase and decrease, the blending and resolution, of vast numbers of lighting combinations over extended areas with no more fingertip action than in the playing of a musical instrument. This control combination is used in many installations of decorative lighting of interiors or exteriors of buildings to enhance the architectural beauty. It is also used in the blending of rainbow colors in combinations of light that play on rising and falling water in fountains and pools.

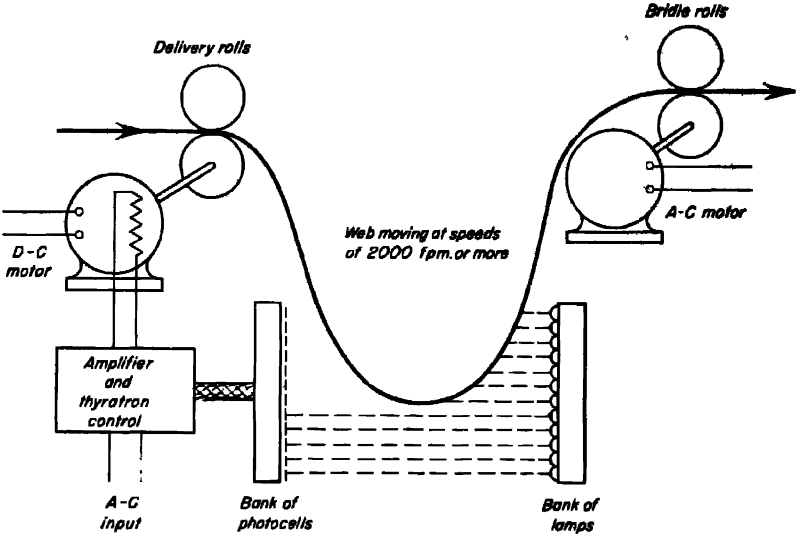

FIG. 28-66. Automatic control of a loop by electronic means.

Electrostatics play an important part in industry where control of the movement of small particles is necessary. Its most extensive application is in electrostatic precipitation and air cleaning. High voltage from transformers (usually of the order of 50,000 volts) is rectified by kenetrons or other high-vacuum-, hot-cathode-type tubes. When an electric charge is imparted to the particles of smoke or dust entrained in a gas and the gas subsequently conducted through the strong electric field existing between plates, the field impels the particles to the plates of opposite potential where they collect. The plates are then vibrated by mechanical means, and the collected particles fall into bins to be disposed of later without the need of shutting down any equipment. The foregoing is a description of the Cottrell method of precipitation. This process is widely used in the recovery of valuable materials, as well as for decontaminating air and other gases. Particles of copper carried out of copper converters in the flue dust during the blowing process are recovered by electric precipitation. Valuable arsenic dust from ore refining is recovered to prevent its settling on surrounding territory in the neighborhood, with the incidental advantage that the value of the recovered product soon pays for the precipitating equipment. Cement dust with a high potash content recovered from cement-kiln gases had a higher value per pound than the cement.

Electrocoating of many materials results in products unattainable by other means. An important example is the manufacture of high-grade abrasive papers and cloths. The arrangement is shown in Fig. 28-67. A backing coated with glue is fed into an electric field, from left to right in the top of the figure. On the bottom rolls a conveyor belt travels in the same direction. It receives grains of abrasive from a hopper which releases them in proper quantity. As they are carried into the electrostatic field they first stand on end with their longitudinal axes parallel to the field. They then fly upward along the field and are embedded in the glue with their long axes still parallel with the field. Since the particles all carry like charges, they are uniformly distributed over the surface with their sharp points outward. This results in an abrasive sheet that cuts much faster and more freely and lasts longer without loading with the abraded material than abrasive sheets with ordinary coating. If greater spacing between grains is desired—an open-coat abrasive—the rate of feed is decreased, and the electric charge evenly distributes the smaller quantity with the same uniformity.

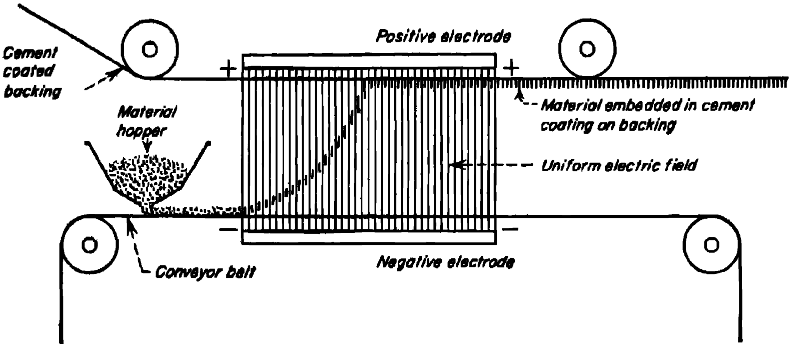

FIG. 28-67. Electrocoating in a high-intensity electric field.

The foregoing principles are made use of in the manufacture of many textile products. Flock materials are made by cementing cotton linters to a cloth or paper backing. Some 250,000 to 300,000 fibers per sq in. (over 500 per lineal in.) can be applied as compared to about 30 per cent of this number (150 per lineal in.) by weaving methods. The flock cloth may be made to resemble suede leather in surface appearance and feel. Other fabrics are produced by the use of longer fibers and resemble plush cloth and long-fibered rugs and carpets. Patterns are also produced by printing the design on the cloth in cement, in which case the fibers adhere to the printed design. The electrocoating process will expand greatly in the future, presenting many possibilities in upholstery and side-wall coverings and in the shoe and pocketbook trade, as well as in floor coverings.

MECHANICAL VIBRATION

Vibration engineering, an extensive and rapidly expanding field, involves many problems in isolation and absorption of mechanical vibrations. Some degree of unbalance is taken to be inherent in nearly all moving machinery, and the forces set up by this unbalance produce varying degrees of vibration and noise. In some degree this reduces the effectiveness of the machine and the efficiency of the employees in the neighborhood and shortens the life of the neighboring structures and equipment. Vibration is transmitted to the neighboring structures mainly by convection, the noise principally by radiation. Best machine design reduces vibration at its source but seldom eliminates it. Static balance is not sufficient to eliminate vibration; complete dynamic balance must be achieved, and this is not possible. A shaft of a machine may not possess uniform stiffness, or the field of a motor may be improperly distributed; a correction of vibration from these causes must be made when the machine is built. The seriousness of vibration effects depends upon their amplitude and frequency; and so far as human response is concerned, curves have been constructed from tests that indicate the tolerable frequencies at varying amplitudes.

The isolation, neutralization, or absorption of the vibration is often the only means of relief. Isolation is accomplished by elastic suspension, with or without damping. Although an isolated system possesses 6 deg of freedom, only motion in the direction of the principal disturbing force is considered in the solution of a vibration problem. In treating vibration troubles, several principles should be followed. The center of gravity of the isolated structure should be as nearly midway between the supports as possible; mounts should have the same static deflection; the center of gravity of the isolated structure should be as low as possible to improve stability (links may be necessary to achieve stability); where large masses and heavy impacts are involved, the center of gravity should preferably be below the supports.

So many unknowns enter into calculations of vibration isolation that results are not uniformly satisfactory. Damping coefficients are difficult to evaluate. Metal springs are easiest to calculate because of their small damping effect. A floor is often considered rigid but may in reality often contribute one of the masses and a restoring force of a two-mass system. Auxiliary devices for vibration damping are often employed, such as dashpots or other fluid damping devices used on automobiles. Friction dampers, such as those used on the crankshafts of automobile engines, are used. A cast-iron crankshaft may sometimes be used to advantage because of its internal damping due to the greater hysteresis of cast iron compared with that of steel. In the general case encountered, experience will reduce the amount of cut-and-try method necessary to solve a problem.

In commercial suspensions, metal springs will meet a wide range of vibration conditions. This is because a large variation in deflection may be had by changes in dimensions or material. Springs have the best stability when the working height is 1 to 1.5 times the diameter. Springs are not subject to damage from oil or temperature. If a spring suspension is found to transmit an excessive amount of sound, a layer of rubber, cork, or felt may be interposed.

Rubber mountings with the rubber bonded to the metal are extensively used with effective results. When used in compression, rubber will carry heavy loads, but allowance for the flow of the rubber must be made in designing the supports. When used in tension it is softer than when used in compression but is unreliable because of its weakness when abraided or nicked. Rubber is a very satisfactory material when loaded in shear, is also softer and more yielding than when used in tension or compression, and furthermore follows Hooke's law approximately when moderately loaded. It is especially useful in isolating the high frequencies of sound when through-metal is avoided in the supports. It is not so suitable as springs for isolating low frequencies. Oil and temperatures above 150°F deteriorate natural rubber, but under good conditions it should be serviceable for 5 or 10 years. Synthetic rubber stands oil and elevated temperatures better but loses its resiliency sooner.

Cork was one of the first materials used for isolation of vibration. Ground cork bonded into large slabs is a material having uniform structure and properties. Control by the bonding permits several densities for varying applications, and the deflection varies inversely as the density. It is also a good material for acoustical isolation, but through-metal must be avoided. It is particularly satisfactory for placing under a large mass and serves well when the machine is bolted to a large concrete block and the whole assembly placed on cork.

Hair felt is another material that has good properties for small deflections and high frequencies. Hence it is also a good material to limit sound transmission. The loading on large pads is usually limited to 50 psi, but on small confined pads it is sometimes used under greater pressures.

For sound and noise, screens and filters are frequently employed. An example of a screen would be a wall of sound-absorbing material interposed between the cause of the noise and the space to be quieted. An example of a filter would be the muffler for an automobile-engine exhaust. Large ducts are made, following the same principles. In such passages, free flow must be allowed in order to prevent excessive back pressure.

Vibration instruments responding to frequency, displacement, velocity, acceleration, or wave form may be necessary in the study of vibration problems. The most common need is a knowledge of frequency. Very low frequencies may often be counted by observation or by mechanical or electrical counters. For higher frequencies, vibrating-reed tachometers are useful. Various forms of stroboscopic instruments may also be used. Amplitudes of large displacements at low frequencies may be observed directly, or records of them made by means of a stylus attached to the vibrating member either with or without an amplifying system. Small vibrations of relatively higher frequencies are the more usual type and may be conveniently and completely studied by means of electrical instruments. A coil moving in a magnetic field is the commonest and most versatile device for this instrument. Its response may be shown on an oscilloscope or recorded on an oscillograph. Since the voltage produced by the movement of a coil in a uniform field is proportional to the velocity of the moving coil, a single integration of the velocity will indicate the displacement or a derivative—the acceleration.

SONICS AND ULTRASONICS

Sound Generators. The human ear can pick up sound with frequencies from about 16 to 20,000 cps, the upper limit often decreasing with increasing age. Ultrasonic frequencies are those above the audible range.

There are three general types of sound generators in commercial use, two using electric energy as the source of power and the other using compressed air. The *piezoelectric* generator employs a crystal, *e.g.*, quartz or Rochelle-salt, through which is passed a high-frequency alternating current. The crystal expands and contracts at signal frequency, generating sound waves. At the present time (1950) commercial equipment using this principle could probably be built to deliver in the order of 500 kw output. The *magnetostriction* generator employs a metal tube, bar, or laminated rod (nickel gives good results) inside a coil of wire through which is passed a high-frequency alternating current. The tube, bar, or rod expands and contracts in the resulting alternating magnetic field, and when attached to a metal diaphragm, high-frequency sound waves are produced at twice the signal frequency unless a polarizing current is applied. A machine using *compressed air*, similar to a siren, has been built which operates over a frequency range of 1,000 to 200,000 cps with intensities from about 100 to 170 db. Between 50 and 70 per cent of the energy in the compressed air is converted to sound energy.

Sonic vibrations produced by sonic oscillators are made use of in industrial processes to an increasing extent as their potentialities are recognized. One example is the application to the homogenizing of the fat and curd in milk. It consists of a stainless-steel cylindrical chamber about 2 ft in diameter with a bolted-on cover on one side and the other enclosed with a stainless-steel diaphragm. The diaphragm is driven electromagnetically at a frequency of 360 cps. Milk to be treated is introduced into this chamber at the outer edge of the cover and leaves the chamber at a central opening where it is subjected to the intense vibration at the center of the diaphragm. This treatment breaks up the fat globules so that no cream line can form and results in a softening and fining of the curds.

Another application of the vibration principle is an oscillator for Fourdrinier shake sections on paper machines. This device is termed a compensator. It makes use of a compensating mass in conjunction with a spring system which is built to resonate as a system with the Fourdrinier section. During operation the Fourdrinier mass and the mass of the compensator move in opposite directions so that the energies are balanced. This permits greater shake speeds and stroke with the result of better felting and higher wire speeds. This is an example of slow vibrations with a mass of several tons at something like 500 to 750 strokes per minute through an amplitude of ½ in. or so.

For experimental work with sound-wave frequency operating on liquid chemicals and living organisms, a 9-kc oscillator was found to be of optimum frequency. It depends for its operation on the magnetostricture principle. Its operation is an extension and contraction of a laminated-nickel rod activated by an alternating magnetic field. The bottom of the test cup forms the diaphragm and transfers the energy of vibration to the liquid under test within the cup. Its use in bacteriological and biological research has shown that the disintegration of bacteria is more rapid at this rate of vibration than at higher rates. Research workers in the production of stable emulsions for heavy chemical industries have also found that more energy can be introduced into the sample at this frequency than at ultrasonic frequencies.

Ultrasonic vibrations, on the order of 0.5 to 12 megacycles, have found use for the transmission of waves directed in straight lines through solids for the detection of flaws. One such electronic instrument embodies a pulse generator whose frequency is controlled by a quartz crystal. By the reflection method, one crystal may be employed as both transmitter and receiver. Ultrasonic vibrations are sent through a material under test and are reflected from the opposite end or from an intermediate point where a flaw exists. The indicator is a 9-in. cathode-ray tube on which the pattern of the wave appears, together with timing wave by which the flaw is located. The quartz-crystal searching unit must be applied to a relatively smooth and flat surface—about GE5 f* is recommended. As the search unit is moved over the reflected surface, the reflected pattern is observed and interpreted by the operator. This method of nondestructive testing is particularly valuable in the testing of large assembled equipment where disassembly is costly and inconvenient. Such cases are represented by large shafts, crank pins, and axles such as are used on large machines and locomotives.

The compressed-air high-intensity sound generator is being used to agglomerate fine particles of furnace black made from natural gas. A frequency of 3.5 kc and an intensity of 160 db is used to treat about 2,000 cu ft of gas per min and results in the recovery of about 96 per cent of the carbon black in the cyclones which follow the agglomerator tower. In general, sound-wave agglomeration can be used when the particles are below 10 to 15 microns in diameter and their concentration exceeds 2 grains per cu ft. Similar applications under consideration are soda and salt-cake recovery from paper- and pulp-mill stack gases, precipitation of sulfuric acid mist, and recovery of dust from flue gases. The generator may also have application in the dispersal of fog, in the more rapid spray drying of powdered soap, milk, and drugs, and in other fields.

Among other actual or possible applications of ultrasonics may be mentioned the acceleration of chemical reactions, modification of crystallizations, control of polymerization and depolymerization, creation and destruction of emulsions and colloidal suspensions, improvement in photographic-emulsion stability and grain size, aging of beer, wine, and whisky, degasification of liquids and foundry melts, alloying of molten metals, submarine signaling and detection of underwater objects, destruction of microorganisms, and knifeless surgery.

GRAVITATION AND CENTRIFUGAL SEPARATION

Gravity for separation has been a fundamental in mining practice for centuries. Now the same methods, certainly as to fundamentals, can be applied with advantage to many manufacturing processes. The classification of grain size by screening with perforated sheets, wire screens, and bolting cloth needs no remarks. As a review, the general principles of classification by free settling in a liquid may be stated.

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* A surface-smoothness measure.

1. When two particles have different specific gravities, the one with the higher will fall faster.

2. Of two particles, the larger will settle faster.

3. Of two particles that will just pass through the same screen, the more nearly spherical particle will settle faster than the long slim one, and the latter will settle faster than the flat one.

4. Of two particles—one to which air bubbles adhere, the other free of bubbles—the latter will settle faster.

5. With a group of particles strongly magnetic and another group not, the former will clot together by mutual attraction and settle faster.

6. With two liquids of different density, a particle settles more rapidly in the lighter liquid.

7. With two liquids of different viscosity, a particle will settle more rapidly in the less viscous liquid.

The foregoing facts are the results of man's observations, probably extending back to the dawn of civilization. For spherical bodies, which are seldom the ones involved, Stokes's equation can be used, where the velocity of fall V under the force of gravity of a spherical particle of density ρ_p and coefficient of viscosity η is given by

$$V = (2/9\eta)gr^2(\rho_p - \rho_d)$$

Tables of settling times for different substances under the influence of gravity may be found in texts on ore dressing and thus save much calculation. Fine abrasives are commonly classified by settling in water or light oil. When ground in water and followed by fractional sedimentation, the process is called levigation. The product is often referred to as 2-hr water, 4-hr water, etc., as a statement of fineness. Concentrators, coarse-sand jigs, fine-sand jigs, flotation processes, etc., as described in mining literature, can be reviewed with profit if quantities of materials are to be classified according to size.

The centrifuge and centrifugal separator permit the use of a force many times that of gravity, and the larger centrifugal field makes possible the separation of particles of smaller size and mass difference. If we consider the force in a centrifuge as a ratio to the force due to gravity and express it as pounds force on the periphery of the basket,

$$Fg = \tfrac{1}{2}\omega^2 D \qquad \text{or} \qquad F = Dn^2/5{,}870$$

where F = force, lb, at the periphery of the basket
g = 32.2
n = rpm
ω = radians per sec
D = diameter of the basket, ft

Centrifugal extraction is usually the problem of removing a liquid from a solid mass. The problem of the extractor design is one of balancing the load or else providing structure and bearing that will endure the unbalance. Special-purpose machines are made for the extraction in chemical industries, sugar refineries, coal washing, laundries, chips in machine shops, and many other kinds of waste. Extractors for different materials will require different designs, and the problem of loading and unloading the machine is often the most difficult factor encountered.

Separators for milk, oil, and other liquids call for a bowl- or plate-rotating member rather than a basket. The problem of balance is largely eliminated, but centrifugal force becomes such that particular attention must be given to the strength of rotating parts. In cream and oil separators, groups of hollow cones surround the central shaft, and the liquid is fed to the cones continuously, the heavy portion being carried in a continuous flow to the low outside edges of the cones, while the cream rises to the top of the cones in the center. Other arrangements embody the same principles.

Considerable work has been done in recent years on ultracentrifuges. Professor Svedberg's early ultracentrifuges gave centrifugal fields of 500 to 15,000 gravity. His second type of oil-turbine ultracentrifuge will give centrifugal forces within the range of 15,000 to 750,000 gravity. *This is an example of results that can be achieved by precise design.

RADIOACTIVITY

Natural radioactivity was discovered by Becquerel in 1896. The same year the Curies isolated from barium, radium, which has 3 million times the activity of barium. They isolated a substantial amount (about 0.1 gm of radium chloride) by 1902. It was later found that this element was a step in the spontaneous transmutation of radioactive uranium into a stable isotope of lead. In 1919, Lord Rutherford accomplished the artificial transmutation of nitrogen into carbon. This was the first answer to the alchemists' search of centuries for conversion of one element into another until they finally lost hope, and chemists incautiously declared that the elements were immutable. After 1930 the advance in this field was very rapid, and in 1939 nuclear fission was discovered.

In the structure of matter there are seven known building blocks*—the proton, the neutron, the positive meson, the negative meson, the positron, the electron, and the neutrino—whose existence is suspected but not demonstrated. Three of these particles are fundamental—the proton, neutron, and electron. The atomic weight of the element is determined by the number of protons and neutrons combined to form the nucleus. The atomic number and chemical properties of the element are determined by the number of protons in the nucleus. If a proton is added or subtracted from the nucleus, the chemical characteristics of the element are completely changed; and since each positron carries a positive charge, its electric charge is increased or decreased by one unit. If a neutron is added or removed, only the weight of the atom is changed. The weight of a neutron is approximately equal to the weight of a positron. For each positron in the stable atom there is an electron which must carry a negative charge equal in magnitude to the positive charge in the positron. Hence the number of electrons in a stable atom is equal to the atomic number of the element. The electron has a mass about 1/1,840 the mass of the proton or neutron. The atomic weight of an atom is found by comparing its weight with the weight of the oxygen atom, O = 16.0000. The number of neutrons in an atom is the atomic weight minus the atomic number; *e.g.*, $_{92}U^{238}$ has 238 − 92 = 146 neutrons as well as its 92 positrons and 92 electrons.

Isotopes are substances having identical chemical properties but different atomic weights. For example, zinc is a mixture of substances having atomic weights of 64, 66, 67, 68, and 70 in such proportions as to give a mean value of 65.38 for the atomic weight of zinc. These are stable isotopes having the nuclear charge equal to the number of extranuclear electrons. Zinc may also have the radioactive, or unstable isotopes, 63, 65, and 69. Carbon, $_6C^{13}$, is a natural heavy isotope which exists as about 1.1 per cent with $_6C^{12}$ as the other 98.9 per cent of normal carbon. Four other isotopes of carbon can be produced. Natural unstable radioactive isotopes change to stable ones by emitting particles which in the early days of their discovery, when little was known of their character, were termed alpha, α, beta, β, and gamma, γ, rays. Further research revealed that α particles consist of two protons and two neutrons with two units of positive charge, really nucleuses of helium atoms. Beta particles are electrons and hence have unit negative charge. Gamma rays are similar to very-high-voltage X-rays, contain no charge, and cannot be deflected by a magnetic field. They have penetrating power exceeding ordinary X-rays, and it was soon discovered that they have an inhibitory action on many cancerous growths without destroying adjacent healthy tissue when the treatment is suitable. When a radioactive substance decomposes, the emission of an α-particle transmutes the element into a new substance with an atomic number of two units less and an atomic mass of four units less. When it emits a β-particle, it changes to a substance with an atomic number one unit greater but with no change in atomic mass. The time that it takes for a radioactive element to decay to half its original activity is called its half-life. The half-life of any radioactive isotope is a definite value, but different isotopes of the same element have greatly varying half-lives; for example, the half-life of $_6C^{10}$ is 8.8 sec, that of $_6C^{11}$ is 20.5 min, and that of $_6C^{14}$ is 5,000 years.

The uranium pile is a massive structure of alternate layers of pure uranium and carbon. The first one made had over 6 tons of uranium in it. Cavities are maintained in the pile for the introduction of cadmium rods, which will arrest the reaction by absorbing neutrons and will keep the reactor from getting out of hand. Pure carbon rods, called moderators, may also be inserted to slow down the neutrons without absorbing them. In this manner the pile produces quantities of slow-speed neutrons which are effective in disintegrating nucleuses. The sample to be bombarded is placed in welded aluminum vessels and inserted into openings in the pile where the neutrons penetrate the aluminum without being absorbed and transform the enclosed element into a radioactive isotope. This transformation is accomplished in several ways. If the nucleus of the atom picks up a neutron, it may emit an α-particle, a positron, or a γ-ray. Radioactive carbon C^{14} is readily made in the uranium pile by the action of slow neutrons on nitrogen.

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* 1950.

$$N^{14} + n^1 \rightarrow C^{14} + H^1$$

A solution of ammonium nitrate may be subjected to the neutron bombardment, and the result is a bicarbonate ion. Radioactive-carbon samples contain as much as 5 per cent of the radioactive isotope.

The curie is the unit of measure of radioactivity. One curie of radium is approximately equal to 1 gram of the element. It is actually an amount of the substance such that 3.71×10^{10} atoms per sec of the substance are undergoing disintegration. The unit is a large one, and it is found convenient to state radioactivities of commercial isotopes in millicuries.

The Geiger-Mueller counter is commonly used to determine the radioactivity of a sample of an isotope. This may be a metal cylinder for one electrode and a concentric axial wire for the other, all enclosed in a glass vessel with the enclosed gas at a pressure of a few centimeters of mercury. A direct voltage a little below the glow-discharge voltage is applied to the electrodes. When any ionizing particle enters the ionizing chamber, it produces ions which produce other ions by collision. A momentary current flows through an amplifier, similar to an amplifier on a radio receiver, and an automatic counter records the number of pulses.

Radioactive isotopes have innumerable applications, and while their use has been extensive in medicine, the extension of the use to industrial processes is taking place rapidly. For example, the sodium of ordinary table salt is activated and made into an isotonic saline solution and introduced into the veins of a patient. A Geiger counter will show the efficiency, time, and extent of the circulation. The importance of this will be recognized if it is considered that a limb with defective circulation is to be amputated. The point where effective circulation ends is readily found. Many organs of the body selectively absorb certain elements or compounds; *e.g.*, the thyroid gland uses iodine in synthesizing thyroxine, and the bones selectively absorb phosphorus. These facts point to the use of elements or compounds containing radioactive material that will go directly to the desired part of the body and there release their radioactivity. This can be used as a tracer or may even be used for radiation treatment where X-rays cannot be used. The time of dosage is also controllable by selecting an isotope whose life is of the desired magnitude, it being remembered that their half lives may be a few seconds or thousands of years. As tracer elements the radioactive isotopes are revealing many facts otherwise unobtainable. Many biochemical reactions can be studied and their rates determined by following the course of the tracer atoms.

This tagging of atoms aids, too, in the study of plant physiology and plant pathology. Phosphorus (P^{32}), because of the extensive use of phosphorus in plant growth, is suitable for these studies as well as for tracing the path of fertilizer from its place in the soil through the plant tissues to the fruit. C^{14} may be used in studies of photosynthesis by exposing the leaves that have grown in an atmosphere of radioactive carbon dioxide to photographic film. The imprint of every part of leaf or stem that contains the radioactive element will be shown on the film.

In **metallurgical processes** the importance of radioactive tracers will be obvious. If it is suspected that a particular element is being carried away in the slag, a quick reading with a Geiger counter before the metal is poured will indicate the presence or absence of the required amount. A textile or paper operation may be hampered by accumulations of static electricity. Small quantities of radioactive isotopes suitable for alloying with parts of the machine such as rolls or shafts would act to discharge the static electricity from an electrified body.

Safety of personnel in radioactive work can be assured by careful supervision. When tolerances in counts per minute on Geiger counters for various articles of laboratory clothing, for workers' hands, and for other parts of the body are established, it is a simple matter to perform checks at proper daily periods. A dental X-ray film with a paper clip on it can be worn by the worker and replaced weekly. The removed film is then developed and observed for an image of the paper clip on its emulsion and density of background measured. A regular commercial film-badge service is also available for health-maintenance purposes. Normal tolerance is conventionally taken as 0.1 roentgen per 24-hr period. The badges are usually equipped with two films, one of high sensitivity, the other of low sensitivity. A weekly exchange of badges between the supplier and the subscriber ensures a constant check on the radioactivity to which the worker is exposed. A record of the measured roentgens is kept and reported to the subscriber.

Nuclear fission of uranium occurs when a neutron collides with a U^{235} nucleus and combines with it, forming a U^{236} nucleus. Since the resulting nucleus is unstable, it will split into two fragments, each itself an atomic nucleus the sum of whose atomic weights is 92. Suppose an atom split into strontium and xenon nucleuses with the production of 17 neutrons:

$$_{92}U^{235} + {}_0n^1 \rightarrow {}_{38}Sr^{88} + {}_{54}Xe^{131} + 17\,{}_0n^1$$

There must be a conservation of charge $38 + 54 = 92$ and mass number

$$88 + 131 + 17 = 236$$

Usually there is unsymmetrical splitting with one nucleus having a mass number between 80 and 110, and the other between 125 and 155 being formed. This nuclear fission results in the emission of a large amount of energy, about 5×10^{12} cal per gram-atom (235 grams of U). This is about 38.2×10^9 Btu per lb. It is about 2,800,000 times the amount of energy produced by burning the same amount of coal, or 14,600,000 times the amount liberated by 1 lb of nitroglycerine. Uranium as a source of energy may be 2,800,000 times as valuable as coal. Thorium isotope $_{90}Th^{232}$ has also been shown to undergo fission under the bombardment of fast neutrons. It may be that all the elements with atomic number greater than 90 can be made to react to release atomic energy. There are large amounts of thorium and uranium in the earth's crust—about 12 parts per million and 4 parts per million, respectively. Thorium is the thirty-fifth element in abundance and uranium the forty-sixth; the two together about equal in amount lead, which is thirty-first in abundance.

It cannot be assumed from the foregoing discussion that commercially produced atomic power is just around the corner. Methods of handling such prodigious quantities of power and temperature must be found. Ways may be found to release the energy efficiently and slowly, and long-time prospects look bright. There are several essentials in the problem of making atomic power a commercial reality. The energy must be produced in a usable form; the material used must have a usable rate of generating power; there must be adequate stocks of the fuel to operate a power industry; wastage of this costly material must be prevented, and the valuable ashes—unspent nuclear fuel—must be recovered. Improvement in fuel-burning plants may be safely continued without fear of early obsolescence because of atomic power. There will be ample warning before atomic power even supplements the present prime forms of energy release.

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SECTION 29

GRAPHICAL MATHEMATICS

BY R. D. DOUGLASS, PH.D., SC.D.

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GRAPHICAL MATHEMATICS

BY R. D. DOUGLASS, PH.D., SC.D.

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GEOMETRICAL CONSTRUCTIONS

To Bisect a Line AB (Fig. 29-1). From A and B as centers, and with equal radii, describe arcs intersecting in C and D, and draw CD, which bisects AB at E.

To Construct a Perpendicular to a Given Line from a Point A **Outside the Line** (Fig. 29-2). With A as a center, describe an arc cutting the line in B and C, and bisect BC at D. D is the foot of the perpendicular.

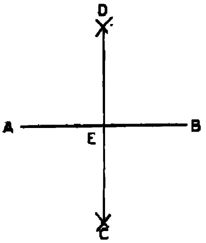


FIG. 29-1. To bisect a line AB.

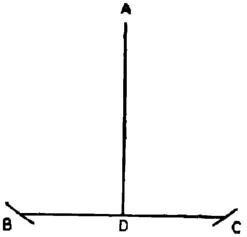


FIG. 29-2. To construct a perpendicular to a given line from a point A outside the line.

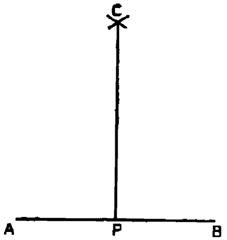


FIG. 29-3. To construct a perpendicular to a given line at a given point P.

To Construct a Perpendicular to a Given Line at a Given Point P. Lay off $PA = PB$ (Fig. 29-3), and with A and B as centers draw arcs intersecting at C. PC is the required perpendicular.

To Divide a Line into n **Equal Parts** (Fig. 29-4). Through A draw a line AC making any angle with AB, and lay off n equal segments on the line. Connect the end of the last segment with B, and draw parallels through the other divisions. These parallels will divide the line AB into n equal parts.

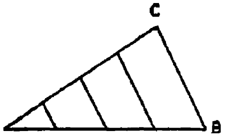


FIG. 29-4. To divide a line into n equal parts.

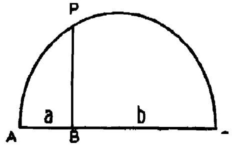


FIG. 29-5. To construct a mean proportional to two lengths a and b.

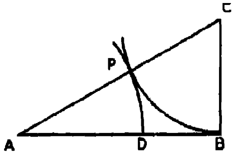


FIG. 29-6. To divide a line AB in extreme and mean ratio.

To Construct a Mean Proportional to Two Lengths a **and** b (Fig. 29-5). Lay off $AB = a$, $BC = b$, and construct a semicircle on AC as a diameter. Draw a perpendicular at B meeting the circle at P. $BP = \sqrt{ab}$.

To Divide a Line AB **in Extreme and Mean Ratio** (Fig. 29-6). At B construct a perpendicular BC equal to one-half AB, and connect the points A and C. Along CA lay off $CP = CB$, and along AB lay off $AD = AP$. D is then the required point, *i.e.*, $(AD)^2 = AB \cdot BD$.

To Bisect an Angle AOB **(Fig. 29-7).** Lay off $OA = OB$. With A and B as centers, describe arcs intersecting at C. OC is the required bisector.

To Pass a Circle through Three Given Points A, B, **and** C **or to Locate the Center of a Given Circle (Fig. 29-8).** Draw the perpendicular bisectors of AB and BC. These will intersect at the center O.

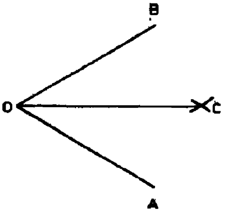


Fig. 29-7. To bisect an angle.

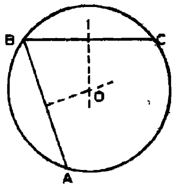


Fig. 29-8. To pass a circle through three given points A, B, and C or to locate the center of a given circle.

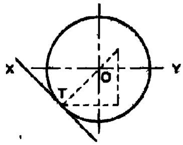


Fig. 29-9. To construct a tangent to a circle at a given point of contact T.

To Construct a Tangent to a Circle. (1) At a given point of contact T (Fig. 29-9): Draw TX perpendicular to OT. (2) From a given external point A (Fig. 29-10): Construct a semicircle on OA intersecting circle at T; then T is the required point of tangency.

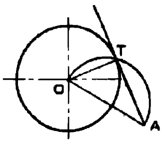


Fig. 29-10. To construct a tangent to a circle from a given external point A.

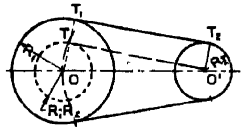


Fig. 29-11. To construct a common tangent to two given circles.

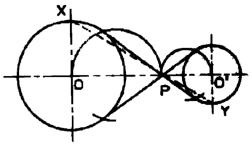


Fig. 29-12. To construct a common tangent to two given circles.

To Construct a Common Tangent to Two Given Circles. (1) Let O and O' be the centers and R_1 and R_2 the radii ($R_1 > R_2$) (Fig. 29-11). With O as center draw a circle of radius $R_1 - R_2$; draw a tangent to this circle from O'; through point of tangency T draw OT_1; draw $O'T_2$ parallel to OT_1. T_1 and T_2 are the points of tangency. (2) Draw line XY (Fig. 29-12) intersecting OO' at P; construct tangents to the given circles from P.

To Pass a Circle through Two Fixed Points A **and** B **and Tangent to a Given Line** L **(Fig. 29-13).** Let AB cut the line L in C. Pass any circle through A and B, and let CT be tangent to this circle. Along L lay off CD and CE equal to CT. Then both D and E satisfy the condition of tangency. The centers of the required circles lie on the perpendicular bisector of AB.

To Draw a Circle Tangent to Each of Three Given Lines, No Two of Which Are Parallel (Fig. 29-14). Draw the bisectors of the three angles; these will meet in the center O. The perpendiculars from O to the three lines will determine the points of tangency.

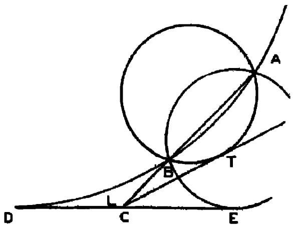


Fig. 29-13. To pass a circle through two fixed points A and B and tangent to a given line L.

To Construct a Circle of Radius R **Tangent to Each of Two Fixed Lines** XY **and** MN **(Figs. 29-15a, 29-15b).** Draw lines parallel to and at a distance R from each of the lines. The intersection of these lines is the center of the circle.

To Construct a Circle of Radius R **Tangent to a Circle of Radius** R_1 **with Center at** O **and Tangent to a Fixed Straight Line** MN **(Fig. 29-16).** Draw a line XY parallel to MN and at a distance R from it. With O as a center and $R + R_1$ as a radius, describe an arc intersecting line XY at A. A is the center of the desired circle.

To Construct a Circle of Radius R Tangent to Each of Two Fixed Circles of Radii R_1 and R_2 (Fig. 29-17). Draw an arc of radius $R_1 + R$ with the center of the circle of radius R_1 as a center. Draw an arc of radius $R_2 + R$ with the center of the circle of radius R_2 as a center. These arcs intersect at X, the center of the desired circle.

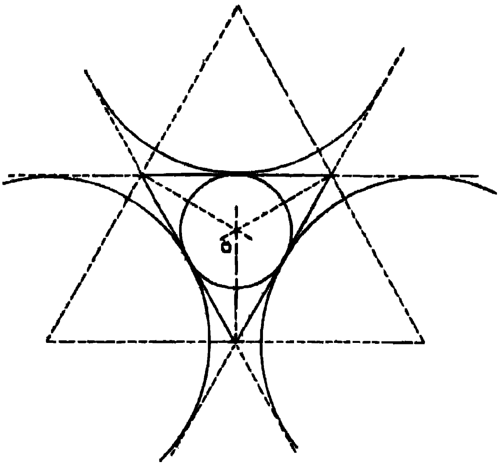


FIG. 29-14. To draw a circle tangent to each of three given lines, no two of which are parallel.

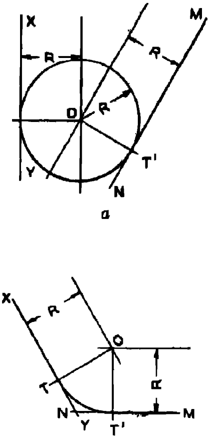


FIG. 29-15. To construct a circle of radius R tangent to each of two fixed lines.

To Construct a Circle of Radius R Tangent to Two Inside Circles of Radii R_1 and R_2, Respectively (Fig. 29-18). Draw an arc of radius $R - R_1$ with the center of the circle of radius R_1 as a center. Draw an arc of radius $R - R_2$ with the center of the circle of radius R_2 as a center. These arcs intersect at X, the center of the desired circle.

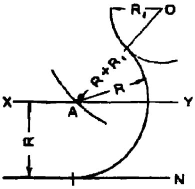


FIG. 29-16. To construct a circle of radius R tangent to a circle of radius R_1 with center at O and tangent to a fixed straight line.

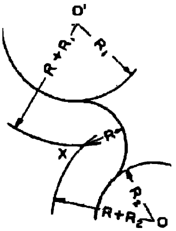


FIG. 29-17. To construct a circle of radius R tangent to each of two fixed circles of radii R_1 and R_2.

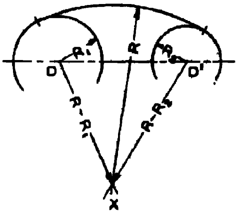


FIG. 29-18. To construct a circle of radius R tangent to two inside circles of radii R_1 and R_2.

Parabola. *Equation.* $x^2 = a^2y/b$ (Fig. 29-19).

Construction. Divide the distance a into n equal parts and the distance b into the same number of equal parts. Draw lines from O to the points of division of b and lines to each of the divisions of a parallel to the y-axis. The intersections of the lines bearing the same numbers in Fig. 29-20 will be points on the required parabola.

The distance from the vertex of the parabola $x^2 = (a^2/b)y$ to its focus and to its directrix is $a^2/4b$.

To Construct a Tangent to the Parabola at a Point P on the Parabola (Fig. 29-21). Lay off $OB = OA$ or $FP = FB$. BP is the required tangent.

To Construct a Tangent to the Parabola from Any External Point P (Fig. 29-22). With P as a center and radius PF, draw a circle cutting the directrix at A. Draw AB parallel to OF; then B is the required point of contact.

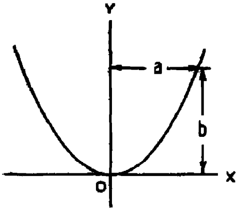


FIG. 29-19. Parabola $x^2 = a^2y/b$.

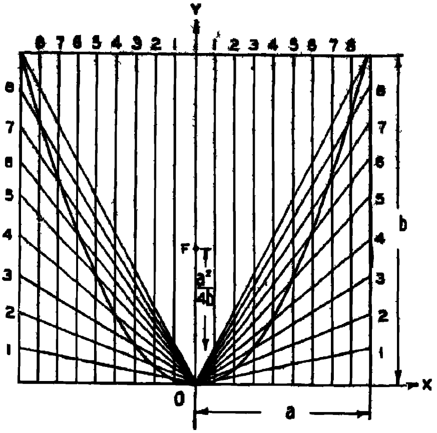


FIG. 29-20. To construct the parabola $x^2 = a^2y/b$.

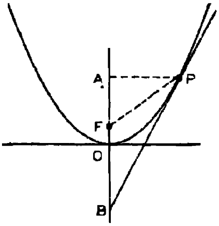


FIG. 29-21. To construct a tangent to the parabola at a point P on the parabola.

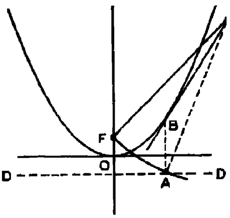


FIG. 29-22. To construct a tangent to the parabola from any external point P.

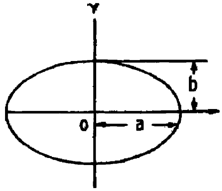


FIG. 29-23. Ellipse.

Ellipse. *Equation:* $\frac{x^2}{a^2} + \frac{y^2}{b^2} = 1$ (Fig. 29-23).

Construction: To construct an ellipse of length $2a$ and width $2b$ $(a > b)$, divide a into n equal intervals, and divide b into n equal intervals. Draw lines from the lower end of the minor axis (Fig. 29-24) through the end of each interval into which a is divided, and draw lines from the upper end of the minor axis through the end of each interval into which b is divided, labeling each line as indicated in Fig. 29-24. The intersections of like numbered lines lie on an ellipse.

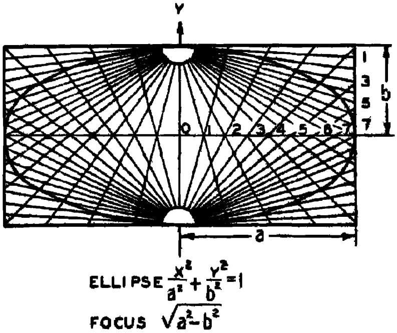


FIG. 29-24. To construct the ellipse.

The distance from the center of the ellipse $(x^2/a^2) + (y^2/b^2) = 1$ to a focus is $\sqrt{a^2 - b^2}$.

To Construct a Tangent to an Ellipse. (1) At a point on the ellipse: Bisect the angle between PF_1 and PF_2 (Fig. 29-25). (2) From a point P external to the ellipse (Fig. 29-26): With P as a center and radius PF_2, draw an arc; with F_1 as a center and radius $2a$, draw an arc intersecting the first in X and Y. XF_1 intersects the ellipse at T_1, YF_1 intersects at T_2. T_1 and T_2 are the points of tangency.

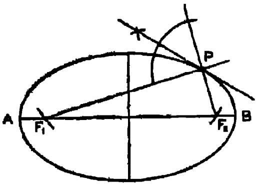


FIG. 29-25. To construct a tangent to the ellipse at a point on the ellipse.

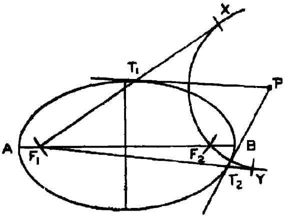


FIG. 29-26. To construct a tangent to the ellipse from a point P external to the ellipse.

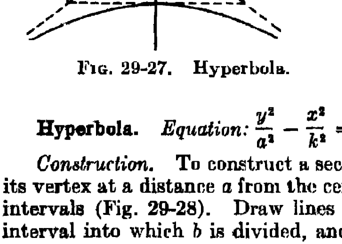


FIG. 29-27. Hyperbola.

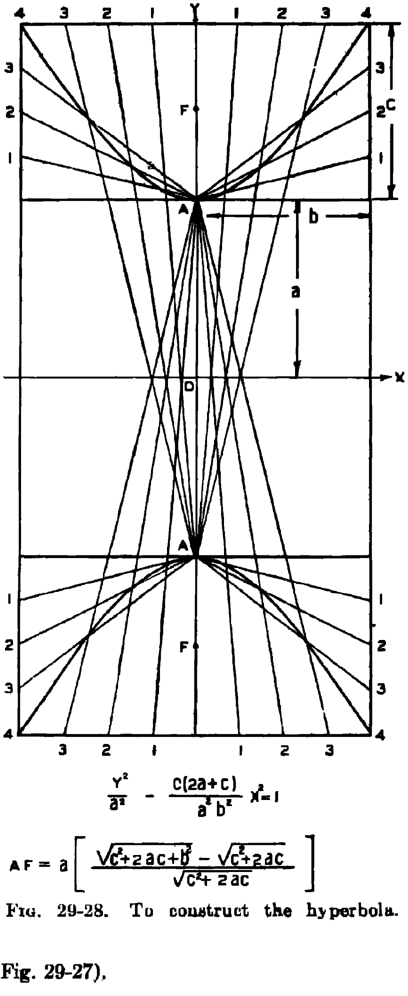

4 3 2 1 Y 1 2 3 4
3
2
1
F
A
b
a
c
O
X
A
F
4
3 2 1 1 2 3
$\frac{Y^2}{a^2} - \frac{c(2a+c)}{a^2b^2}X^2 = 1$
$AF = a\left[\frac{\sqrt{c^2+2ac+b^2} - \sqrt{c^2+2ac}}{\sqrt{c^2+2ac}}\right]$

FIG. 29-28. To construct the hyperbola.

Hyperbola. *Equation:* $\frac{y^2}{a^2} - \frac{x^2}{k^2} = 1$ (Fig. 29-27).

Construction. To construct a section of a hyperbola of width $2b$ and height c with its vertex at a distance a from the center of the hyperbola, divide b and c into n equal intervals (Fig. 29-28). Draw lines from the lower vertex through the end of each interval into which b is divided, and draw lines from the upper vertex through the end of each interval into which c is divided, labeling the lines as indicated in Fig. 29-28. The intersections of like numbered lines lie on a hyperbola. The opposite branch of the hyperbola is constructed in a similar manner.

The distance from the center of the hyperbola $(y^2/a^2) - (x^2/k^2) = 1$ to a focus is $\sqrt{a^2 + k^2}$.

To Construct a Tangent at the Point P on the Hyperbola (Fig. 29-29). Draw PA and PB parallel to the asymptotes, and let OC equal $2OA$ and OD equal $2OB$. CD is the required tangent at P.

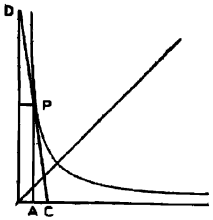


Fig. 29-29. To construct a tangent to the hyperbola.

GRAPHICAL REPRESENTATION OF FUNCTIONS

Scale. A graphical scale is a curve or straight line on which are marked a series of points or strokes corresponding to a set of numbers arranged in order of magnitude. If the distances between successive strokes are equal, the scale is said to be *uniform* (Figs. 29-30*a*, 29-30*b*). If the distances between strokes are unequal, the scale is said to be nonuniform (Fig. 29-30*c*).

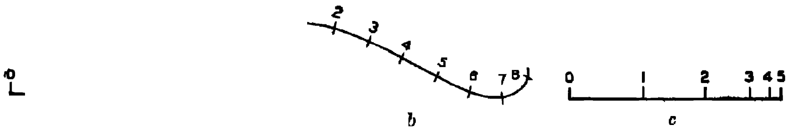


Fig. 29-30. Scales: (*a* and *b*) uniform; (*c*) nonuniform.

Representation of a Function by a Scale. Consider the function u. Lay off from a fixed point O on a straight line or curve lengths equal to mu units; mark at the strokes indicating the end of each segment the corresponding value of u. If m is unity and the unit of measure is an inch, the resulting scale is $x = u$ inches (Fig. 29-31*a*).

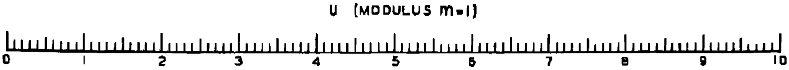


Fig. 29-31*a*.

For a function u^2 of the variable u; from the table

$$u = 0, 1, 2, 3, \cdot \quad \cdot 10$$
$$u^2 = 0, 1, 4, 9, \cdot \quad \cdot 100$$

and lay off the distances $x = mu^2$. If $m = \frac{1}{10}$ and the unit of measure is the inch, then u will be the number on the scale at the end of $\frac{1}{10}u^2$ inches measured from the fixed point, or origin, of the scale (Fig. 29-31*b*).

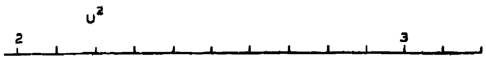


Fig 29-31*b*.

m is called the scale *modulus*, and $x = mf(u)$ is called the *equation of the scale* of the function $f(u)$. For a given range of the variable u and for a desired length of scale, m is determined. Thus if it is desired to represent $f(u)$ on a scale l inches long and u varies from $u = u_0$ to $u = u_l$, then the equation of the scale is $x = mf(u)$. When $u = u_0$, $x_0 = mf(u_0)$; and when $u = u_l$, $x_l = mf(u_l)$; and it follows that

$m[f(u_l) - f(u_0)] = x_l - x_0 = l$. Therefore

$$m = \frac{l}{f(u_l) - f(u_0)}$$

and the resulting scale equation is

$$x = \frac{l}{f(u_l) - f(u_0)} f(u)$$

Two of the most commonly used scales are $x = mu$ (uniform) and $x = m \log u$ (nonuniform). These and other scales are illustrated in Figs. 29-31*a* to 29-31*i*.

Figure 29-31*c* shows a uniform scale of length 10 in. with a range of values from 0 to 1. Its modulus m is 10, and its equation is $x = 10u$. Figure 29-31*a* shows the same scale reduced in size to one-tenth that of Fig. 29-31*c*, and its equation is $x = u$.

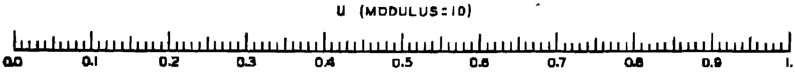


Fig. 29-31*c*.

Figure 29-31*d* shows a scale of log u and is called a logarithmic scale. The range of values of u is from 1 to 10, its original length is 10 in., and its scale equation is $x = 10 \log u$. This is the *C* scale of the so-called 10-in. slide rule. The equations of the *A* and *K* scales on the same slide rule are $x = 5 \log u$ and $x = {}^{10}\!/\!_3 \log u$, respectively.

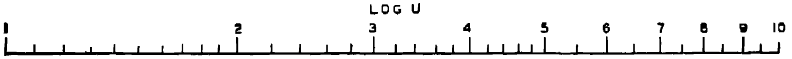


Fig. 29-31*d*.

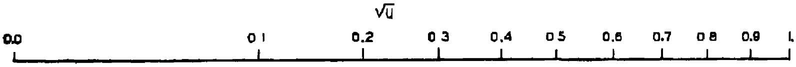


Fig. 29-31*e*

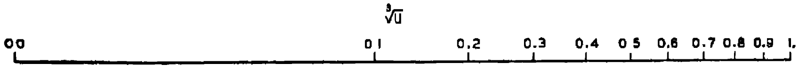


Fig. 29-31*f*.

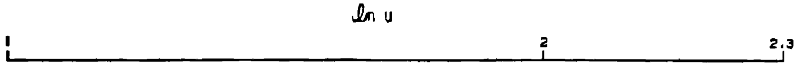


Fig. 29-31*g*.

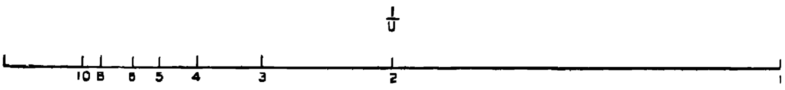


Fig. 29-31*h*.

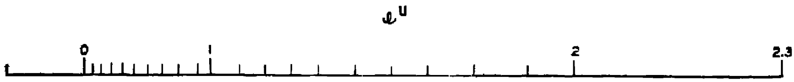


Fig. 29-31*i*.

When a scale of $mf(u)$ is constructed, any scale $kmf(u)$ may be immediately obtained by use of similar triangles. Figure 29-32 shows how a scale of log u with any modulus may be obtained. Figures 29-33, 29-34 show the scales of sin u and log (10 sin u).

Stationary Scales. A relation between two variables u and v of the form $v = f(u)$ can be represented by constructing the two scales $x = mf(u)$ and $x = mv$ on the opposite sides of the same axis. Each scale has the same modulus and is measured from a common fixed point or origin. In some cases it may be convenient to represent the functional relationship $\varphi(v) = f(u)$ directly without transforming the equation to $v = \varphi^{-1}[f(u)]$. It is evident that the adjacent scales are $x = mf(u)$ and $x = m\phi(v)$.

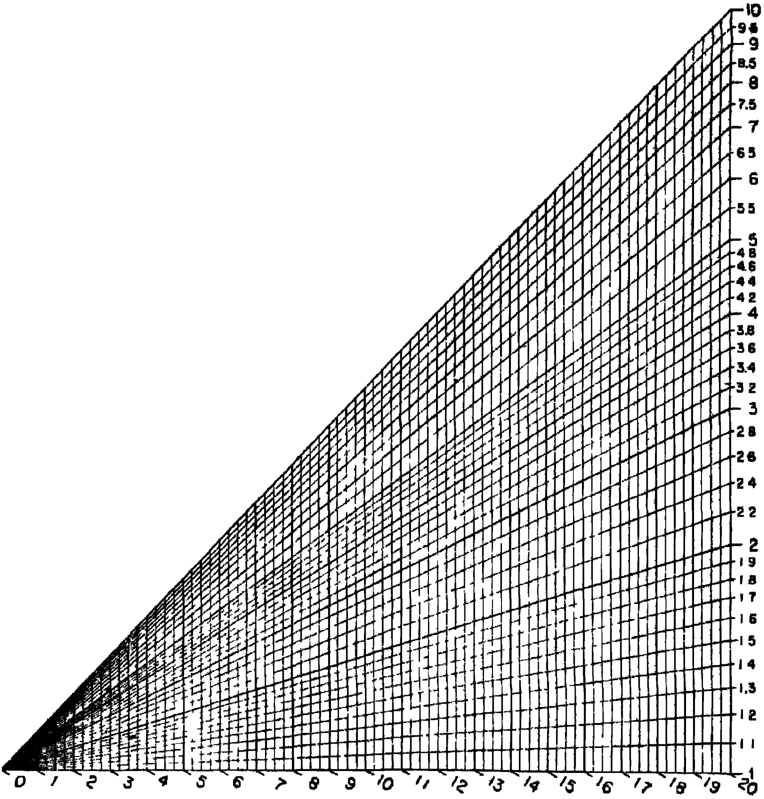


FIG. 29-32. Scale for log u (any modulus).

The scales previously mentioned are called *natural scales*. Usually it is desirable o use logarithmic scales in representing relationships of the form $\phi(v) = f(u)$ or $= f(u)$. The following illustrations show the use of the above scales for the rela- onship $u = \sqrt{v}$.

$$u = \sqrt{v} \qquad \text{and} \qquad mu = m\sqrt{v} \tag{1}$$

Using the scales in Figs. 29-31c, 29-31e, we have the stationary scales shown in ig. 29-35.

$$u^2 = v \qquad \text{and} \qquad mu^2 = mv \tag{2}$$

Using the scales in Figs. 29-31a, 29-31b we have the stationary scales shown in ig 29-36.

$$u = \sqrt{v} \qquad 2 \log u = \log v \qquad \text{and} \qquad m(2 \log u) = m \log v \tag{3}$$

Using the scales of Figs 29-31d or 29-32, we have the stationary scales shown in Fig. 29-37. It is to be noticed that, whenever logarithmic scales are used, neither

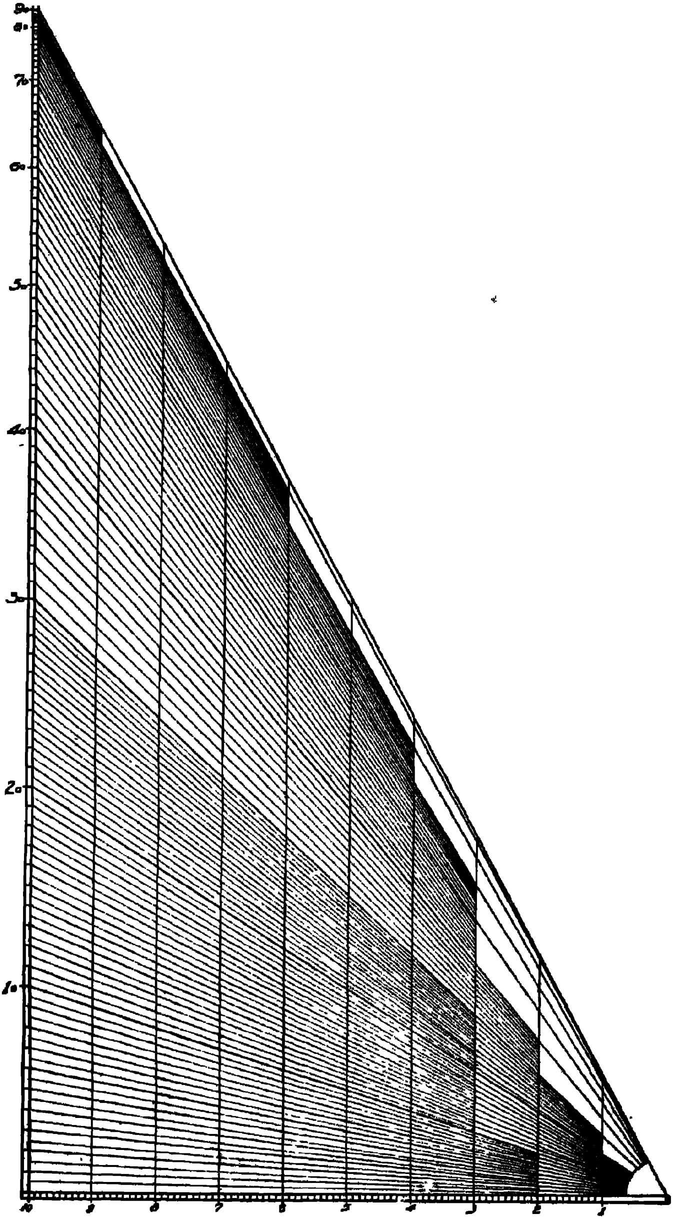


FIG. 29-33. Scale for sin u (any modulus).

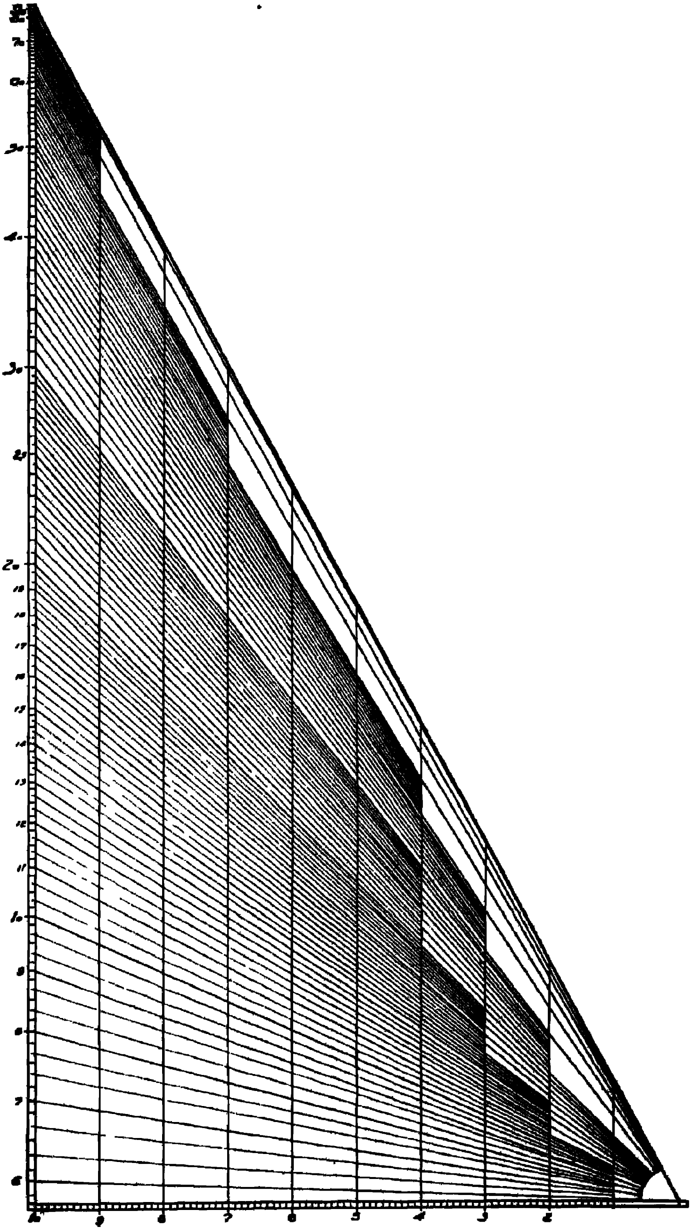


Fig. 29-34. Scale for log (10 sin u) [any modulus].

u nor v can be taken to be zero. However, because of the cyclic character of the logarithmic scale, any number on either scale may be taken as that number multiplied by any positive or negative integral power of 10.

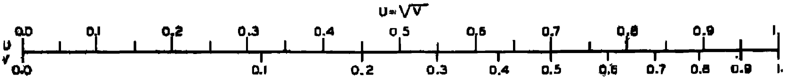


Fig. 29-35.

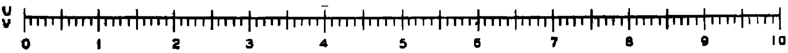


Fig. 29-36.

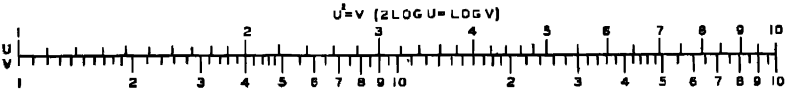


Fig. 29-37.

Adjacent stationary scales may be used to advantage in representing the relationship between the two variables in a conversion formula. Consider the formula

$$I = 0.3937C$$

which converts centimeters to inches and vice versa. Assume that C has values from 0 to 25 and that the length of the scale is to be 10 in. approximately.

Using logarithms,

$$\log I = \log C + \log 0.3937$$

Since the constant term log 0.3937 affects only the position of the scales with respect to each other, it can be neglected in the process of calculating the scale modulus m.

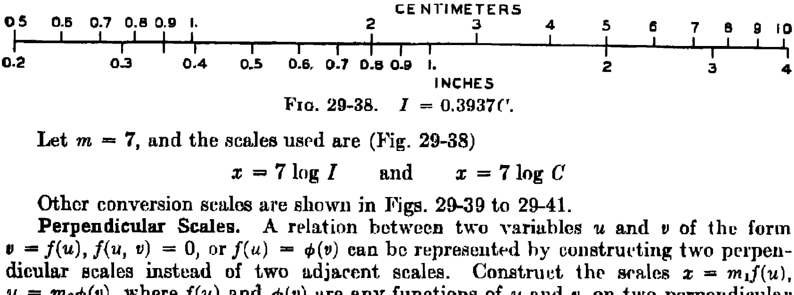


Fig. 29-38. $I = 0.3937C$.

Let $m = 7$, and the scales used are (Fig. 29-38)

$$x = 7 \log I \qquad \text{and} \qquad x = 7 \log C$$

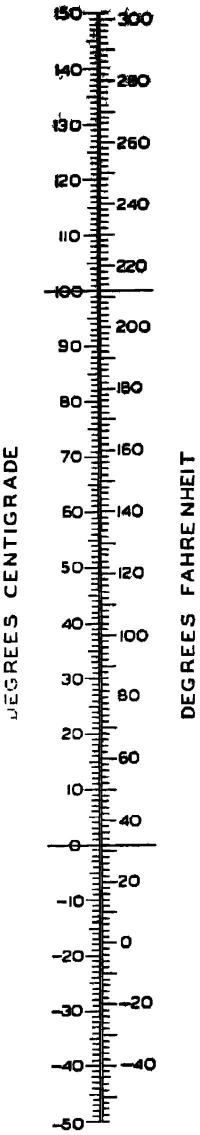
Other conversion scales are shown in Figs. 29-39 to 29-41.

Perpendicular Scales. A relation between two variables u and v of the form $v = f(u)$, $f(u, v) = 0$, or $f(u) = \phi(v)$ can be represented by constructing two perpendicular scales instead of two adjacent scales. Construct the scales $x = m_1 f(u)$, $y = m_2\phi(v)$, where $f(u)$ and $\phi(v)$ are any functions of u and v, on two perpendicular axes, and through the points marked on these scales draw perpendiculars to the axes. Any pair of values of u and v that satisfies the equation $f(u, v) = 0$ will determine a point in the plane of the axes. The locus of all such points is a curve which is said to represent the functional relationship $f(u, v) = 0$. The rectangular or cartesian equation of the curve, referred to the x- and y-axes, is found by solving for u and v from the scale equations and substituting these values in the equation $f(u, v) = 0$.

It is evident that the nature of the locus will vary with the equations of the scales. For many purposes scale equations will be chosen so that the resulting locus will be a straight line.

Various Types of Coordinate Paper. *Rectangular.* The equations of the scales are $x = m_1u$ and $y = m_2v$. With the exception of a linear relationship between u and v, the locus of $f(u, v)$ will be a curve.

The graph of the curve $uv = 12$ is shown in Fig. 29-42.



$C = \frac{5}{9}(F - 32)$

Fig. 29-39.

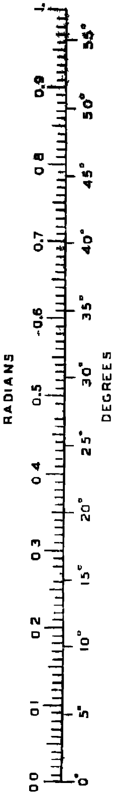


Fig 29-40 Relationship of degrees and radians.

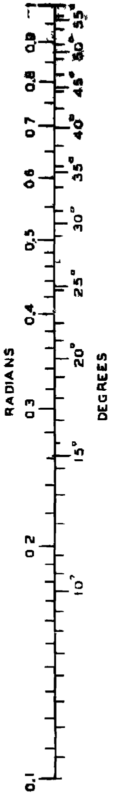


Fig 29-41 Relationship of degrees and radians.

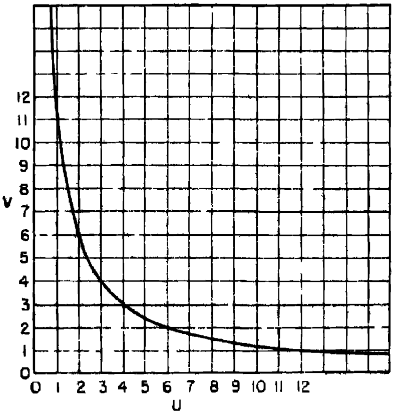


Fig. 29-42. $uv = 12$.

Logarithmic. If the scales $x = m_1 \log u$ and $y = m_2 \log v$ are marked off on the x- and y-axes, respectively and perpendiculars are drawn to both axes, there will result a logarithmic coordinate paper. The prepared logarithmic paper usually has scales of the same modulus. For example, if we are to plot the graph of $u^pv^q = k$, where p, q, and k, are constants, on logarithmic paper, the result will be a straight line. If $u^pv^q = k$, then $p \log u + q \log v = \log k$. Now if the perpendicular scales are $x = m \log u$ and $y = m \log v$, then $(p/m)x + (q/m)y = \log k$, which is a straight line.

EXAMPLE: Sketch the graph of the curve $uv = 12$ on logarithmic coordinate paper Since $\log u + \log v = \log 12$, the equations of the scales are $m \log u$ and $m \log v$ (Fig. 29-43).

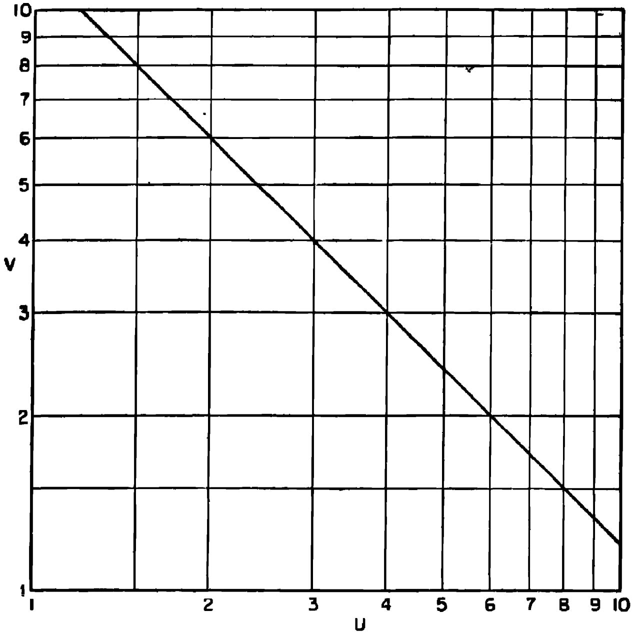


FIG. 29-43. $uv = 12$ ($\log u + \log v = \log 12$).

Semilogarithmic. If $u = ab^v$, where a and b are constants, then

$$\log u = v \log b + \log a$$

Now let $x = m_1v$ and $y = m_2 \log u$ be perpendicular scales. The graph of the above relationship will then be a straight line whose cartesian equation is

$$y/m_2 = \left(\frac{\log b}{m_1}\right) x + \log$$

Coordinate paper with scales of this kind is called *semilogarithmic.*

Example: If $v = 3\cdot 2u$, then $\log v = u \log 2 + \log 3$. Figure 29-44 shows the graph when plotted on two-cycle semilogarithmic paper. Figure 29-45 shows the graph on a single-cycle paper.

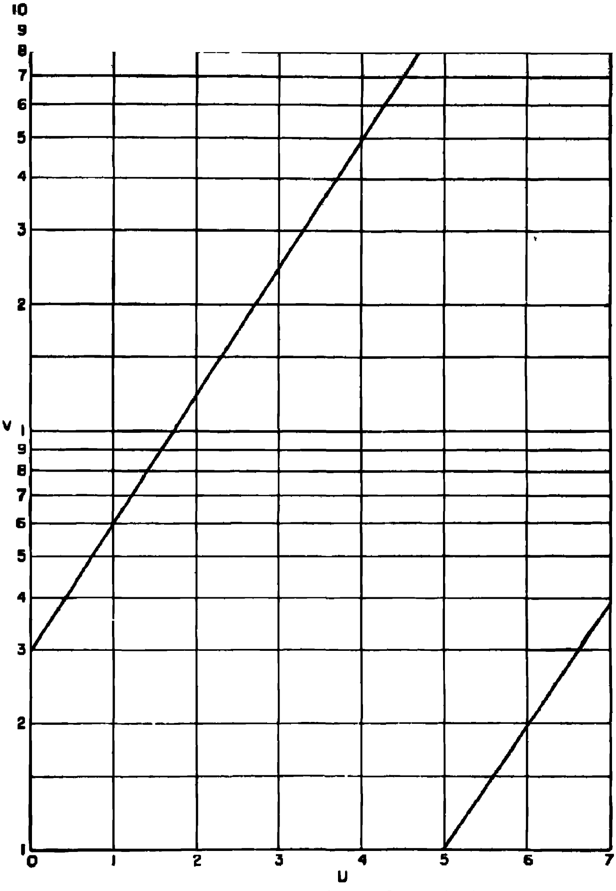


Fig. 29-44. $v = 3\cdot 2^u$ (two-cycle semilogarithmic paper).

Reciprocal. Equations of the type $v = u/(au + b)$ may be represented by a straight line on a coordinate paper whose perpendicular axes carry reciprocal scales. The equation $v = u/(au + b)$ may be written in the form $1/v = (b/u) + a$. If the x- and y-axes carry the scales $x = m(1/u)$ and $y = m(1/v)$, respectively, then the graph will be the straight line

$$y = bx + am$$

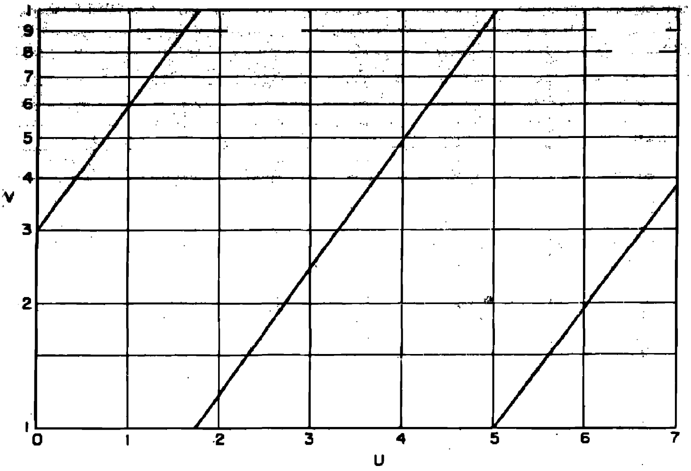


FIG. 29-45. $v = 3 \cdot 2^u$ (single-cycle semilogarithmic paper).

EXAMPLE: Sketch the equation $3u + 2v = uv$ on reciprocal coordinate paper. The equation may be written

$$\frac{2}{u} + \frac{3}{v} = 1$$

The graph is a straight line (Fig. 29-46).

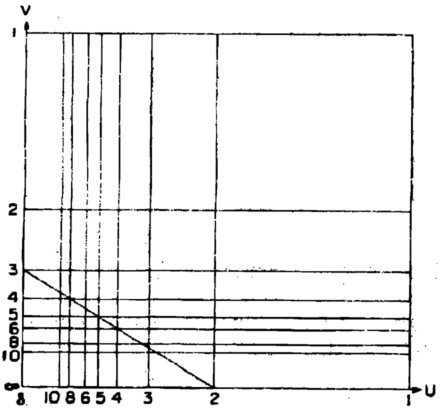


FIG. 29-46. $\frac{2}{u} + \frac{3}{v} = 1.$

Semisine. One axis on semisine paper carries a sine scale, the other a uniform scale. Figure 29-46*a* shows the plot of the equation 20 sin $v = u + 10$ on this type of paper.

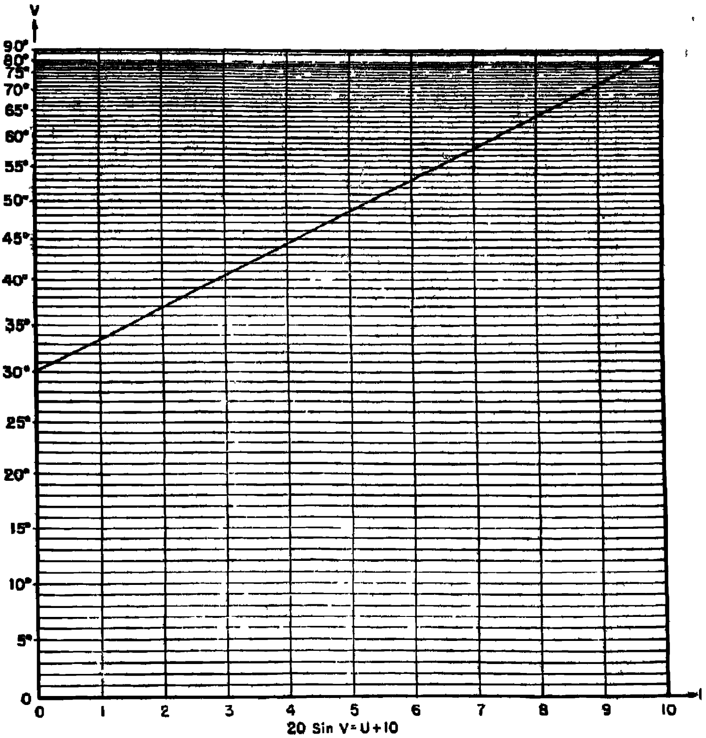


Fig. 29-46*a*.

Representation of a Relation between Three Variables. An equation in three variables of the form $f(u, v, w) = 0$ can be represented graphically by extending the methods used for graphing a two-variable relationship $\phi(u, v) = 0$. If one of the variables, say w, is held constant, then $\phi(u, v, c) = 0$, and the graph for each value of c held constant will be a curve or a straight line depending on the scales used.

To illustrate some of the possibilities, graph the relationship

$$\left(1 - \frac{S}{100}\right) = D$$

1. If uniform scales are used for S and P, D is represented by a family of curves (Fig. 29-47a).

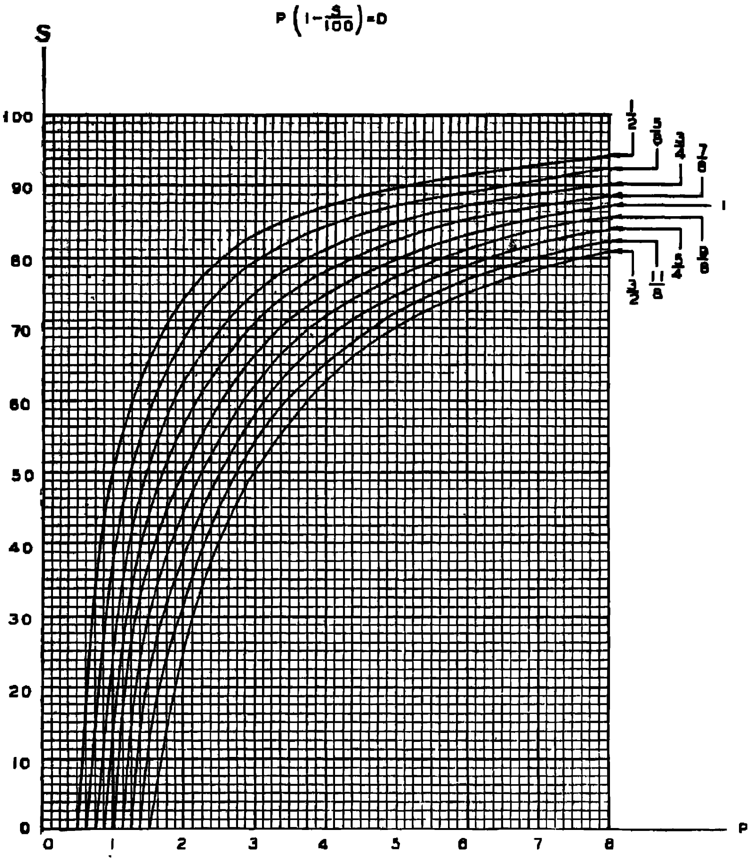


Fig. 29-47a.

2. If logarithmic scales are used for S and P, D is represented by a family of parallel straight lines (Fig. 29-47*b*).

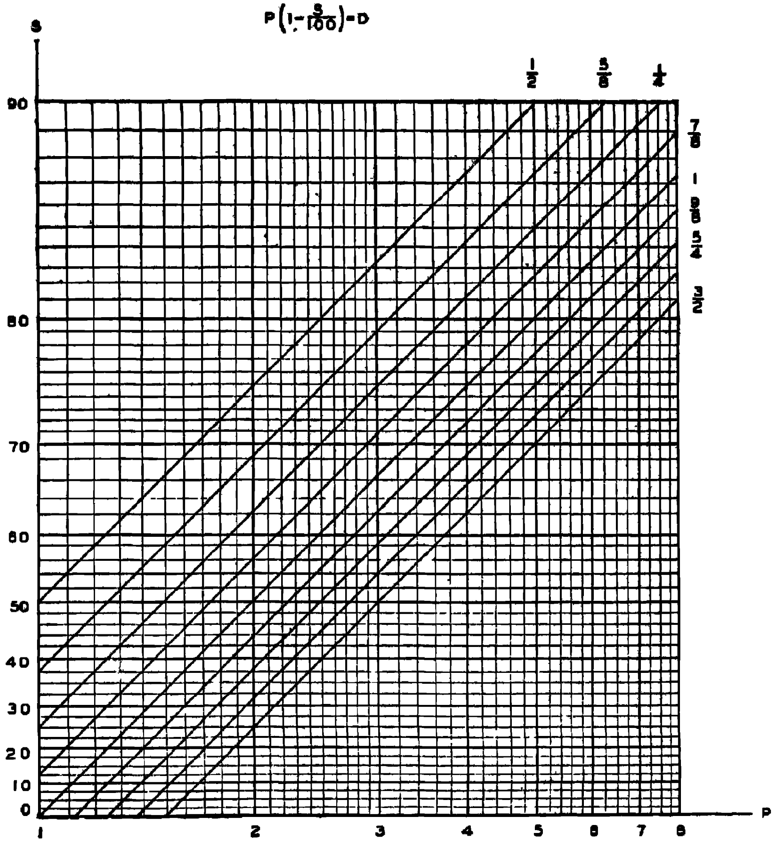


Fig. 29-47*b*.

3. If uniform scales are used for P and D, S is represented by a family of radiating straight lines (Fig. 29-47c).

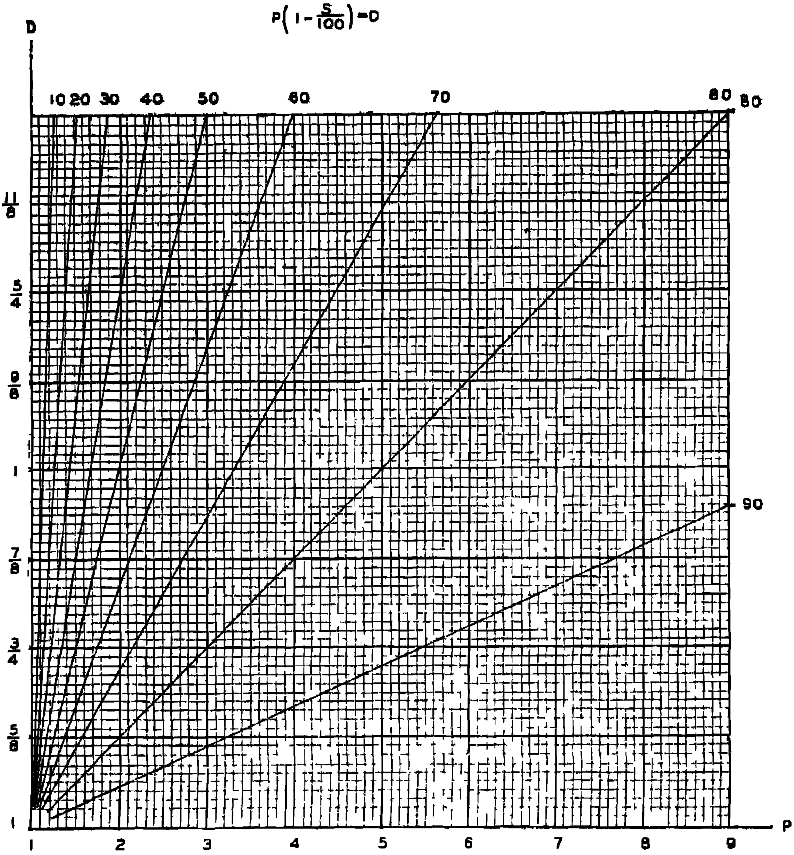


FIG 29-47c

Figure 29-48 shows corresponding graphs for the equation

$$R = 0.003CS^2$$

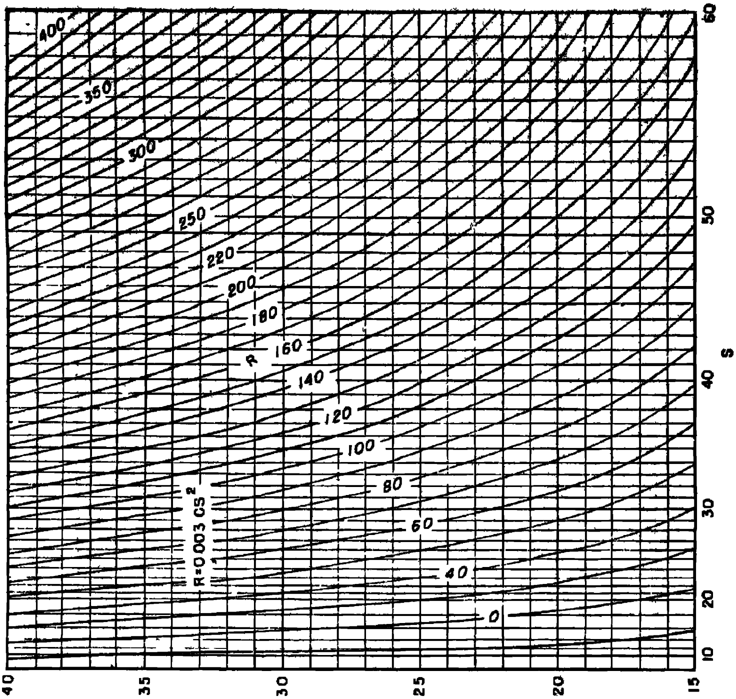


FIG. 29-48b.

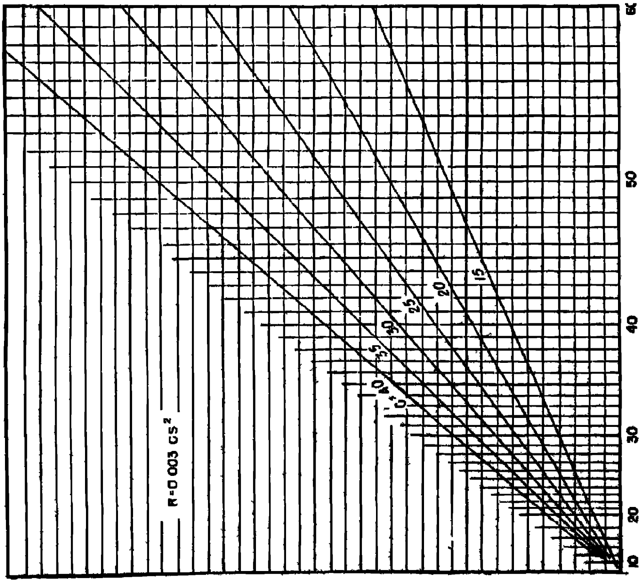


FIG. 29-4

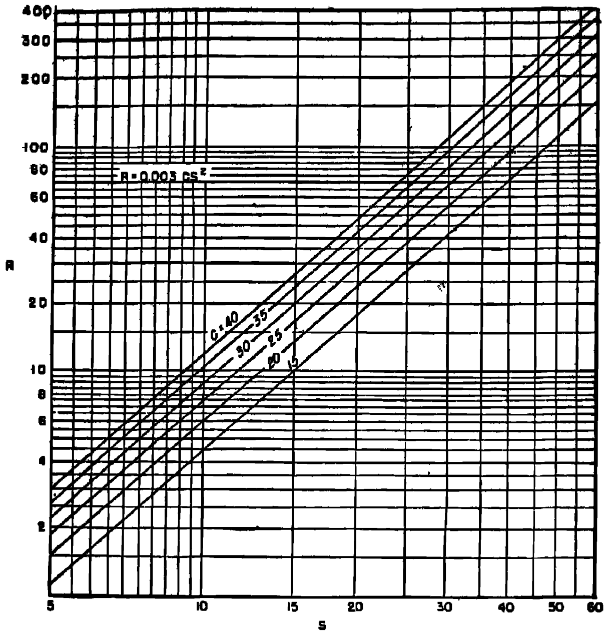
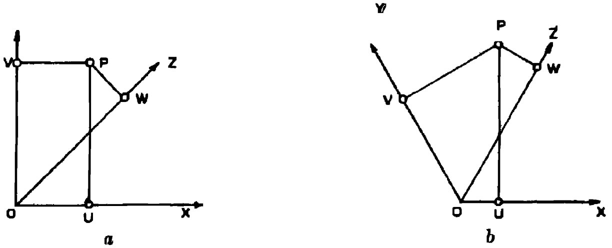


FIG. 29-48c.



a *b*

FIG. 29-49. Hexagonal chart.

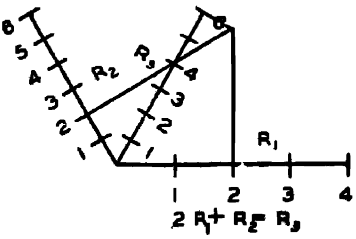


FIG. 29-50a.

Hexagonal Chart. This is a useful type of chart for the relation

$$f_1(u) + f_2(v) = f_3(w)$$

In Fig. 29-49*a* let OX and OY be perpendicular axes, and let OZ be the bisector of the angle between them. Lay off the scales $mf_1(u)$, $mf_2(v)$, and $(m/\sqrt{2})f_3(w)$ on the X-, Y-, and Z-axes, respectively. From any point P, draw perpendiculars to OX, OY, and OZ. The three points of intersection will give values of u, v, and w, satisfying the equation $f_1(u) + f_2(v) = f_3(w)$.

If the angle between OX and OY is 120 deg and OZ is the bisector of the angle, the modulus of each of the three scales will be the same (Fig 29-49*b*).

For examples see Figs. 29-50*a*, 29-50*b*.

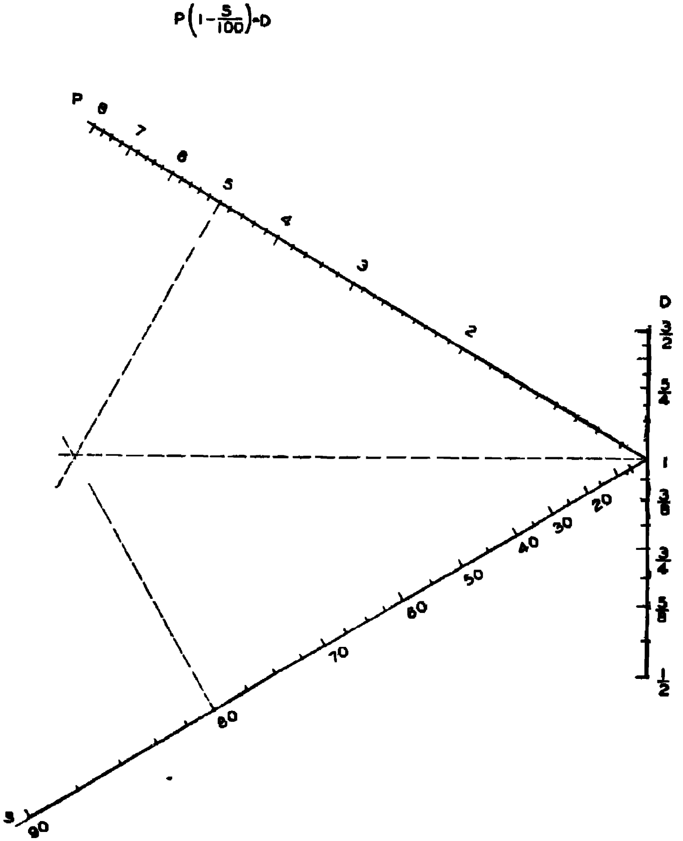


FIG. 29-50*b*.

QUADRATIC, CUBIC, AND BIQUADRATIC EQUATIONS

Quadratic Equation. Every quadratic equation in one variable can be put in the form

$$x^2 + Ax + B = 0 \tag{1}$$

This equation may be written

$$B = -Ax - x^2$$

If A and B are taken to be the coordinates of a point in a plane, the equation will represent a straight line for each real value of x. These lines are drawn in Fig. 29-51.

The complex roots of the equation may also be found graphically. Since the solution of Eq. (1) is known to be

$$x = -\tfrac{1}{2}A \pm \sqrt{\tfrac{1}{4}A^2 - B}$$

we may replace the quantity under the radical by $-k^2$; *i.e.*,

$$\tfrac{1}{4}A^2 - B = -k^2$$

or

$$B = \tfrac{1}{2}A^2 + k^2 \qquad (2)$$

where k is the coefficient of i ($i = \sqrt{-1}$) in the solution.

Equation (2) represents a family of parabolas for different values of k^2, where k^2 is the intercept on the b-axis (see Fig. 29-51).

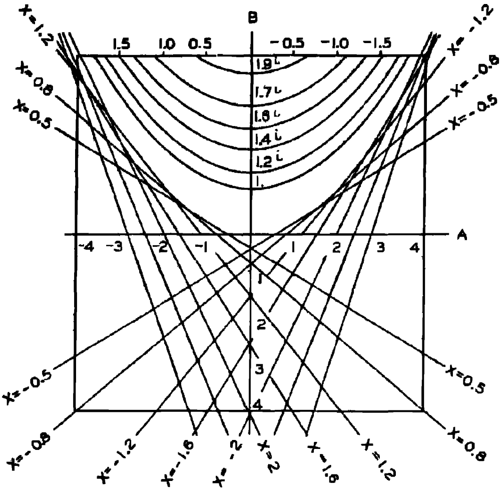


Fig. 29-51. $x^2 + Ax + B = 0$.

Cubic Equation. Every cubic equation can be written in the form

$$Y^3 + aY^2 + bY + c = 0 \qquad (1)$$

By making the substitution

$$Y = x - (a/3)$$

Eq. (1) may be written

$$x^3 + Ax + B = 0 \qquad (2)$$

where $A = b - \tfrac{1}{3}a^2$

$B = c - \tfrac{1}{3}ab + \tfrac{2}{27}{}^3$

Equation (2) may be written in the form

$$B = -Ax - x^3$$

which is the equation of a straight line for each real value of x plotted in the AB plane. The straight lines are shown in Fig. 29-52.

If Eq. (2) has complex roots, the three roots are of the form $2a$, $-a + ib$, and $-a - ib$, where $i = \sqrt{-1}$.

From Eq. (2)

$$A = b^2 - 3a^2 \qquad (3)$$

$$B = -2a^3 - 2ab^2 \qquad (4)$$

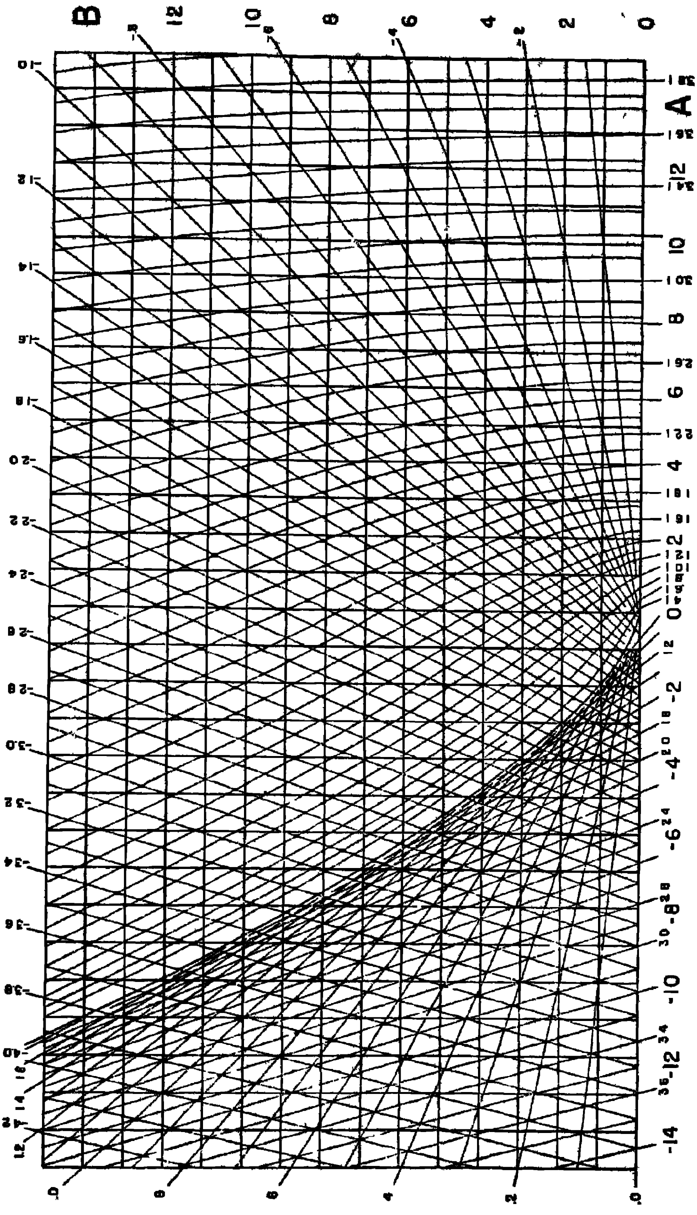


FIG. 29-52. $Y^3 + aY^2 + bY + c = 0$.

and from Eq. (3)

$$a = \pm\sqrt{\frac{b^2 - A}{3}} \tag{5}$$

Substituting the value of a in Eq. (5) into Eq. (4), we have

$$B^2 = -\tfrac{4}{27}(A - b^2)(A - 4b^2)^2 \tag{6}$$

Note that when $b = 0$, Eq. (6) becomes

$$B^2 = -\tfrac{4}{27}A^3$$

This curve forms the boundary between the region of all real roots and the region of complex roots. The completed chart is shown in Fig. 29-52.

Example 1: Find the roots of the cubic equation

$$x^3 - 13x + 12 = 0$$

$A = -13$, $B = 12$; then $x = 1, -4$, and 3.

Example 2: Solve the equation

$$x^3 - 15x - 126 = 0$$

Here $A = -15$ and $B = -126$. Since only positive values of B from 0 to 15 are on the chart, let $x = -3z$, for example. Then the equation becomes

$$z^3 - 1.67z + 4.7 = 0$$

From Fig. 29-52 we read

$$z = -2,\ 1 \pm 1.13i$$

therefore $x = 6,\ -3 \pm 3.39i$.

Example 3: Solve the equation

$$y^3 - 3y^2 + 9y + 13 = 0$$

Let $y = x + 1$. Then the resulting equation is

$$x^3 + 6x + 20 = 0$$

As in Example 2, $B = 20$ is not on the chart. Now let $x = 2z$. Then the equation becomes

$$z^3 + 1.5z + 2.5 = 0$$

Hence $A = 1.5$ and $B = 2.5$.
From the chart we read

$$z = -1,\ 0.5 \pm 1.5i$$

and $x = -2,\ 1 \pm 3i$

Therefore $y = -1,\ 2 \pm 3i$

Biquadratic Equation. The equation of the biquadratic is

$$x^4 + Ax^3 + Bx^2 + Cx + D = 0 \tag{1}$$

This equation may be written as

$$(x^2 + ax + b)(x^2 + cx + d) = 0$$

where $A = a + c$
$B = b + d + ac$
$C = ad + bc$
$D = bd$

Let $K = ac$ and $L = b + d$; then $B = L + K$.
From the equations $L = b + d$ and $D = bd$, we obtain

$$b = \tfrac{1}{2}L + \tfrac{1}{2}\sqrt{L^2 - 4D}$$

and

$$d = \tfrac{1}{2}L - \tfrac{1}{2}\sqrt{L^2 - 4D}$$

From the equations $A = a + c$ and $K = ac$, we obtain

$$a = \tfrac{1}{2}A + \tfrac{1}{2}\sqrt{A^2 - 4K}$$

and

$$c = \tfrac{1}{2}A - \tfrac{1}{2}\sqrt{A^2 - 4K}$$

Substituting the values of a, b, c, and d in the equation $C = ad + bc$, together with the relation $K = B - L$, we obtain the cubic equation

$$L^3 - BL^2 + (AC - 4D)L + 4BD - C^2 - A^2D \quad 0 \qquad (2)$$

By letting $L = Z + \frac{1}{3}B$, the last equation becomes

$$Z^3 + (AC - 4D - \tfrac{1}{3}B^2)Z + \tfrac{1}{3}(ABC + 8BD - \tfrac{2}{9}B^3 - 3C - 3A^2D) = 0$$

It is to be noted that the coefficients of cubic Eq. (2) are obtained directly from the coefficients of the original biquadratic Eq. (1).

The values of Z are obtained from the cubic chart (Fig. 29-52), and $b + d$ is obtained from the relation $L = b + d = Z + \frac{1}{3}B$. The numerically greatest of the real values of $Z + \frac{1}{3}B$ will always give real values of b and d.

Since $b + d = L$ and $bd = D$, then b and d are roots of the equation

$$u^2 - Lu + D = 0$$

and are found from the quadratic chart (Fig. 29-51).

Any method for obtaining the exact values of the roots of a biquadratic equation is laborious. However, making use of the resultant cubic and its chart reduces the time element to a minimum. The following example will illustrate the process of transforming an equation.

Example: Solve for the roots of the equation

$$6x^2 - 8x - 3 \quad 0$$

Here $A = 0$, $B = -6$, $C = -8$, and $D = -3$. Therefore $Z = 0$, and $L = -2$. Then $u^2 + 2u - 3 = 0$, and $u = 3, -1$; *i.e.*, $b = -3$, and $d = -1$. $k = -6 + 2 = -4$, and $ac = -4$; $a + c = 0$. Therefore $a = \pm 2$, and $c = \mp 2$. The biquadratic reduces to

$$(x^2 + 2x - 3)(x^2 - 2x + 1) = 0$$

or

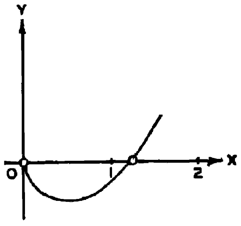
$$(x^2 - 2x - 3)(x^2 + 2x + 1) = 0$$

Hence $x = 1, -3, 1, 1$; or $x = -1, 3, -1, -1$. The first set of values does not satisfy but the second set does. Therefore $x = -1, -1, -1, 3$.

SOLUTION OF EQUATIONS CONTAINING A SINGLE VARIABLE

It is often necessary to solve an equation of the form $f(x) = 0$, where the exact solution cannot be found by any known method. An approximate solution may be found by one of several methods. Three of these methods are as follows:

1. Sketch the curve $y = f(x)$, and estimate where the curve crosses the x-axis.



$e^x - 2x - 1 = 0$
FIG. 29-53.

Example: Solve the equation

$$e^x - 2x - 1 = 0$$

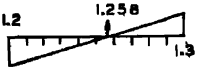
Sketch the curve $y = e^x - 2x - 1$ from the following table of values (Fig 29-53).

x	y	x	y	x	y
0	0	1.1	−0.2	1.21	−0.07
1	−0.28	1.2	−0.08	1.25	−0.01
2	+2.30	1.3	+0.07	1.26	+0.005

The first table shows that the value of x is between $x = 1$ and $x = 2$.

The second table shows that the value of x is between $x = 1.2$ and $x = 1.3$.

Now magnify the scale between $x = 1.2$ and $x = 1.3$, and draw a line from point (1.2, −0.08) to point (1.3, 0.07). The point where this line crosses the x-axis is a good approximation. The value is approximately 1.258 (Fig. 29-54).



$e^x - 2x - 1 = 0$
FIG. 29-54.

2. *a*. If scales are available, divide the function $f(x)$ into two parts, *e.g.*, $f_1(x) = f_2(x)$, and lay off stationary scales for $f_1(x)$ and $f_2(x)$. Locate on these scales where the markings opposite each other have the same value (Fig. 29-55).

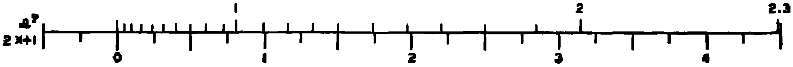


FIG. 29-55. $e^x - 2x - 1 = 0$.

EXAMPLE: Solve $e^x = 2x + 1$.

b. Instead of stationary adjacent scales, use perpendicular scales. This amounts to sketching each of the curves $y = e^x$ and $y = 2x + 1$ on the same diagram. At the point or points of intersection drop an ordinate to the x-axis. This value of x will be an approximate answer (Fig. 29-56).

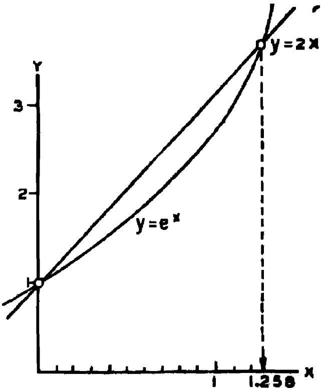


FIG. 29-56. $e^x - 2x - 1 = 0$.

3. Lay off an e^y scale on the y-axis and a uniform scale on the x-axis. The sketch of the equation $e^y = 2x + 1$ will be a straight line.

If perpendicular axes are drawn on a sheet of transparent paper and their intersection is moved along the straight line, always keeping the respective axes parallel, until the numbers at both feet of the moving axes are the same, this number will be the required answer (Fig. 29-57).

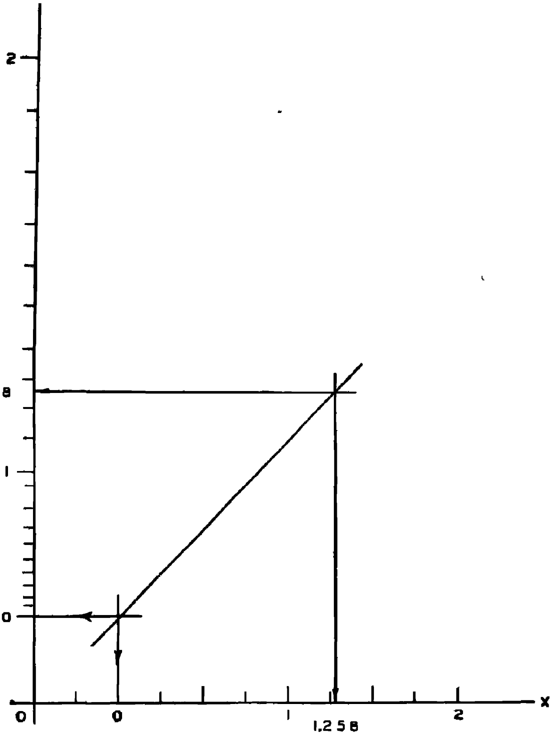
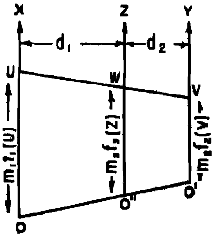


FIG. 29-57. $e^x - 2x - 1 = 0$.

NOMOGRAPHIC OR ALIGNMENT CHARTS

Types. The fundamental principles involved in the construction of a nomographic, or alignment, chart consists of the representation of the relationship between three variables of the form $f(u, v, w) = 0$ by means of three scales along three axes (straight or curved) in such a way that a straight line cuts the three scales in values of u, v, and w, satisfying the equation $f(u, v, w) = 0$. Some of the most common methods of constructing charts are described below. In the notation used, $f_1(u)$ is any function of u alone, $f_2(v)$ is any function of v alone, etc.



$f_1(u) + f_2(v) = f_3(w)$
FIG. 29-58.

1. $f_1(u) + f_2(v) = f_3(w)$

a. Draw two parallel lines OX and $O'Y$ any distance apart, and on these axes construct the scales $x = m_1f_1(u)$ and $y = m_2f_2(v)$, measured from convenient points 0 and 0′, respectively, where m_1 and m_2 are convenient moduli.

b. Draw a third line $O''Z$ parallel to OX and $O'Y$ such that

$$\frac{\text{Distance } (d_1) \text{ from } OX \text{ to } O''Z}{\text{Distance } (d_2) \text{ from } O''Z \text{ to } O'Y} = \frac{m_1}{m_2}$$

c. Determine a starting point for the graduations of the w scale. It is convenient to draw a line from a value u_0 on the u scale to a value of v_0 on the v scale. This line will cut the w scale at a value w_0, which is computed from the equation

$$f_1(u_0) + f_2(v_0) = f_3(w_0)$$

Now take any other pair of values of u and v, and on the w scale mark the corresponding value of w, say w_1.

d. The equation of the w scale is

$$z = \frac{m_1 m_2}{m_1 + m_2} f_3(w)$$

e. Move the z scale in (*d*) along the z-axis until the number w_0 on the scale is opposite the number w_0 already calculated by (*c*). A check on the work up to now reveals that w_1 on the scale is opposite the number w_1 already calculated by (*c*).

f. If we have a scale of $f(u)$ constructed as in Figs. 29-31*a* to 29-31*k*, it is easy to make the check suggested in (*e*) by using compasses to take off measurements.

EXAMPLE: Construct a nomogram for the equation $2U + W = 10 \log V$. Let the variables range as follows: V from 1 to 10, U from 0 to 5, and W from 0 to 10.

If it is desired to have the W scale in the center, rewrite the equation as

$$10 \log V - 2U = W$$

The scales are $m_1(10 \log V)$, $m_2(2U)$, and $m_3(W)$ where $m_3 = m_1m_2/(m_1 + m_2)$. The V and U scales will be constructed in opposite directions because of the negative sign. The length of the chart is to be 10 in. Therefore, $m_1 = 1$, $m_2 = 1$, and $m_3 = \frac{1}{2}$. The scales to be constructed are $10 \log V$, $2U$, and $\frac{1}{2}W$. Since $(d_1/d_2) = (m_1/m_2) = 1$, $d_1 = d_2$, and the W scale is midway between. When $U = 0$ and $V = 1$, $W = 0$; and when $U = 0$ and $V = 10$, $W = 10$. The completed chart is shown in Fig. 29-59. When $V = 8$ and $W = 3$, $U = 3.01$.

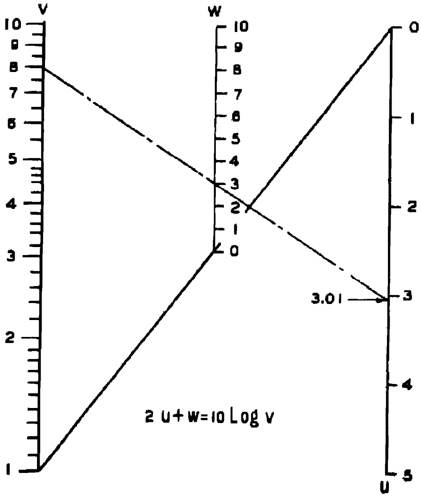


FIG. 29-59.

A chart for the equation of net radiation $q/A = \sigma(T_1^4 - T_2^4)$ is shown in Fig. 29-59*a*.

2. $$f_1(u) \cdot f_2(v) = f_3(w)$$

Any equation in this form can be put into the form of (1) by taking logarithms of both sides. Thus

$$\log f_1(u) + \log f_2(v) = \log f_3(w)$$

This equation then takes the form

$$F_1(u) + F_2(v) = F_3(w,$$

and the method of (1) applies.

Example: Construct a nomogram for the equation

$$E = \pi B \sin^2 \theta$$

where E is illumination in lumens per square centimeter, B is brightness in candles per square centimeter, and θ is the solid angle subtended by the source.

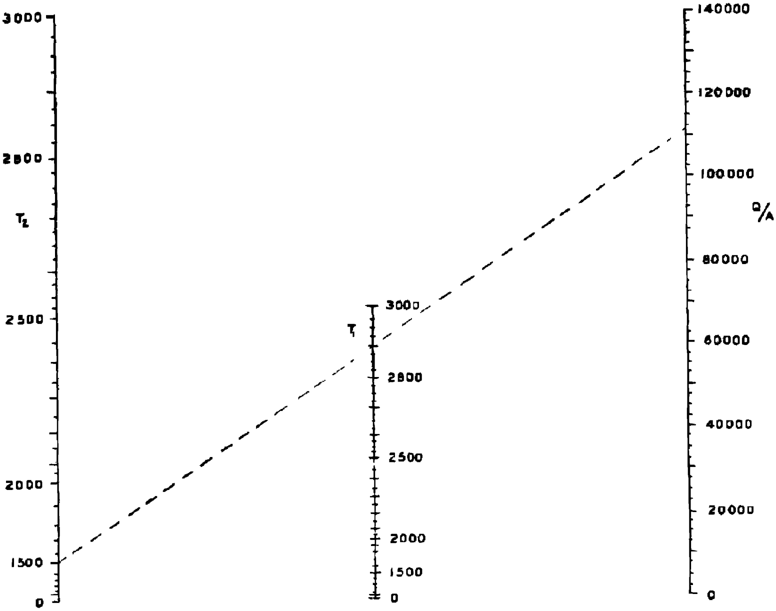


FIG. 29-59a. $b/A = \sigma(T_1^4 - T_2^4)$.

Taking logarithms of both sides of the equation, we have

$$\log E = \log B + 2 \log \sin \theta + \log \pi$$

The ranges of the variables are B from 1 to 100, θ from 1 to 16 deg, and E from 0.01 to 20. It is desired to have the B and θ scales each of length 8 in. The scales are $m_1 \log B$, m_2 $(2 \log \sin \theta)$, and $m_3 \log E$.

Since $\log 100 = 2$ and $\log 1 = 0$, $m_1 = 4$. Since $\log \sin 16$ deg $= -0.5597$ and $\log \sin 1$ deg $= -1.7581$,

$$m_2(-1.1194 + 3.5162) = 8 \qquad \text{or } m_2 = 3.34$$

$(d_1/d_2) = (m_1/m_2) = 4/3.34$. Let the distance between the B- and θ-axes

$$(d_1 + d_2) = 7.34 \text{ in.},$$

then $d_1 = 4$ in., and $d_2 = 3.34$ in. The equations of the scales are $4 \log B$, $6.68 \log \sin \theta$, and $[4(3.34)/(4 + 3.34)] \log E = 1.82 \log E$. The completed chart is shown in Fig. 29-60.

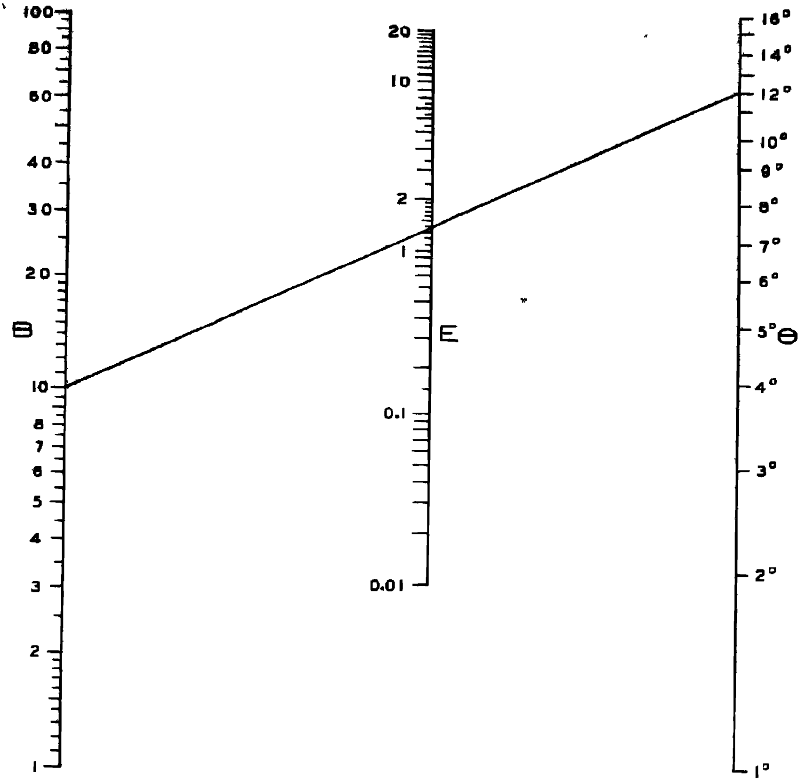


FIG. 29-60. $E = \pi B \sin^2 \theta$.

EXAMPLE: Construct a nomogram for the equation

$$P\left(1 - \frac{S}{100}\right) = D$$

where D varies in $\frac{1}{8}$ units from $\frac{3}{2}$ to $\frac{1}{2}$, P varies from 1 to 8, and S varies from 10 to 90. Taking logarithms of both members of the equation we have

$$\log D = \log P + \log\left(1 - \frac{S}{100}\right)$$

The completed chart is shown in Fig. 29-61.

3. $$[f_1(u)]^{f_2(v)} = f_3(w)$$

Taking logarithms of both sides, the equation then takes the form

$$f_2(v) \log f_1(u) = \log f_3(w)$$

Taking logarithms again of both sides, the equation becomes

$$\log f_2(v) + \log \log f_1(u) = \log \log f_3(w)$$

which, again, is of the form

$$F_1(u) + F_2(v) = F_3(w)$$

4. $$f_1(u) + f_2(v) + f_3(w) + \cdots = f_n(t)$$

or

$$f_1(u) \cdot f_2(v) \cdot f_3(w) \cdots = f_n(t)$$

This equation is merely an extension of (1), and the method of charting is an extension of the method employed in charting (1). For example, consider the case of four variables in an equation of the form $f_1(u) + f_2(v) + f_3(w) = f_4(t)$.

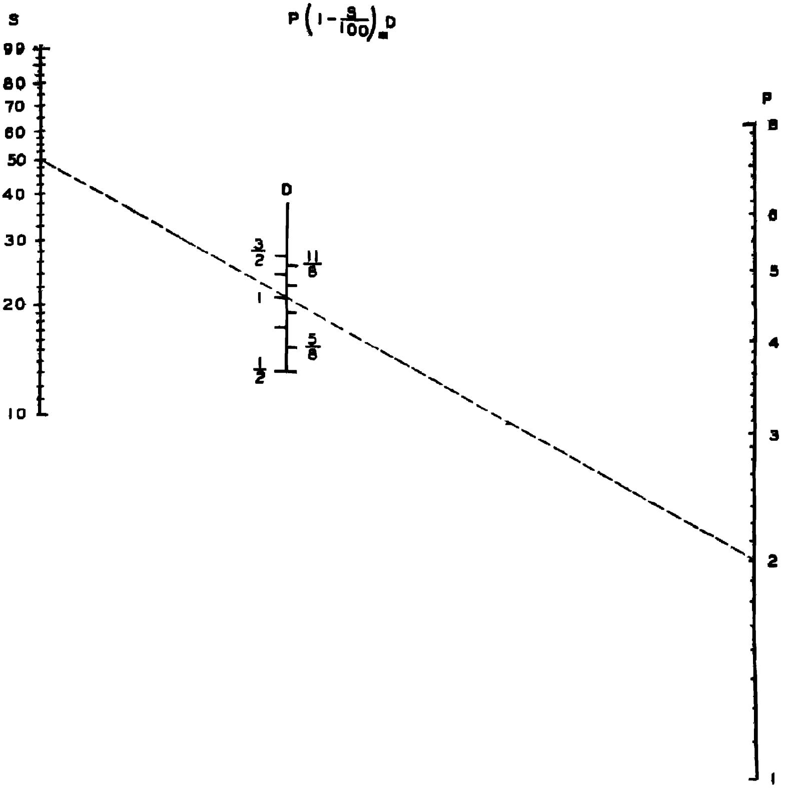


Fig. 29-61

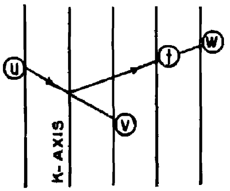


Fig. 29-62

$f_1(u) + f_2(v) + f_3(w) = f_4(t)$.

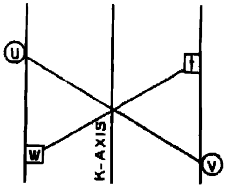


Fig. 29-63.

$f_1(u) + f_2(v) + f_3(w) = f_4(t)$.

Let $f_1(u) + f_2(v) = k$, then $k + f_3(w) = f_4(t)$. Each of these equations is in the form of (1), and each can be charted by means of three parallel scales, but the k scale need not be graduated (Figs 29-62, 29-63).

The first type is illustrated by the following charts (Fig. 29-64):

$$W = 12PA^4/(64 \times 29 \times 10^6T^3)$$

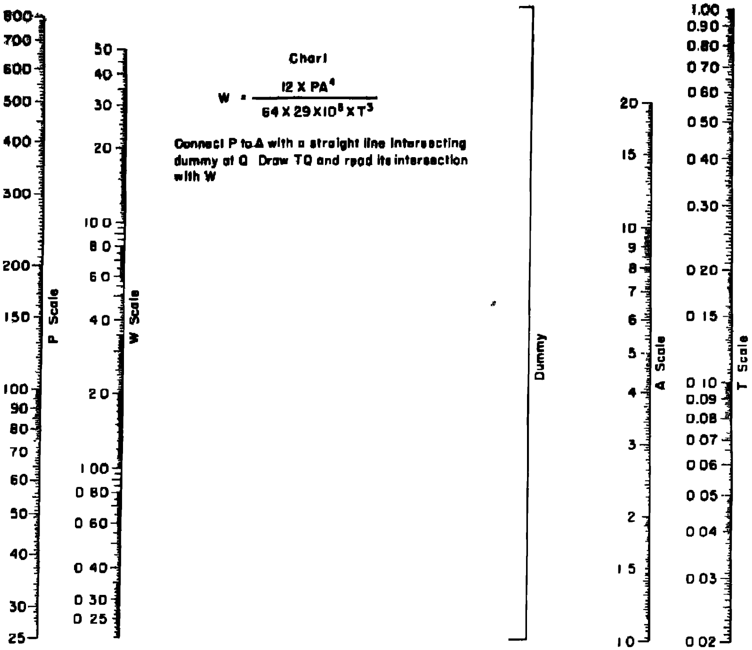


FIG. 29-64.

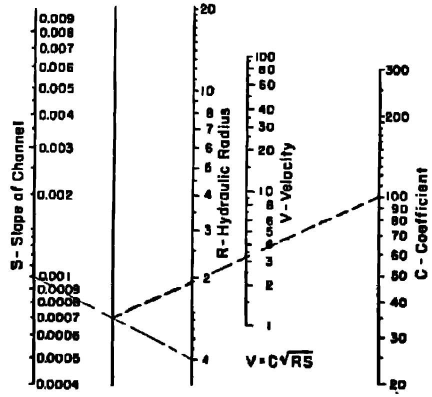


FIG. 29-65.

Chezy formula for the velocity of flow of water, $V = C\sqrt{rs}$ (Fig. 29-65),

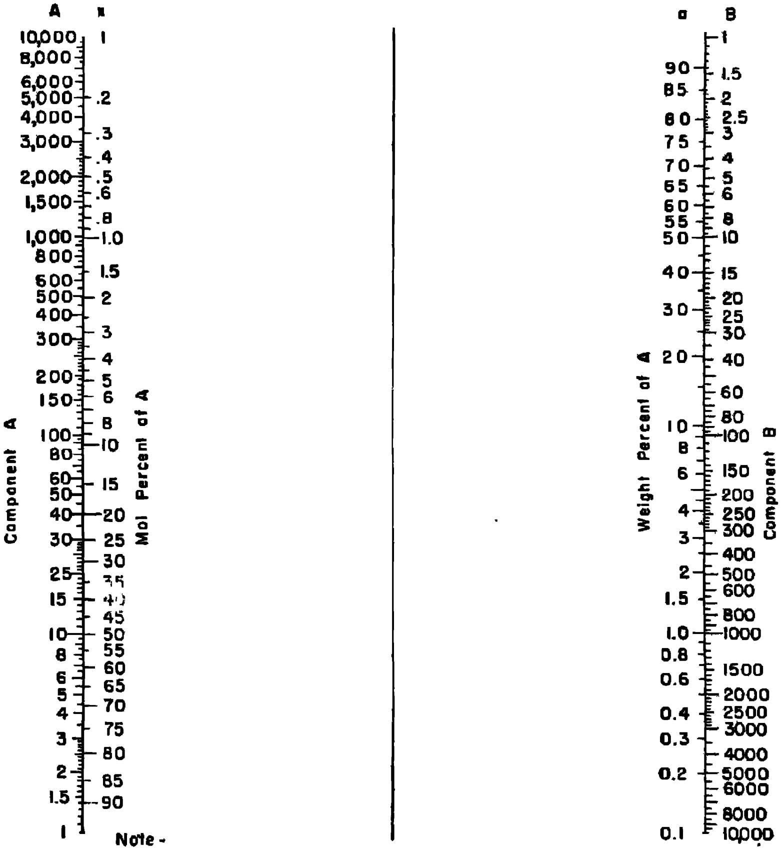
Nomogram on mol per cent—weight per cent (Fig. 29-66).

The second type is illustrated by charts for solution of triangles by the law of tangents (Fig. 29-67), draft correction for salt and fresh-water-going vessels (Fig. 29-68), and the equations $(u_1/u_2) = (v_1/v_2)$ (Fig. 29-69*a*) and $x = v_{0t} \cos \beta$ (Fig. 29-69*b*).

$$5. \qquad f_1(u) - f_2(v) = f_3(w) - f_4(t) \qquad \text{or} \qquad f_1(u)/f_2(v) = f_3(w)/f_4(t)$$

The first equation is obviously of the form (1), and the second equation is immediately transformed into the first form by taking logarithms of both members of the equation. However, sometimes it is convenient to apply the following scheme:

If the range of variables is such that we can use the modulus m_1 for each of the terms $f_1(u)$ and $f_2(v)$ and m_2 for each of the terms $f_3(w)$ and $f_4(t)$, lay off the scales $x = m_1f_1(u)$, $y = m_1f_2(v)$, $z = m_2f_3(w)$, and $q = m_2f_4(t)$, as indicated in Fig. 29-70. To read such a chart, join given values of u and v, and through a given value of w, draw a parallel line cutting the q scale at the value of t which satisfies the original equation.



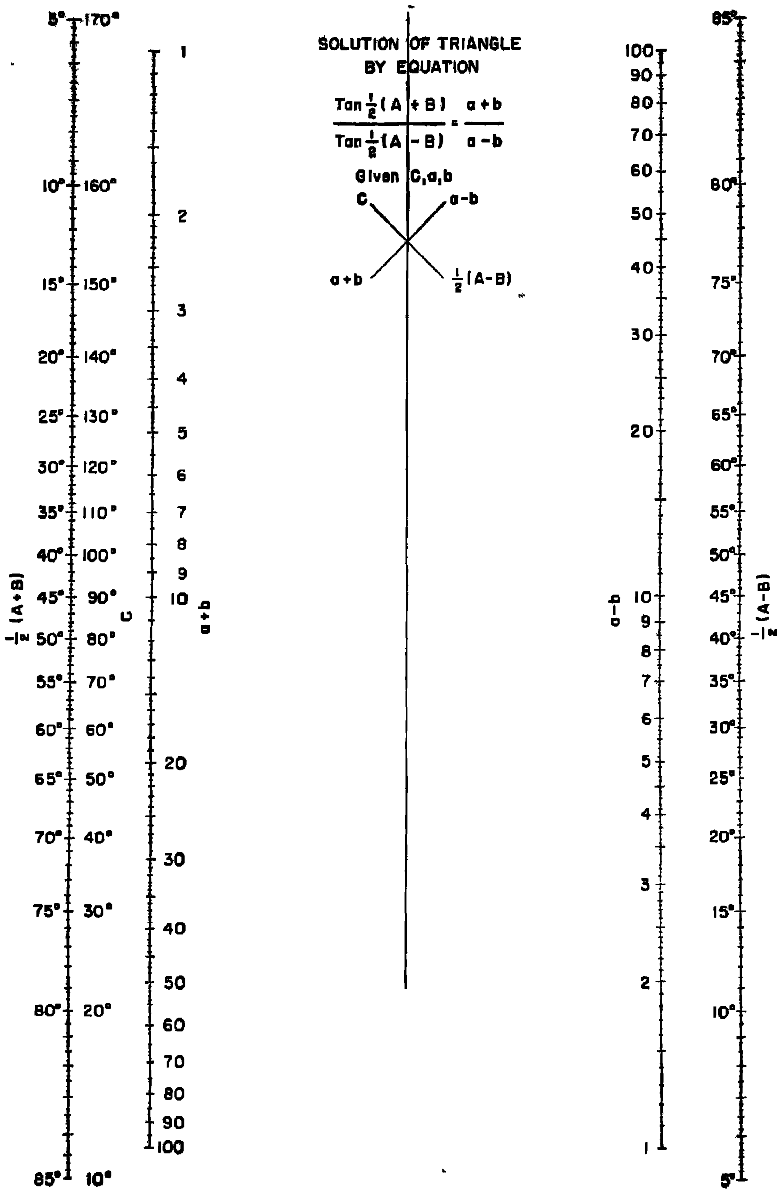
Note -
A line drawn between molecular weights of A and B will give a turning point on the center line. Any other line drawn through this point will connect weight and mol percents of component A

Relation $\frac{A}{B} = \frac{\frac{100-x}{x}}{\frac{100-a}{a}}$

FIG. 29-66.

Figure 29-71 shows a chart for the relation $u_1v_1 = u_2v_2$.

6. $$f_1(u) + f_2(v) = f_3(w)/f_4(t)$$


SOLUTION OF TRIANGLE
BY EQUATION
Tan ½(A + B) / Tan ½(A − B) = (a + b) / (a − b)
Given C, a, b
C
a−b
a+b
½(A−B)
−½(A+B)
C
a+b
a−b
−½(A−B)
5° 170°
10° 160°
15° 150°
20° 140°
25° 130°
30° 120°
35° 110°
40° 100°
45° 90°
50° 80°
55° 70°
60° 60°
65° 50°
70° 40°
75° 30°
80° 20°
85° 10°
1
2
3
4
5
6
7
8
9
10
20
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20
10
9
8
7
6
5
4
3
2
1
85°
80°
75°
70°
65°
60°
55°
50°
45°
40°
35°
30°
25°
20°
15°
10°
5°

Fig. 29-67.

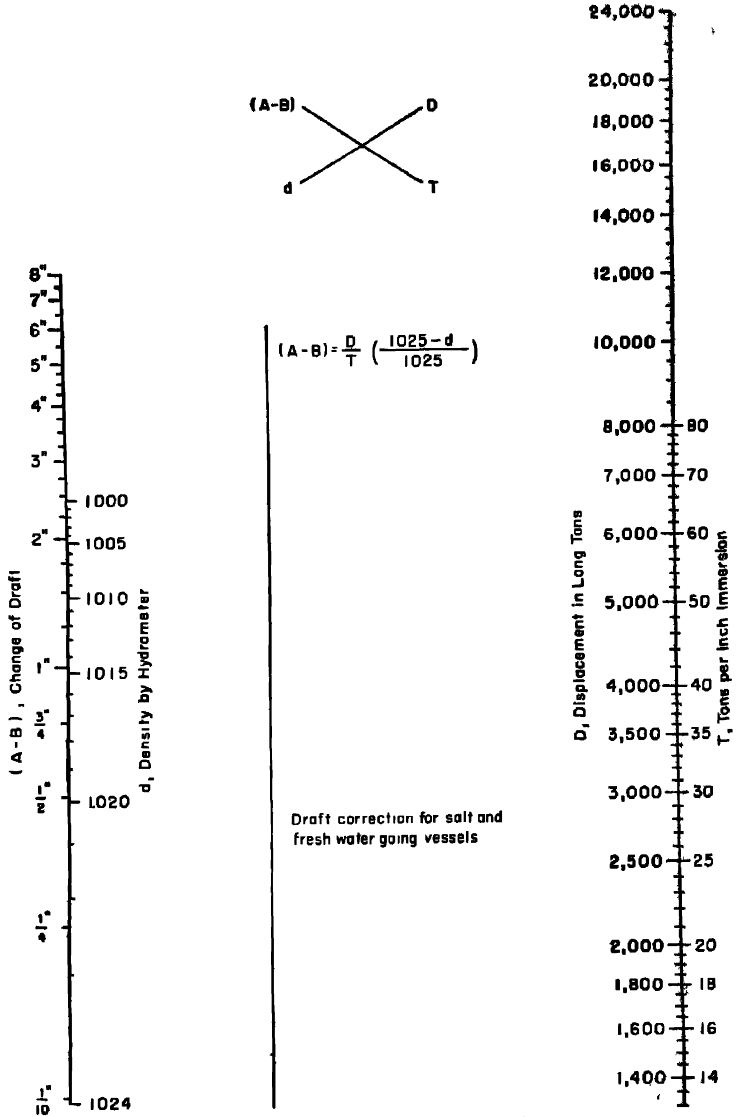


Fig. 29-68 $A - B = \frac{D}{T}\left(\frac{1.025 - d}{1.025}\right)$.

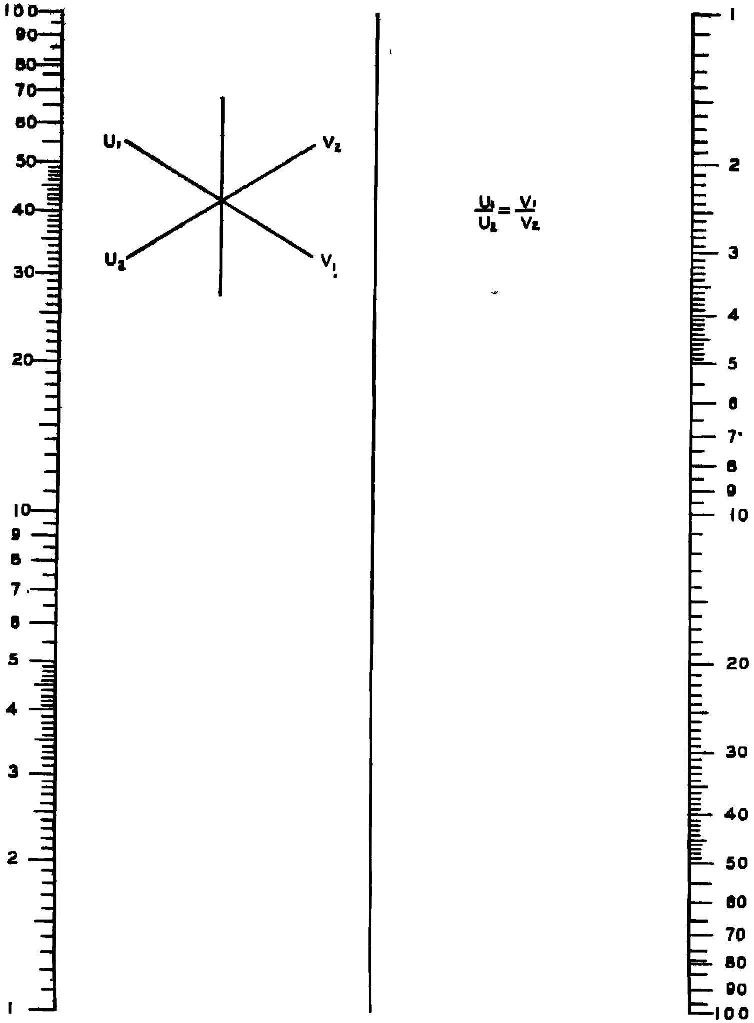


Fig. 29-69a.

Scales are laid off on two parallel axes, OX and $O'Y$, drawn in opposite directions, and the intersecting axis OO'. Let the length of OO' be K where K is any convenient length. Lay off the scales $m_1f_1(u)$ and $m_3f_3(w)$ on OX, $m_1f_2(v)$ on $O'Y$, and $m_4f_4(t)$ on OO', as indicated in Fig. 29-72.

The moduli are related by the equation

$$m_1/K = m_3/m_4$$

Any pair of parallel index lines (u, v) and (w, t) will cut the axes in values of u, v, w, and t, satisfying the equation.

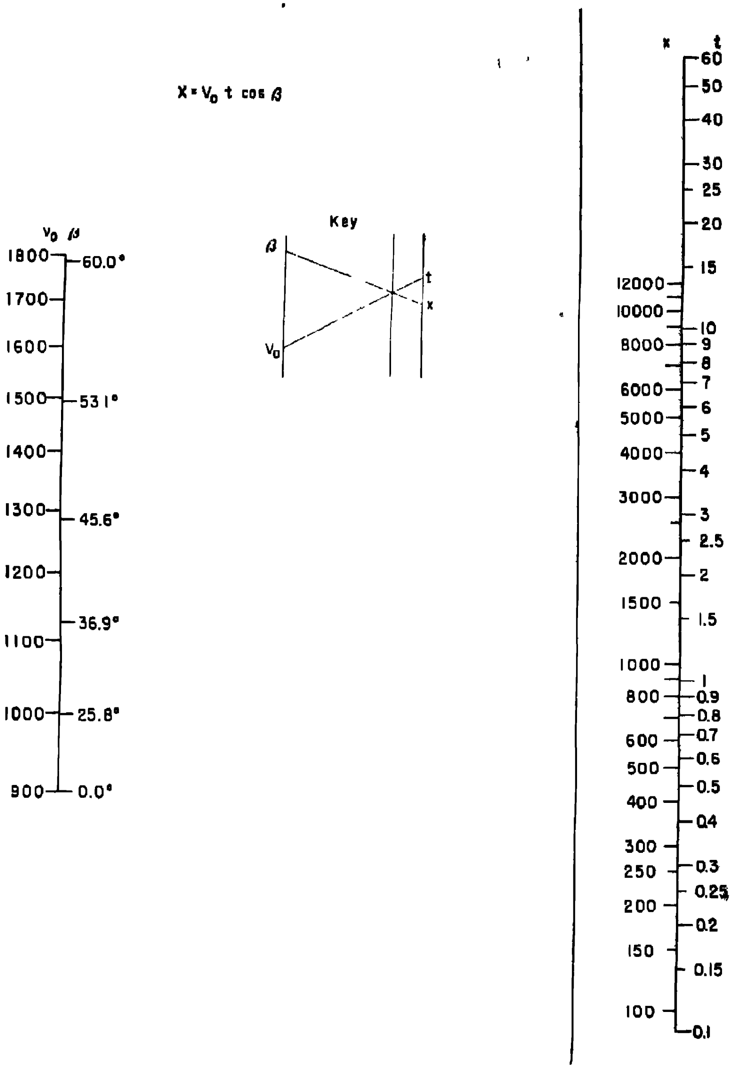


Fig. 29-69b.

7.
$$f_1(u) - f_2(v) = f_3(w)/f_4(t)$$

The chart for this equation is the same as that above except that the axes OX and $O'Y$ are drawn in the same direction. Again

$$m_1/K_4 = m_3/m_4$$

8.
$$\frac{1}{f_1(u)} + \frac{1}{f_2(v)} = \frac{1}{f_3(w)}$$

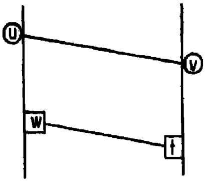


FIG. 29-70. $f_1(u) - f_2(v) = f_3(w) - f_4(t)$ or $f_1(u)/f_2(v) = f_3(w)/f_4(t)$.

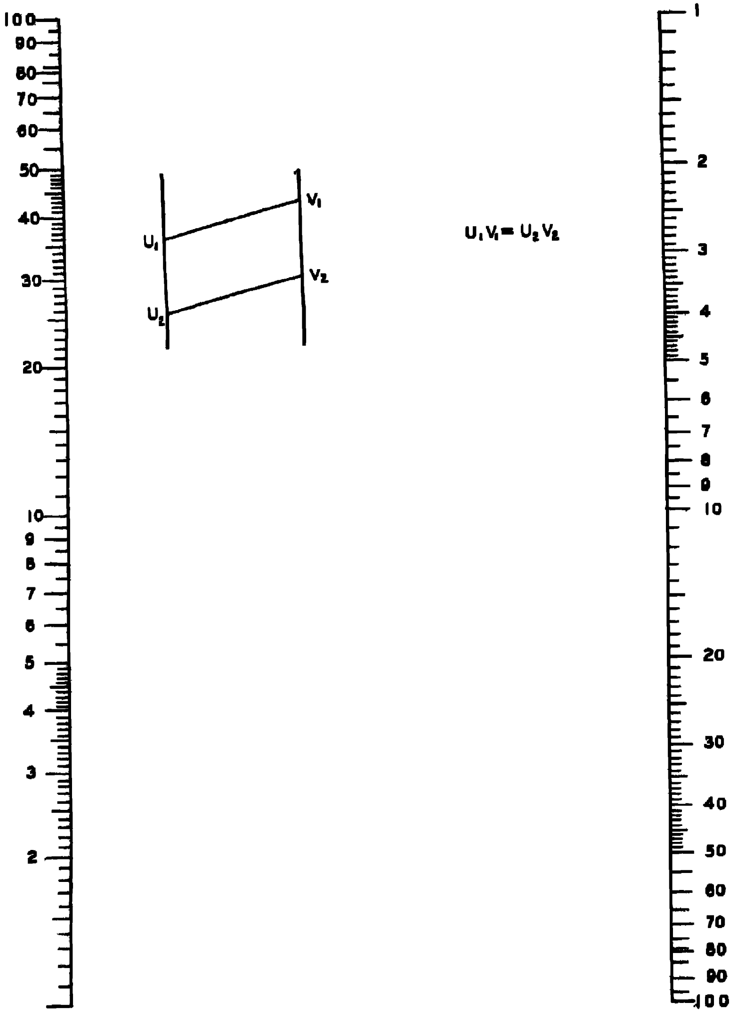


FIG. 29-71.

Draw three intersecting axes OX, OY, and OZ making angles of 60 deg with each other. On OX, OY, and OZ lay off the scales $m_1f_1(u)$, $m_1f_2(v)$, and $m_1f_3(w)$, respectively (Fig. 29-77). An index line cutting the axes in two of the variables will cut the other axis at the value of the third variable, satisfying the equation

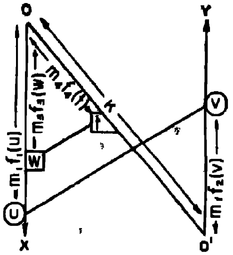


Fig. 29-72 $f_1(u) + f_2(v) = \frac{f_3(w)}{f_4(t)}$

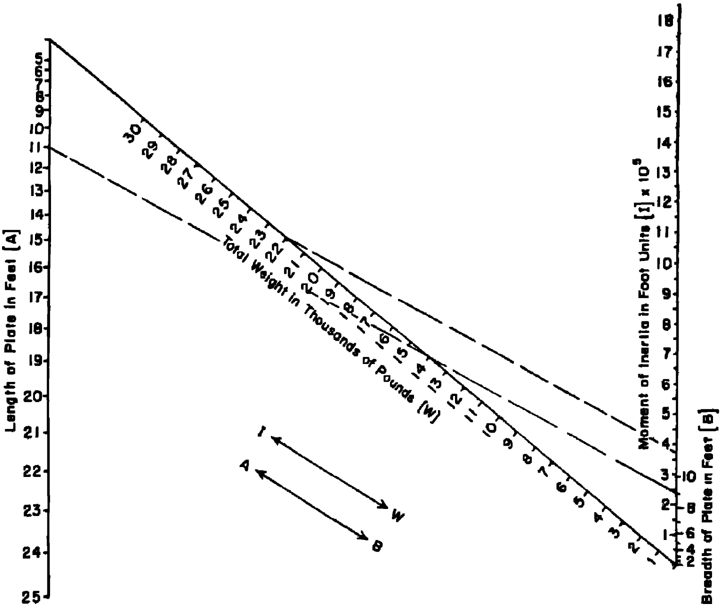


Fig. 29-73. Moment of inertia of a rectangular plate.

$$[1/f_1(u)] + [1/f_2(v)] = [1/f_3(w)]$$

It is to be noticed that negative values will be laid off in opposite directions.

9. $$\frac{1}{f_1(u)} + \frac{1}{f_2(v)} + \cdots = \frac{1}{f_n(t)}$$

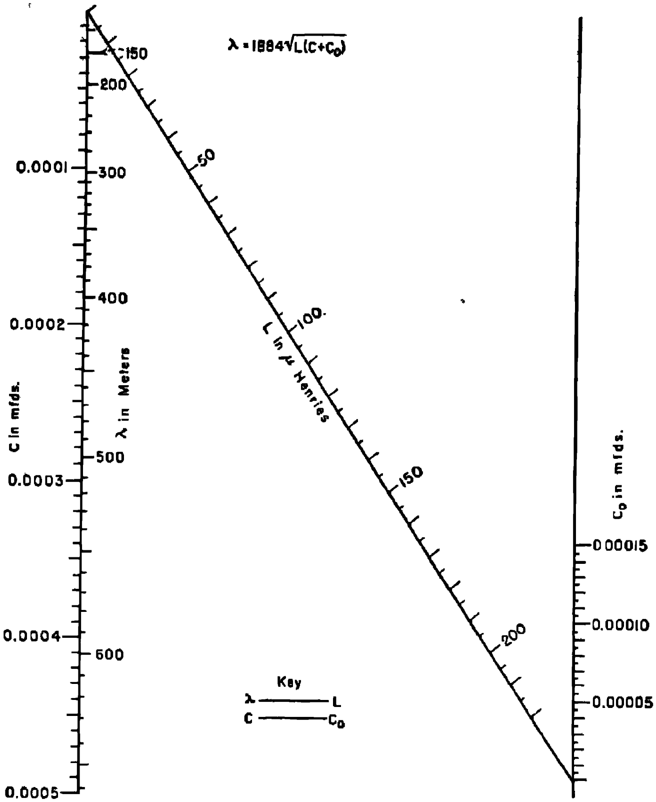


FIG. 29-74.

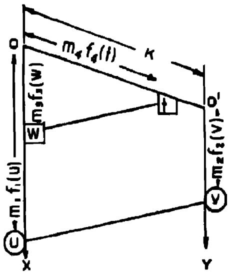
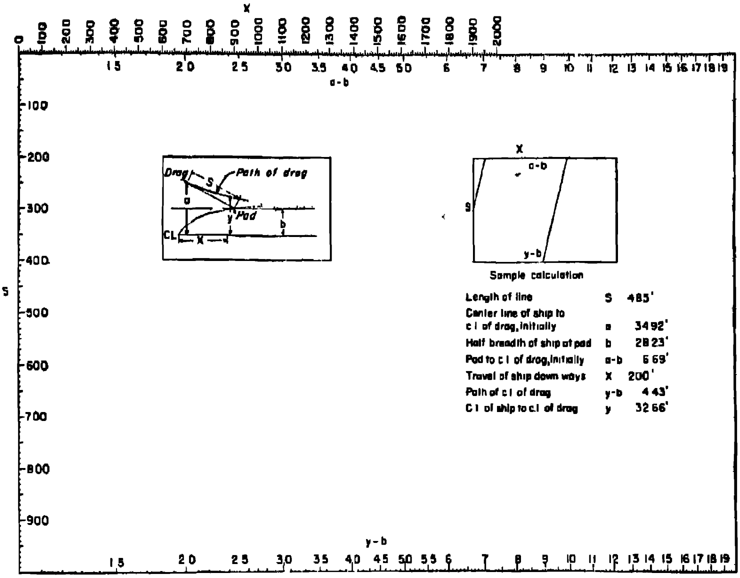


FIG. 29-75. $f_1(u) - f_2(v) = \dfrac{f_3(w)}{f_4(t)}$.



Sample calculation

Length of line	S	485'
Center line of ship to c l of drag, initially	a	34 92'
Half breadth of ship at pad	b	28 23'
Pad to c l of drag, initially	a-b	6 69'
Travel of ship down ways	X	200'
Path of c l of drag	y-b	4 43'
C l of ship to c.l of drag	y	32 66'

FIG. 29-76. Path of drag in ship launching.

The method of charting this equation is an extension of the method employed in the previous case. Consider the case of four variables

$$[1/f_1(u)] + [1/f_2(v)] + [1/f_3(w)] = [1/f_4(t)]$$

If we introduce an auxiliary variable Q, we can write

$$\frac{1}{f_1(u)} + \frac{1}{f_2(v)} = \frac{1}{Q} \qquad \text{and} \qquad \frac{1}{Q} + \frac{1}{f_3(w)} = \frac{1}{f_4(t)}$$

Chart each of these equations with a common Q scale which need not be graduated (Fig. 29-79).

10. $$f_1(u) + f_2(v) \cdot f_3(w) = f_4(w)$$

Draw two parallel axes OX and $O'Y$, and let K be the length of OO'. On OX lay off the scale $m_1 f_1(u)$, and on $O'Y$ lay off the scale $m_2 f_2(v)$ (Fig. 29-81). Construct the scale

$$x_1 = -\frac{m_1 K}{m_1 f_3(w) + m_2} f_3(w)$$

along the axis OO', and the scale

$$y_1 = \frac{m_1 m_2}{m_1 f_3(w) + m_2} f_4(w)$$

parallel to OX.

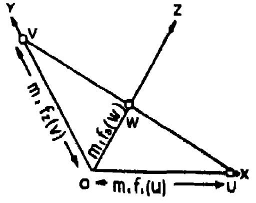


FIG. 29-77. $\frac{1}{f_1(u)} + \frac{1}{f_2(v)} = \frac{1}{f_3(w)}$.

Each value of w determines values of x_1 and y_1, *i.e.*, the position of a point (x_1, y_1) and the series of points determines a w scale on a curve or straight line. Any index line will cut the three scales in values of u, v, and w, satisfying the equation

$$f_1(u) + f_2(v) \cdot f_3(w) = f_4(w)$$

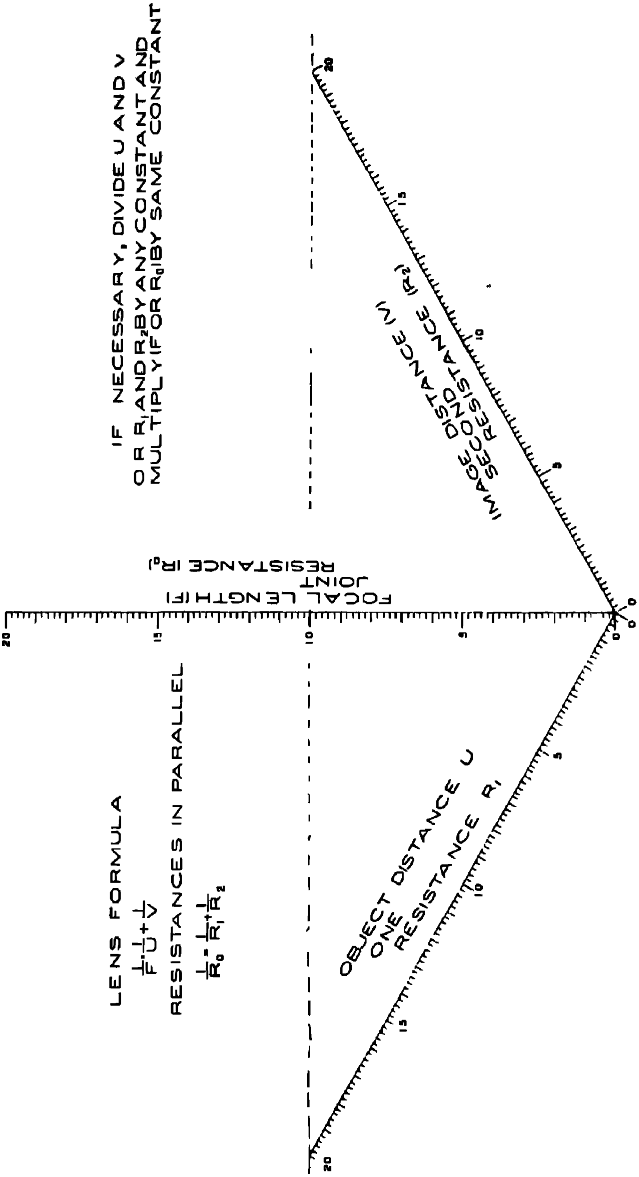


Fig 29-78. $\frac{1}{F} = \frac{1}{u} + \frac{1}{v}$.

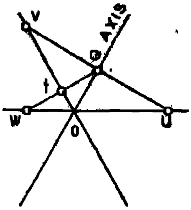


FIG. 29-79. $\dfrac{1}{f_1(u)} + \dfrac{1}{f_2(v)} + \cdots = \dfrac{1}{f_n(t)}$.

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \cdots + \frac{1}{R_n}$$

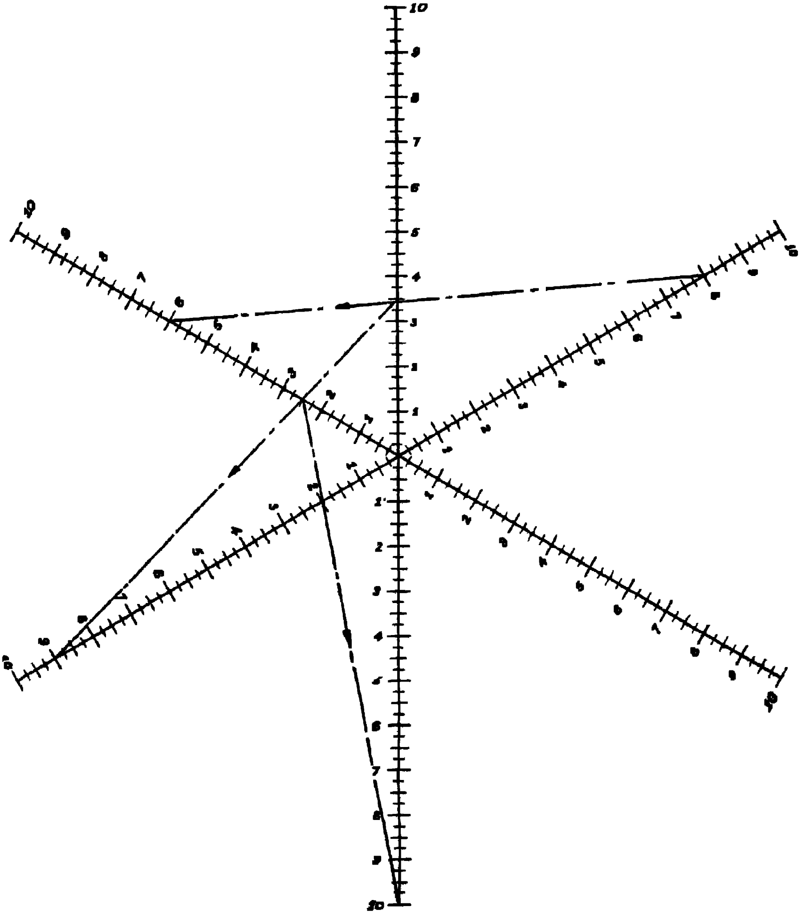


FIG. 29-80

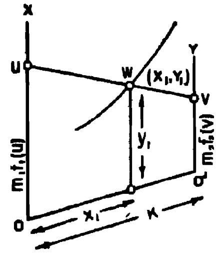


FIG. 29-81. $f_1(u) + f_2(v) \cdot f_3(w) = f_4(w)$.

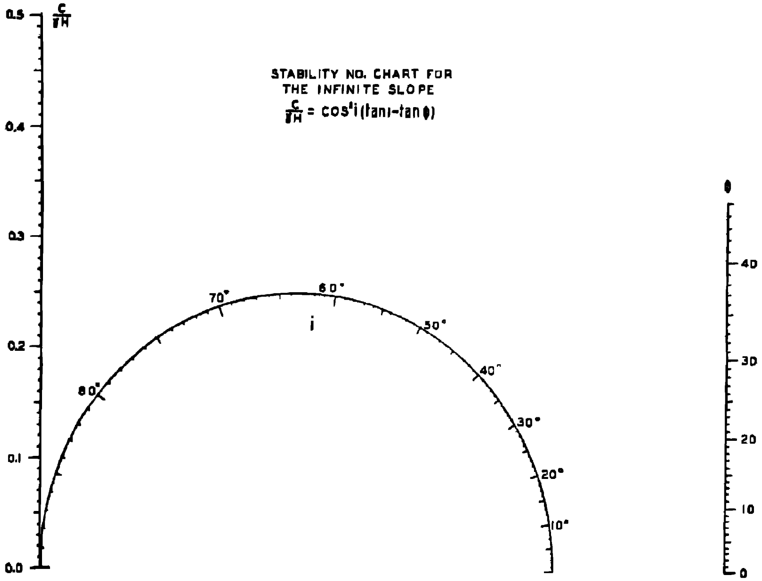


FIG. 29-82.

Charts of this type are illustrated by the following examples:

Example 1. Stability number for the infinite slope in a homogeneous soil

$$\frac{C}{\gamma H} = \cos^2 i(\tan i - \tan \phi)$$

where C = cohesion of soil, lb/sq ft
H = depth of element
i = slope of ground surface
ϕ = angle of internal friction of soil

The dimensionless quantity $C/\gamma H$ is the stability number. This equation may be written

$$C/\gamma H + \tan \phi \cos^2 i = \cos^2 i \tan i$$

and is in the form $f_1(u) + f_2(v)f_3(w) = f_4(w)$.
The scales are $x = m_1\,(C/\gamma H)$, $y = m_2 \tan \phi$,

$$x_1 = \frac{m_1 K}{m_1 \cos^2 i + m_2} \qquad \frac{m_1 m_2}{m_1 \cos^2 i + m_2} \cos^2 i \tan i$$

Let $m_1 = 15$, $m_2 = 5$, and $K = 10$.

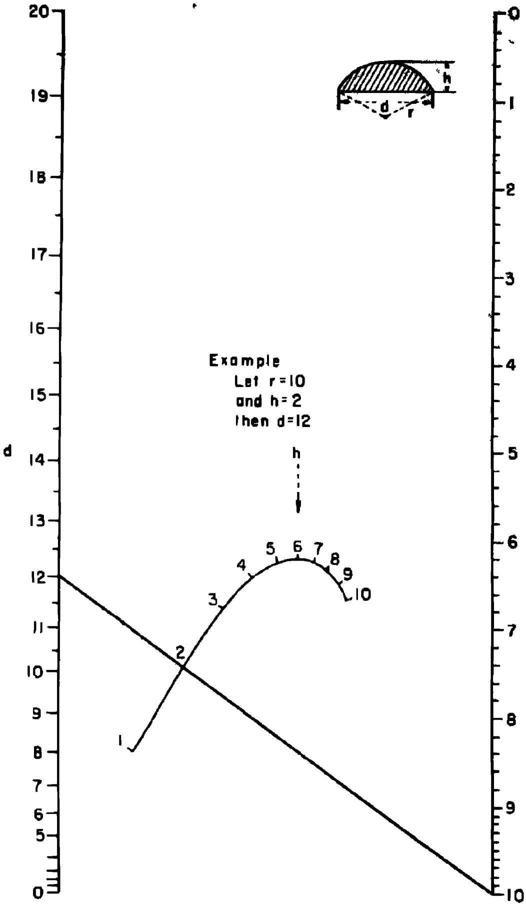


Fig. 29-83. $d^2 = 8rh - 4h^2$.

Construct the scales

$$x = 15(C/\gamma H) \qquad y = 5 \tan \phi$$

$$x_1 = \frac{30}{3\cos^2 i + 1}\cos^2 i \qquad y_1 = \frac{15}{3\cos^2 i + 1}\cos^2 i \tan i$$

Notice that the rectangular equation of the (x_1, y_1) curve is

$$\frac{(x_1 - \frac{15}{4})^2}{(\frac{15}{4})^2} + \frac{y_1^2}{(\frac{15}{4})^2} = 1$$

which is a circle with center at $(\frac{15}{4}, 0)$ and radius equal to $\frac{15}{4}$. The chart in Fig. 29-82 shows that for any given value of ϕ, there are two values of i which give the same stability number.

Example 2. Chordal height and length of chord

$$d^2 = 8rh - 4h^2$$

where r = radius of the circle
d = length of the chord
h = chordal height

On the chart in Fig. 29-83, m_1 0.2, m_1 = 1, and k = 11.2 in.

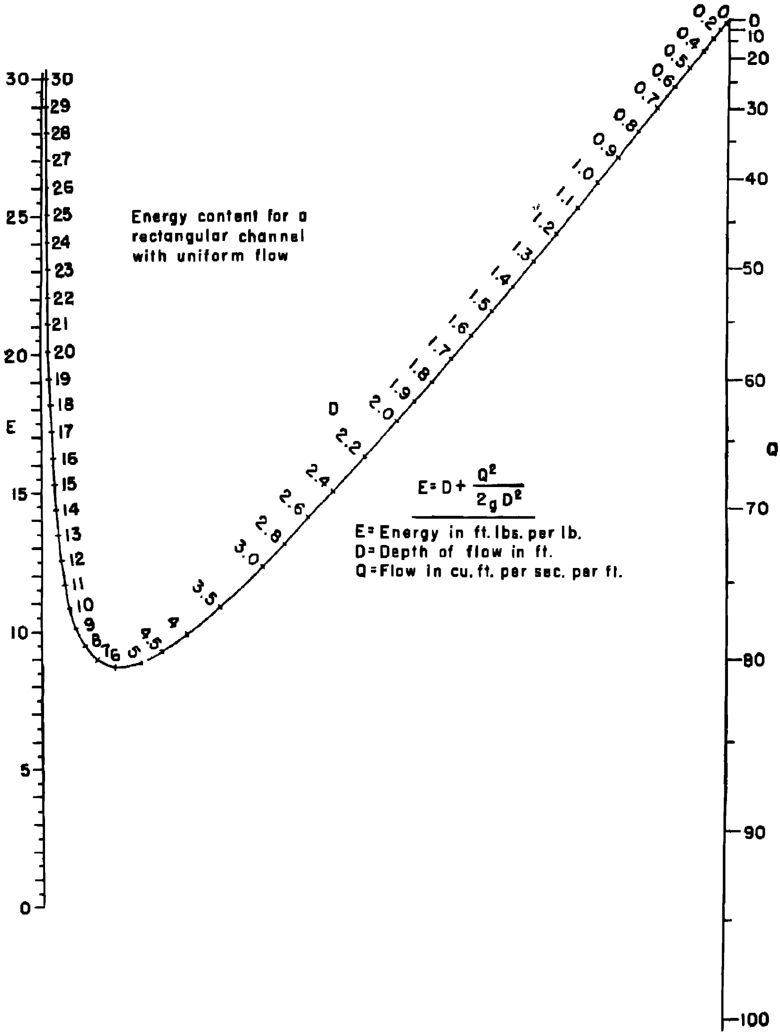


FIG 29-83a.

EXAMPLE 3. Energy content for a rectangular channel with uniform flow

$$E = D + (Q^2/2gD^2)$$

where E = energy, ft-lb per lb
D = depth of flow, ft
Q = flow, cu ft/(sec)/(ft) (Fig. 29-83a)

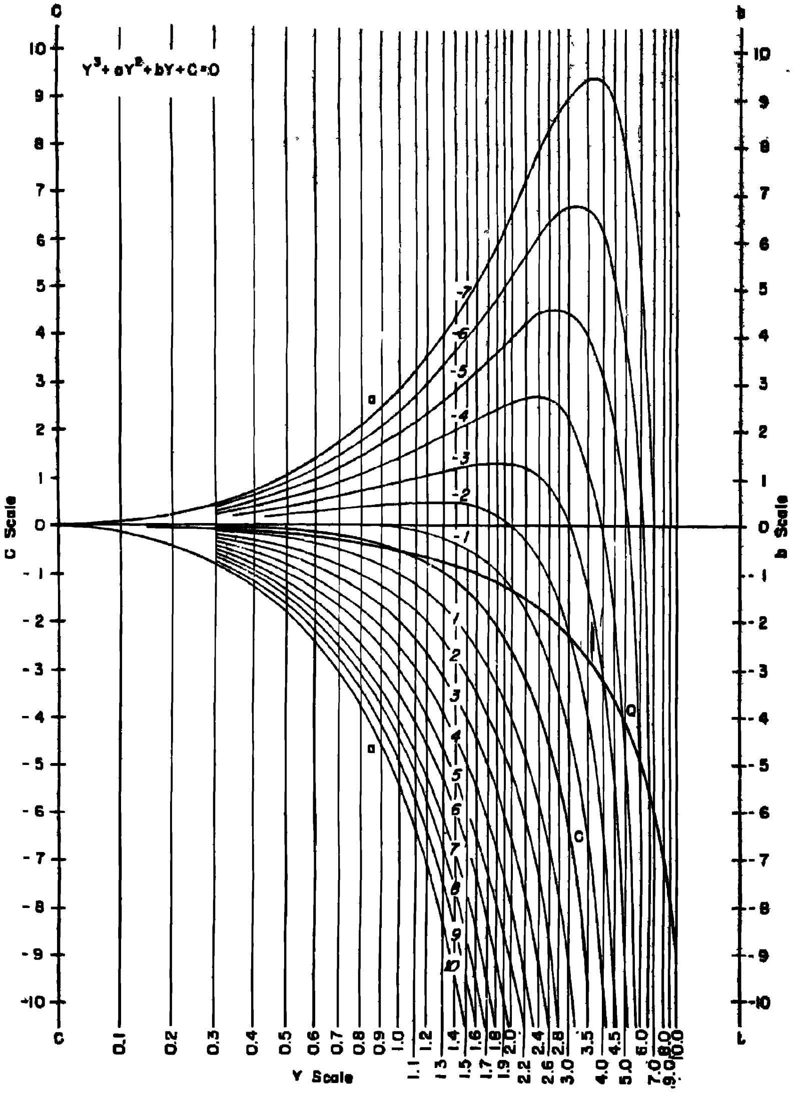


FIG. 29-84.

Example 4. Cubic equation. Figure 29-84 is a chart for the real value solutions of the cubic equation $y^3 + ay^2 + by + c = 0$. For each value of a, a single curve of (x_1, y_1) is produced. If a is zero, only the curve marked C is necessary to solve the equation $y^3 + by + c = 0$. If $c = 0$, then $y^2 + ay + b = 0$, use the curve marked Q.

Example 5. Francis formula for contracted weir (Fig. 29-85).

$$Q = 3.33(B - 0.2H)H^{3/2}$$

11. $$\frac{f_1(u)}{f_2(v)} = \frac{f_3(w)}{f_4(t)} \quad \text{or} \quad f_1(u)f_4(t) = f_2(v)f_3(w)$$

Scales are laid off on two parallel axes OX and $O'Y$, drawn in opposite directions and at any distance K apart (Fig. 29-86a). Lay off the scales $m_1f_1(u)$ and $m_3f_3(w)$ on the OX axis and $m_2f_2(v)$ and $m_4f_4(t)$ on the $O'Y$ axis. The moduli are related by the equation

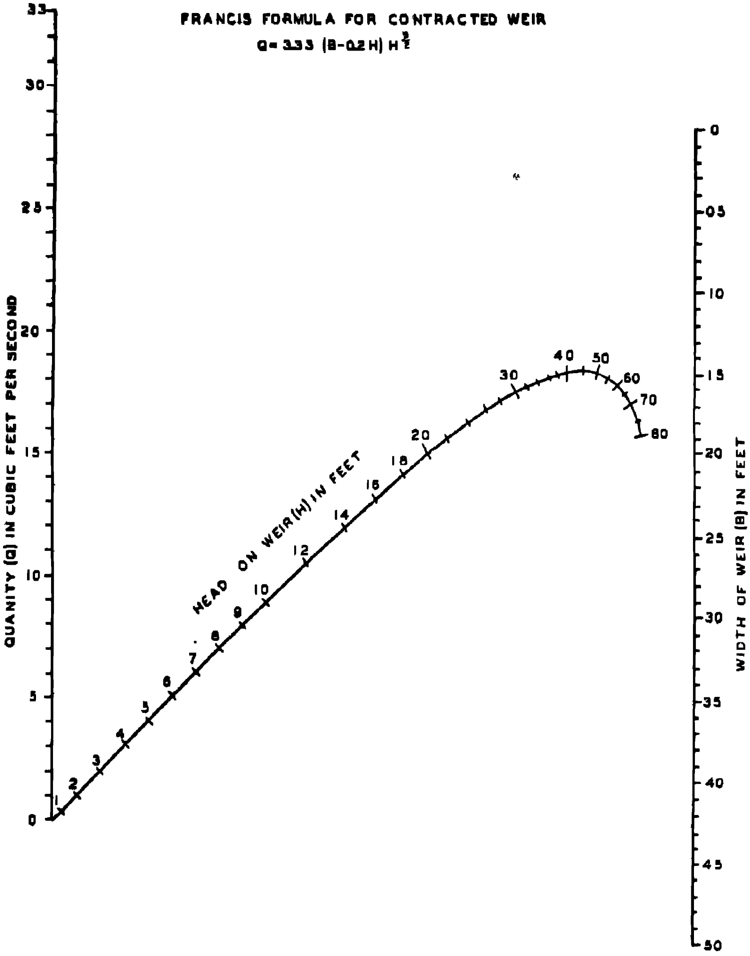
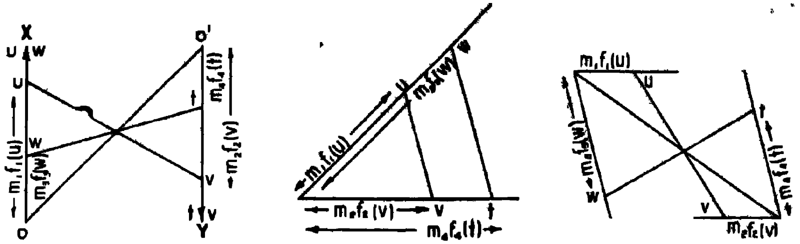


Fig. 29-85.

$$m_1/m_2 = m_3/m_4$$

Two index lines intersecting in a point on OO' will give values of u, v, w, and t satisfying the equation. A chart for the equation $u_1/v_1 = u_2/v_2$ is shown in Fig. 29-87. Boyle's law is shown in Fig. 29-87a. Draft correction for salt- and fresh-water-going vessels is shown in Fig. 29-87b.



a *b*

FIG. 29-86. $\frac{f_1(u)}{f_2(v)} = \frac{f_3(w)}{f_4(t)}$.

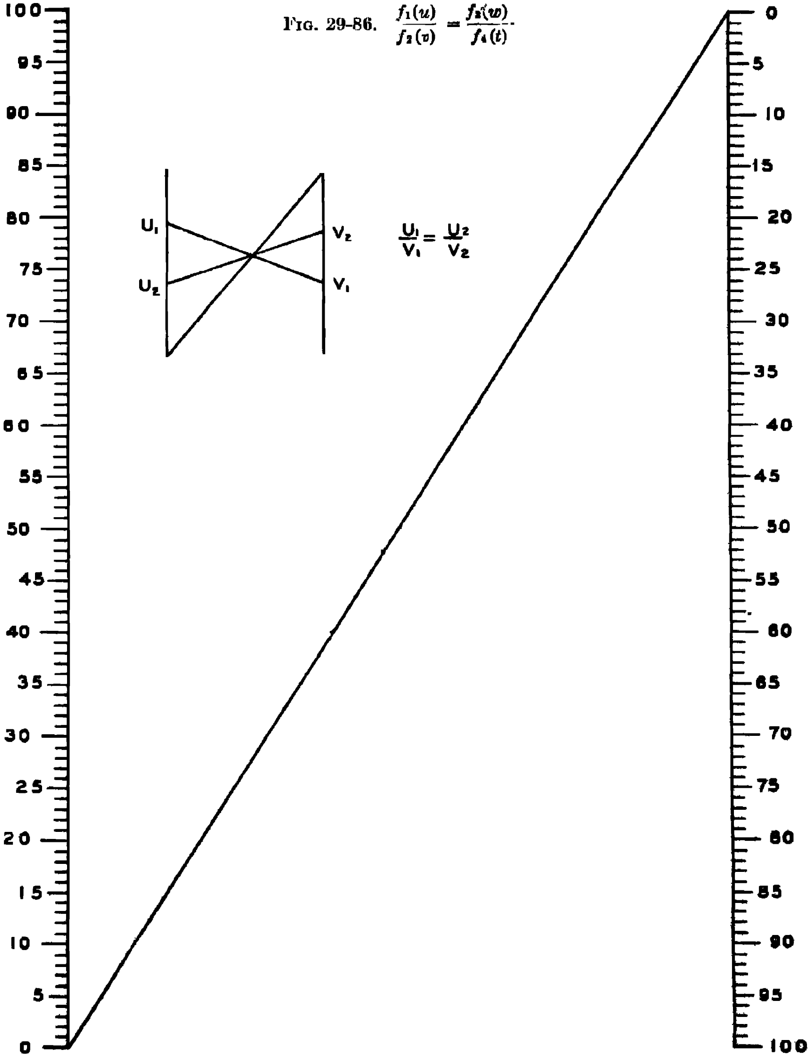


FIG. 29-87.

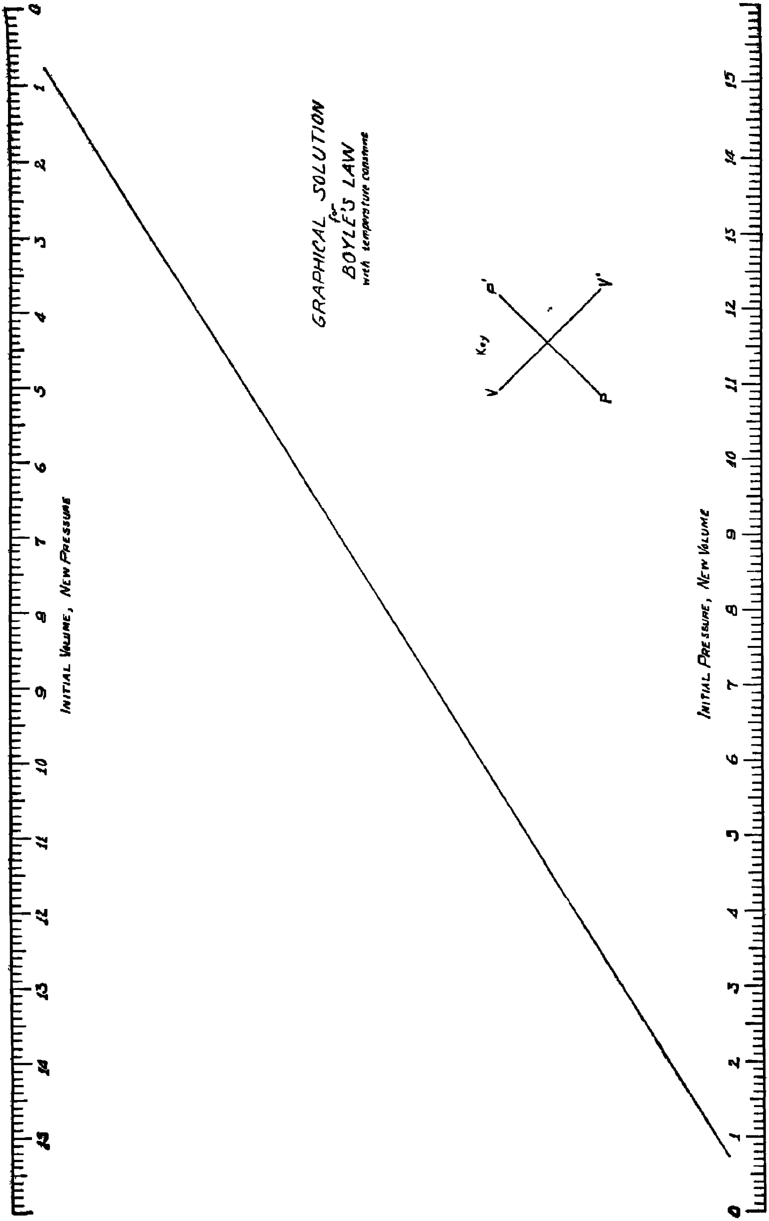

GRAPHICAL SOLUTION for BOYLE'S LAW
with temperature constant
Key
V
P'
V'
P
Initial Volume, New Pressure
Initial Pressure, New Volume
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15

FIG 29-87a

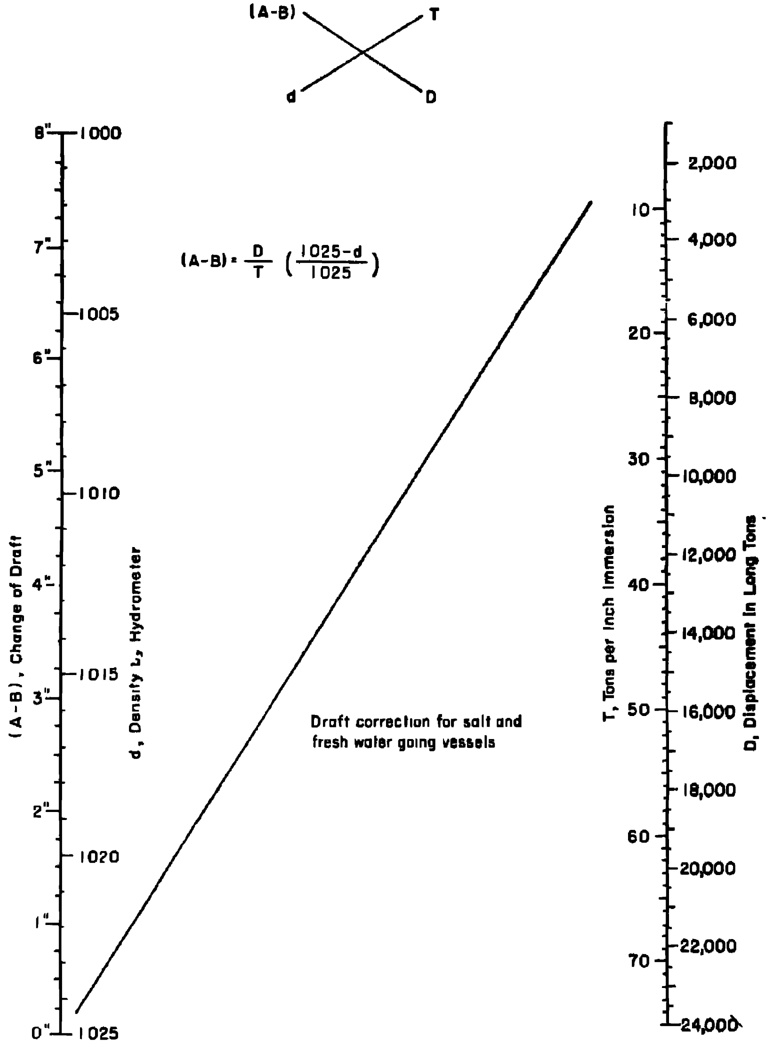

(A-B)
T
d
D
(A-B) = D/T (1025-d / 1025)
8"
7"
6"
5"
4"
3"
2"
1"
0"
1000
1005
1010
1015
1020
1025
(A-B), Change of Draft
d, Density L, Hydrometer
Draft correction for salt and fresh water going vessels
10
20
30
40
50
60
70
T, Tons per Inch Immersion
2,000
4,000
6,000
8,000
10,000
12,000
14,000
16,000
18,000
20,000
22,000
24,000
D, Displacement in Long Tons

FIG 29-87b

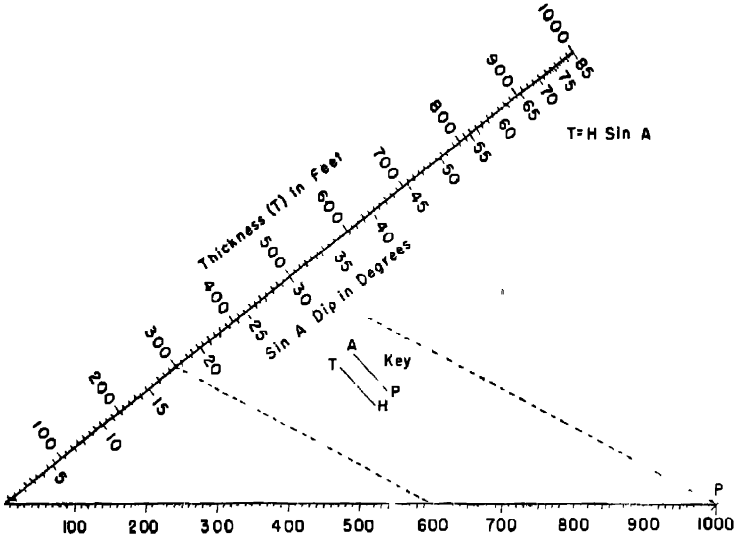


FIG 29-87c.

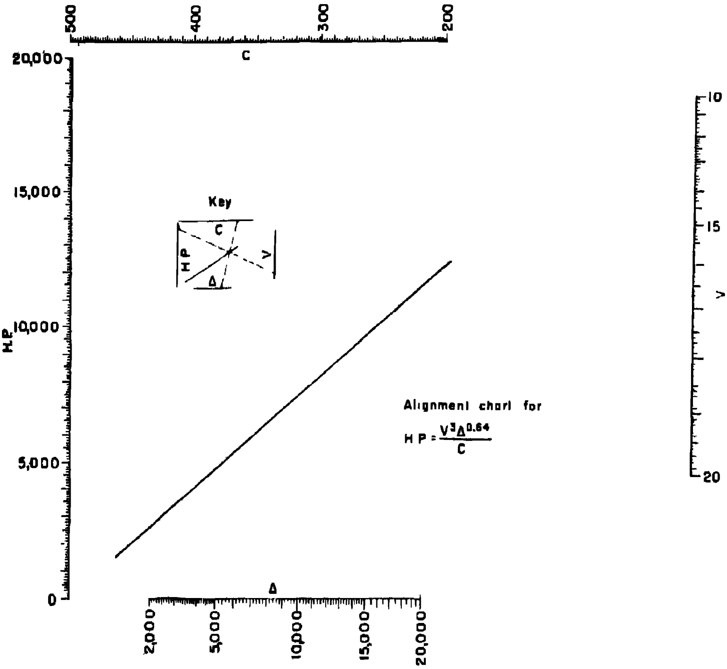


FIG. 29-87d.

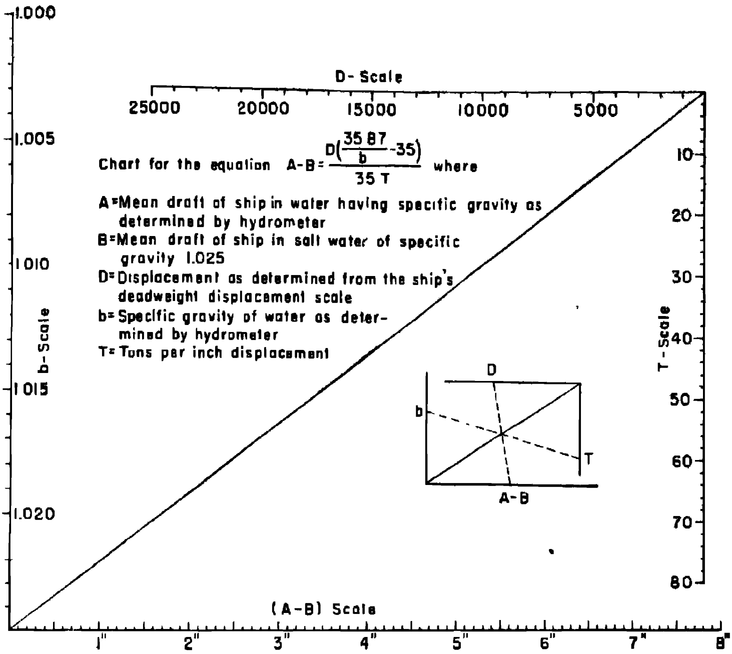


Fig. 29-87e.

It should be noticed that the two equal angles on whose sides scales are constructed may be placed in various positions as indicated in Figs. 29-86b, 29-86c.

The charts for $T = H \sin A$ (Fig. 29-87c), Hp $= V^3\Delta^{0.64}/C$ (Fig. 29-87d),

$$A - B = D\{[(35.87/b) - 35]/35T\}$$

(Fig. 29-87e), and the equation $1/u\sqrt{v} = \sqrt[3]{w}/t$ (Fig. 29-87f) illustrate this type.

11a. $$f_1(u) = f_3(w) \cdot f_2(v)$$

If $f_4(t) = C$ (constant) in type 11, then $f_1(u) = [f_3(w) \cdot f_2(v)]/C$. Consequently the t scale in type 11 is replaced by a single fixed point. In this case, however, it is convenient to project through the fixed point each of the points on the w scale onto the line OO'.

The form of the completed chart is shown in Fig. 29-88c.

11b. $$f_1(u) = Cf_2(v)$$

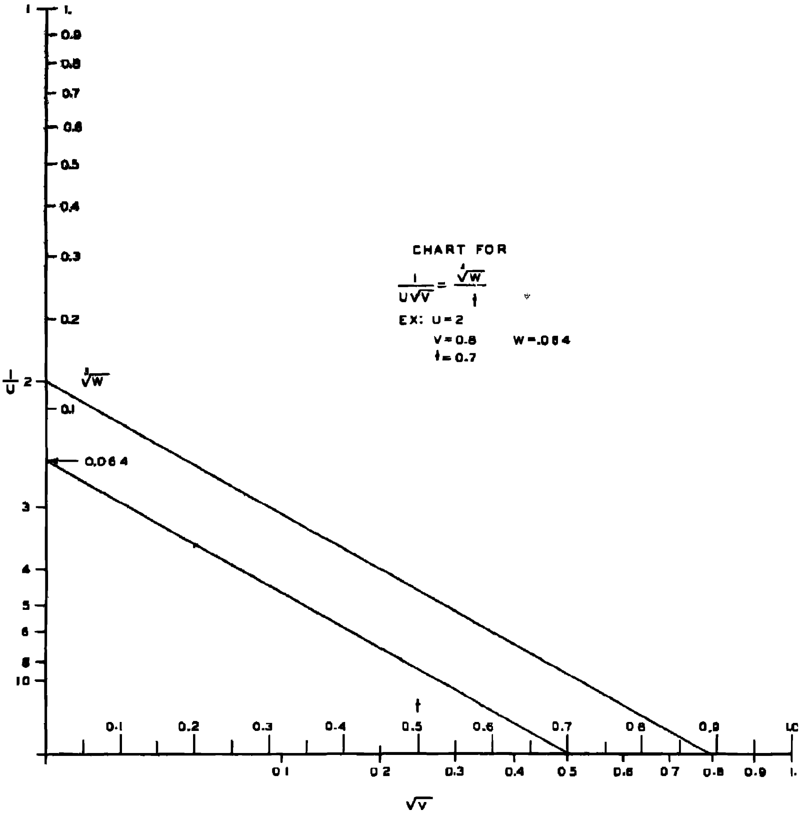


FIG. 29-87*f*.

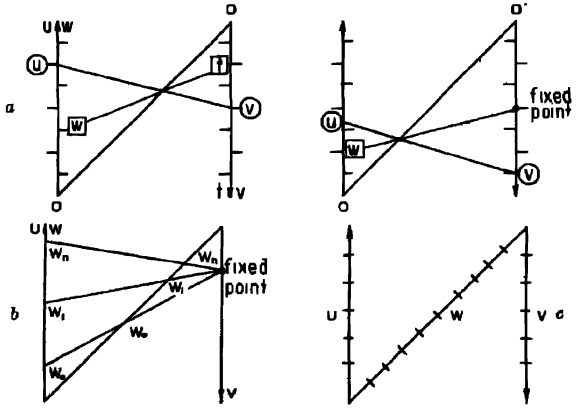
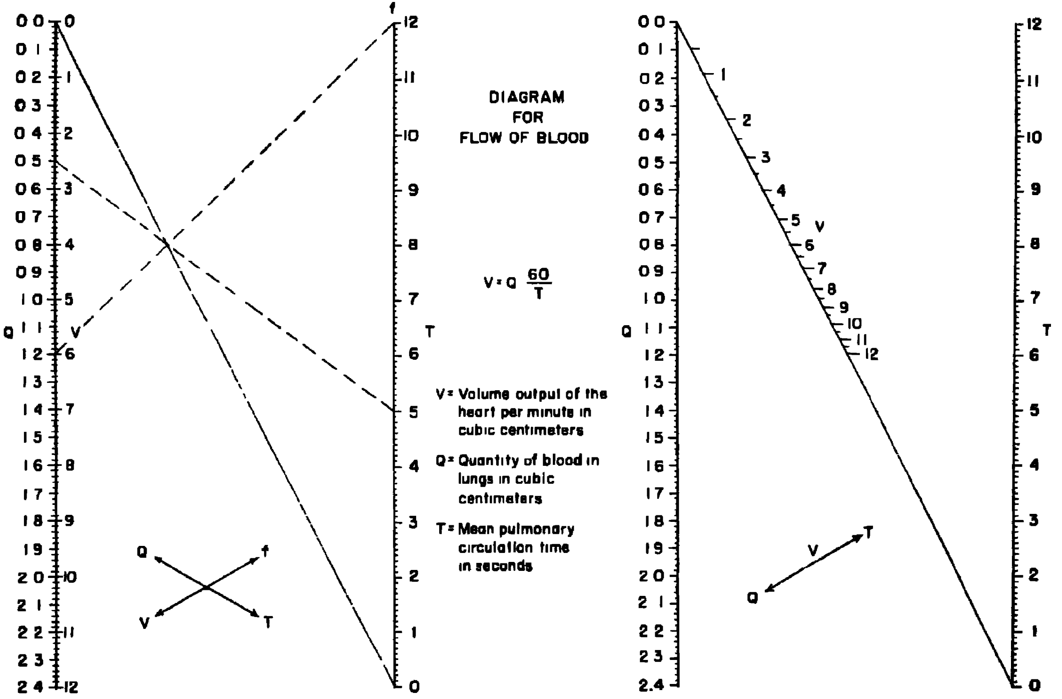


FIG. 29-88. $f_1(u) = f_3(w) \cdot f_2(v)$.



V = Volume output of the heart per minute in cubic centimeters

Q = Quantity of blood in lungs in cubic centimeters

T = Mean pulmonary circulation time in seconds

Fig. 29-89

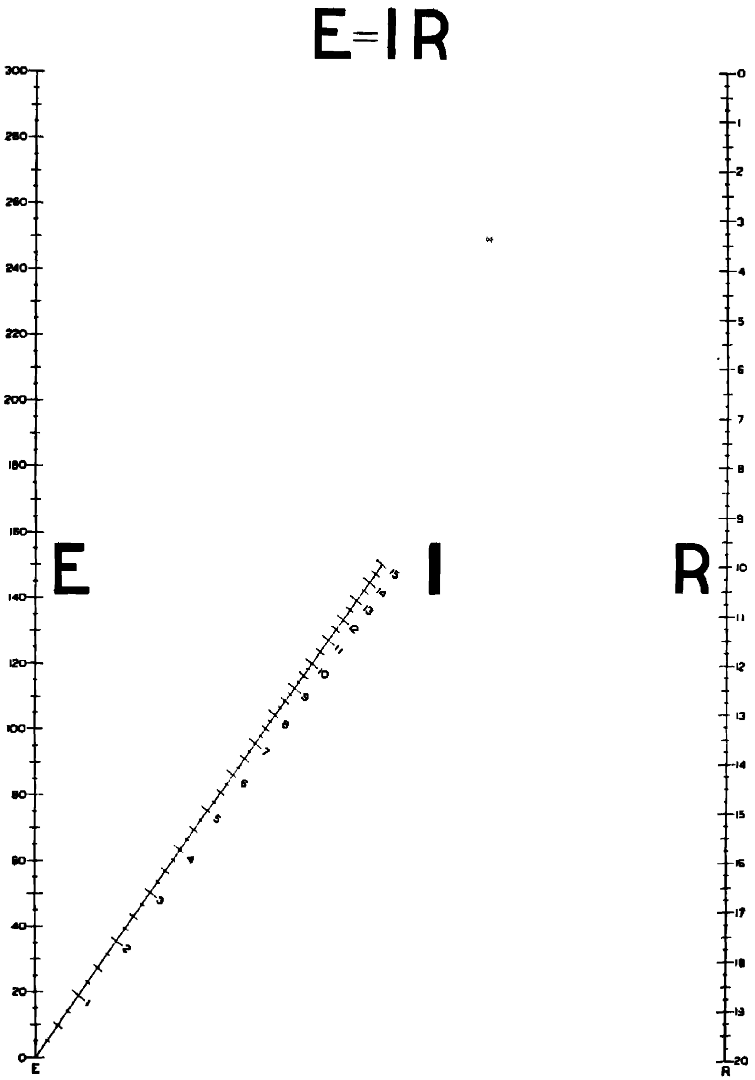

E=IR
E
I
R
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280
300
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20

FIG. 29-90.

If $f_3(w) = C$ (constant) and $f_4(t) = 1$, type 11 reduces to the form $f_1(u) = Cf_2(v)$. By the method used in 11, the chart would appear to be as in Fig. 29-91. An index line joining the two fixed points would determine a fixed point P on the intersecting line OO'.

Consequently, to chart an equation of the form $f_1(u) = Cf_2(v)$, draw the scales for $f_1(u)$ and $f_2(v)$, assign a value to v, and find the corresponding value of u from the equation. Draw the index line between the values of u and v, and where the index line cuts the line OO' at P, a fixed turning point is established.

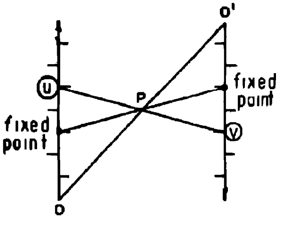


Fig. 29-91. $f_1(u) = Cf_2(v)$.

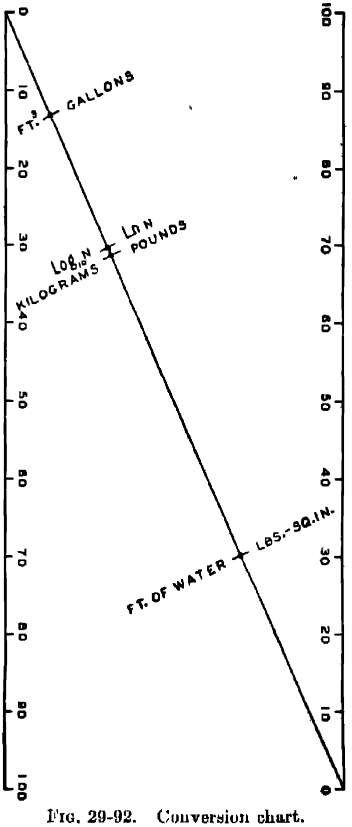


Fig. 29-92. Conversion chart.

This is a convenient type of chart to use for conversion formulas. It is evident that an infinite number of conversion formulas can be solved on one chart. Each fixed point is a key to a definite formula.

Figure 29-92 illustrates this type of chart.

11c. $$f_1(u) \cdot f_2(v) \cdots = f_i(r) \cdot f_j(s) \cdots$$

This is obviously an extension of type 11. For example, if

$$f_1(u) \cdot f_2(v) \cdot f_3(w) = f_4(t) \cdot f_5(s)$$

we can write

$$f_1(u) \cdot f_2(v) \cdot f_3(w) \cdot Q = f_4(t) \cdot f_5(s) \cdot Q \cdot 1$$

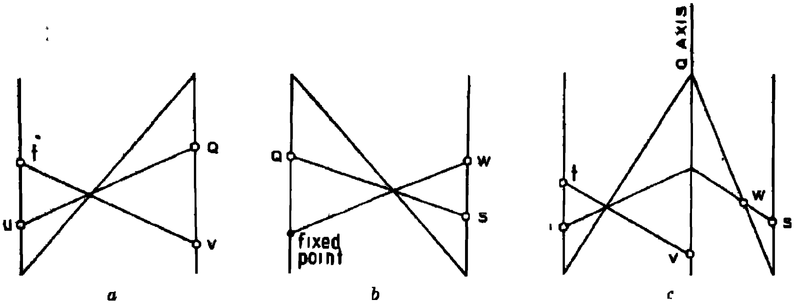


FIG. 29-93. $f_1(u) \cdot f_2(v) \cdots = f_i(r) \cdot f_j(s) \cdots$.

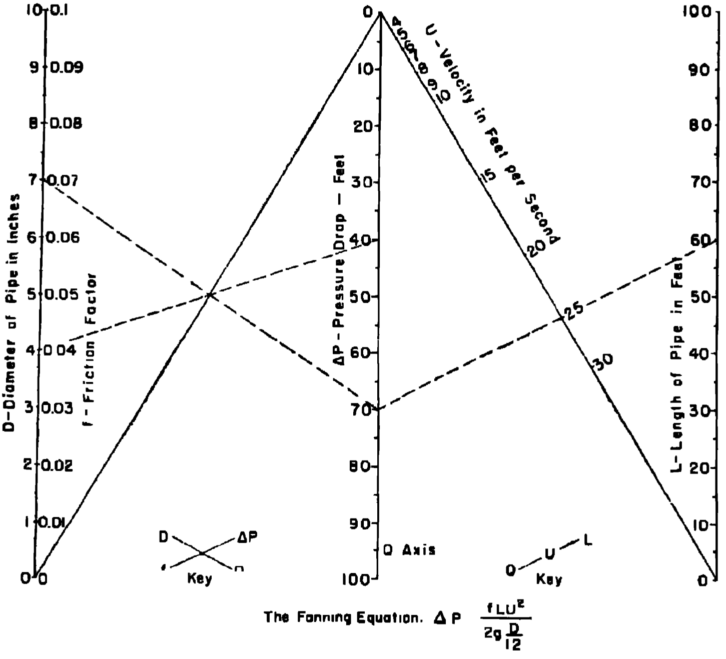


FIG. 29-94a.

where Q is an auxiliary variable. The latter equation may be written in the form

$$\frac{f_1(u)}{Q} = \frac{f_4(t)}{f_2(v)} \quad \text{and} \quad \frac{Q}{f_6(s)} = \frac{1}{f_3(w)}$$

Figures 29-94a, 29-94b, and 29-94c are charts of this type.

11d. $f_1(u) = Cf_2(v)$ (C is constant)

Taking logarithms of both members the equation becomes

$$\log f_2(v) = \log f_1(u) - \log C$$

or

$$\log f_1(u) - \log f_2(v) = \log C$$

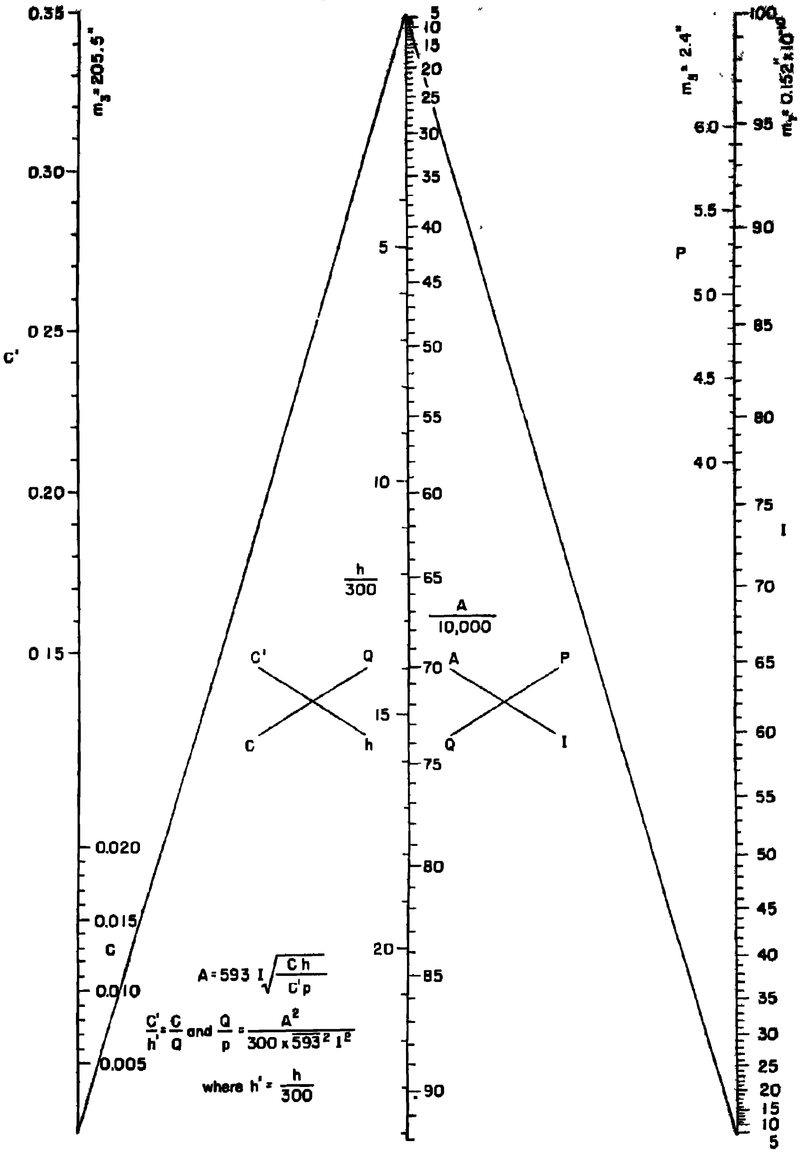

C'
$m_3 = 205.5''$
0.35
0.30
0.25
0.20
0.15
C
0.020
0.015
0.010
0.005
$\frac{h}{300}$
5
10
15
20
$\frac{A}{10{,}000}$
5
10
15
20
25
30
35
40
45
50
55
60
65
70
75
80
85
90
P
$m_3 = 2.4''$
6.0
5.5
5.0
4.5
4.0
I
$m_1 = 0.152 \times 10^{-6''}$
100
95
90
85
80
75
70
65
60
55
50
45
40
35
30
25
20
15
10
5
C'
Q
C
h
A
P
Q
I
$A = 593\,I\sqrt{\frac{Ch}{C'p}}$
$\frac{C'}{h'} = \frac{C}{Q}$ and $\frac{Q}{p} = \frac{A^2}{300 \times 593^2\,I^2}$
where $h' = \frac{h}{300}$

FIG. 29-94b.

By the first equation we have two parallel scales, while the third scale is reduced to a point.

A conversion chart is shown by Fig. 29-96.

Generalization and Combinations. Sometimes an equation may be in such a form as not to adapt itself immediately to one of the previously suggested cases. When this situation arises, it may be possible to introduce one or more auxiliary variables

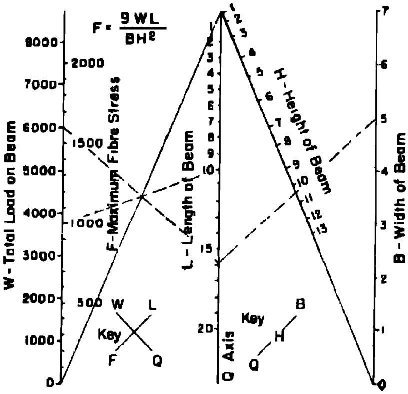


FIG. 29-94c.

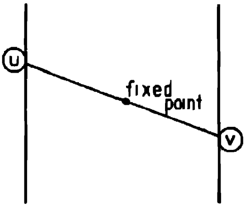


FIG. 29-95. $f_1(u) = Cf_2(v)$.

and split the original equation into two or more parts, applying one or more of the proposed forms to each of the various parts. In other cases, by holding one variable constant, a single scale is produced. Then for the complete range of that variable, a series of scales results. This situation gives rise to a so-called network of scales. Nothing new is involved since it is an obvious extension of the previous forms discussed. It is evident that a six-variable relationship might be represented by three sets of networks, which is analogous to three scales for a three-variable relationship. The possible forms suggested are illustrated by the following typical equations:

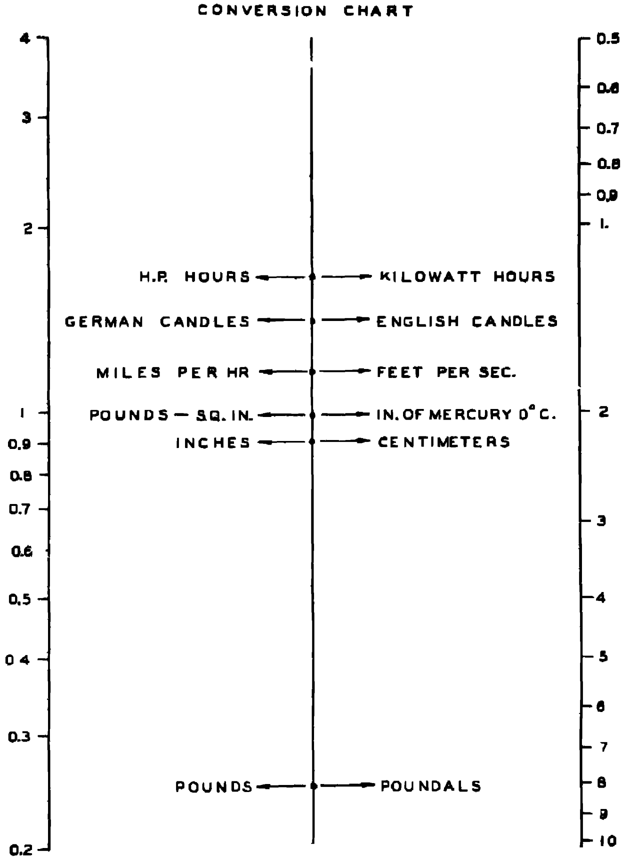

CONVERSION CHART
4
3
2
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
H.P. HOURS
KILOWATT HOURS
GERMAN CANDLES
ENGLISH CANDLES
MILES PER HR
FEET PER SEC.
POUNDS — SQ. IN.
IN. OF MERCURY 0° C.
INCHES
CENTIMETERS
POUNDS
POUNDALS
0.5
0.6
0.7
0.8
0.9
1.
2
3
4
5
6
7
8
9
10

Fig. 29-96.

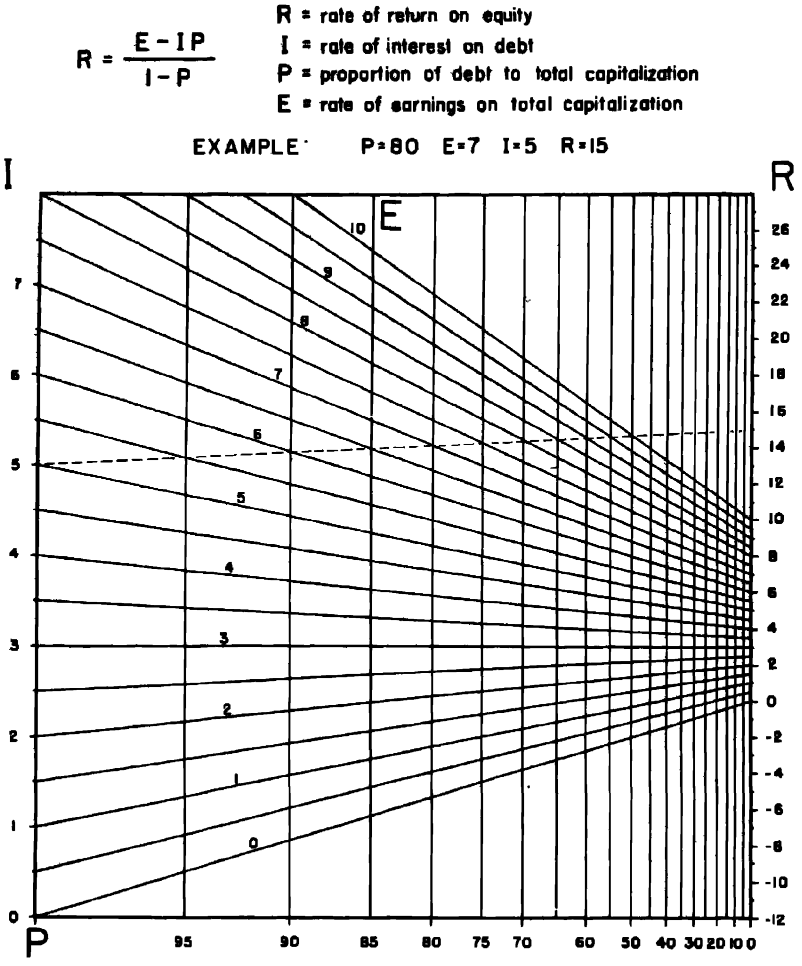


Fig. 29-97.

1. Equity debt equation (Fig. 29-97):

$$R = \frac{E - IP}{1 - P}$$

where R = rate of return on equity
I = rate of interest on debt
P = proportion of debt to total capitalization
E = rate of earnings on total capitalization

2. Rectangular suppressed weir (Fig. 29-98)

3. $$\frac{wH^2}{2 \sin^2 \theta} K$$

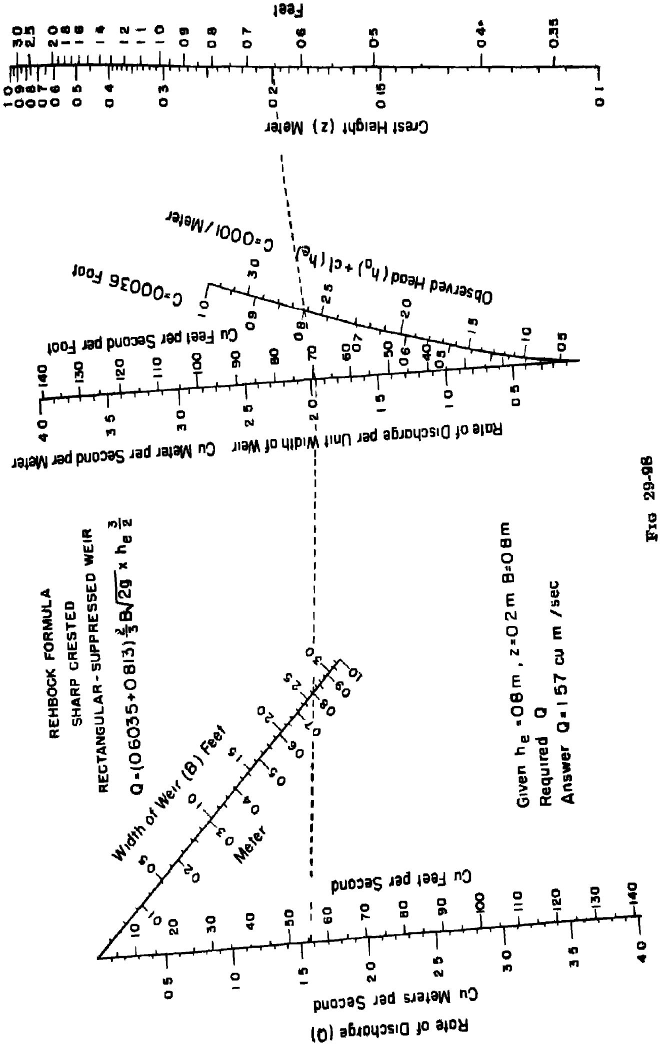
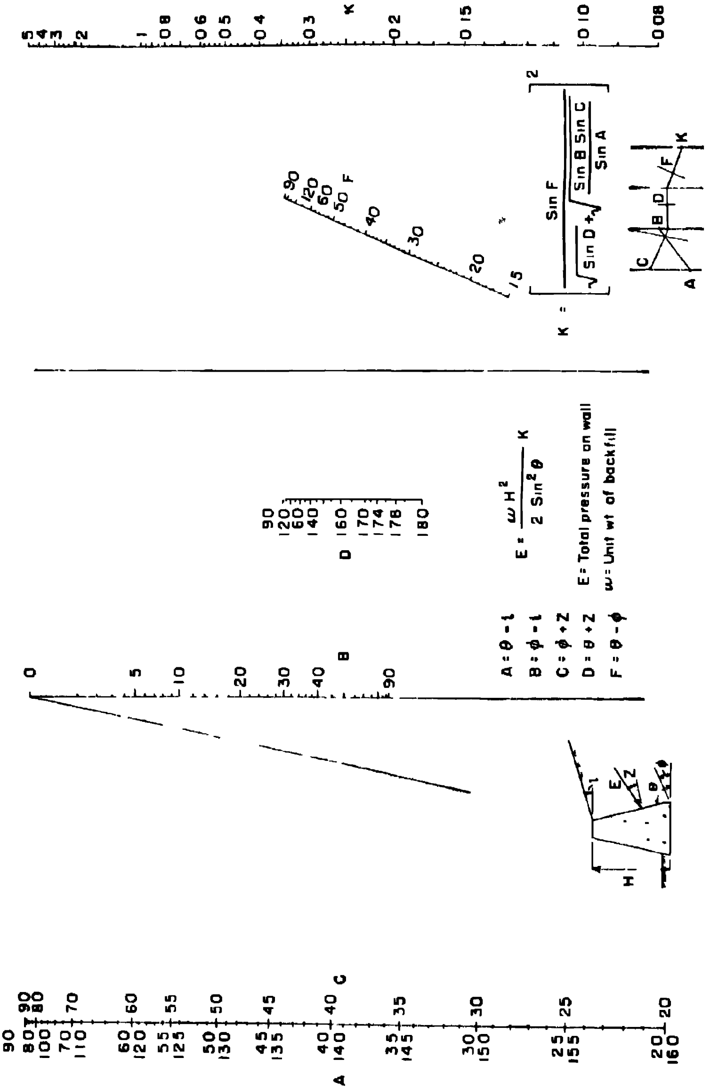

REHBOCK FORMULA
SHARP CRESTED
RECTANGULAR-SUPPRESSED WEIR
Q=(0.6035+0.0813) $\frac{2}{3}$B$\sqrt{2g}$ x $h_e^{\frac{3}{2}}$
Given h_e =0.8 m , z=0.2 m B=0.8 m
Required Q
Answer Q=1.57 cu m /sec
Crest Height (z) Meter
Feet
Observed Head (h_o)+c (h_e)
C=0.001/Meter
C=0.0036 Foot
Cu Feet per Second per Foot
Rate of Discharge per Unit Width of Weir Cu Meter per Second per Meter
Width of Weir (B) Feet
Meter
Cu Feet per Second
Cu Meters per Second
Rate of Discharge (Q)

FIG 29-98



$$K = \left[\frac{\sin F}{\sqrt{\sin D} + \sqrt{\frac{\sin B \sin C}{\sin A}}} \right]^2$$

$$E = \frac{\omega H^2}{2 \sin^2 \theta} K$$

E = Total pressure on wall

ω = Unit wt of backfill

$A = \theta - \iota$

$B = \phi - \iota$

$C = \phi + Z$

$D = \theta + Z$

$F = \theta - \phi$

FIG. 29-99.

$$\text{where } K = \left(\frac{\sin F}{\sqrt{\sin D} + \sqrt{\dfrac{\sin B \sin C}{\sin A}}} \right)^2 \quad \text{(Fig. 29-99)}$$

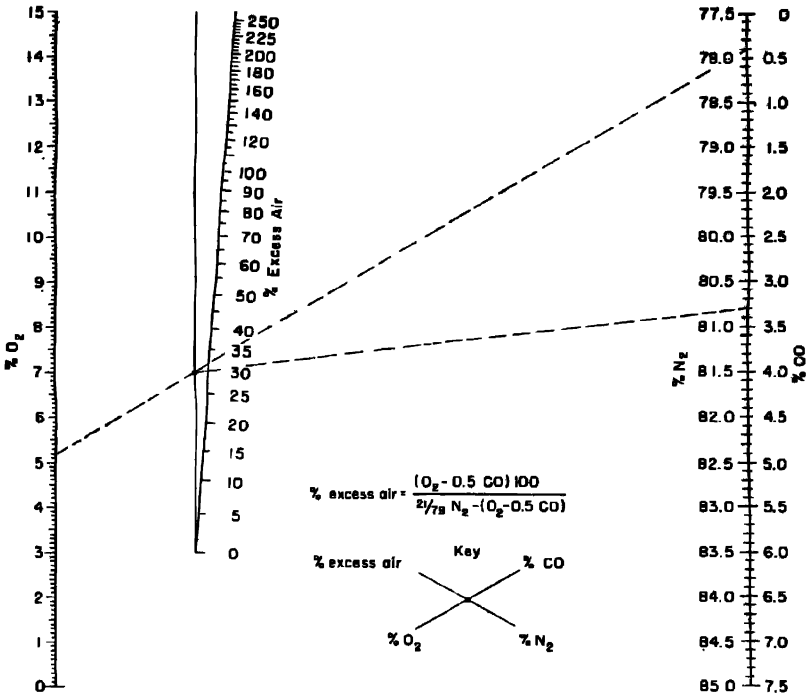


Fig. 29-100. Excess air from stack gas analysis.

4. Excess air from stack gas analysis (Fig. 29-100):

$$\text{Per cent excess air} = \frac{(O_2 - 0.5CO)100}{{}^{21}\!/_{79}N_2 - (O_2 - 0.5CO)}$$

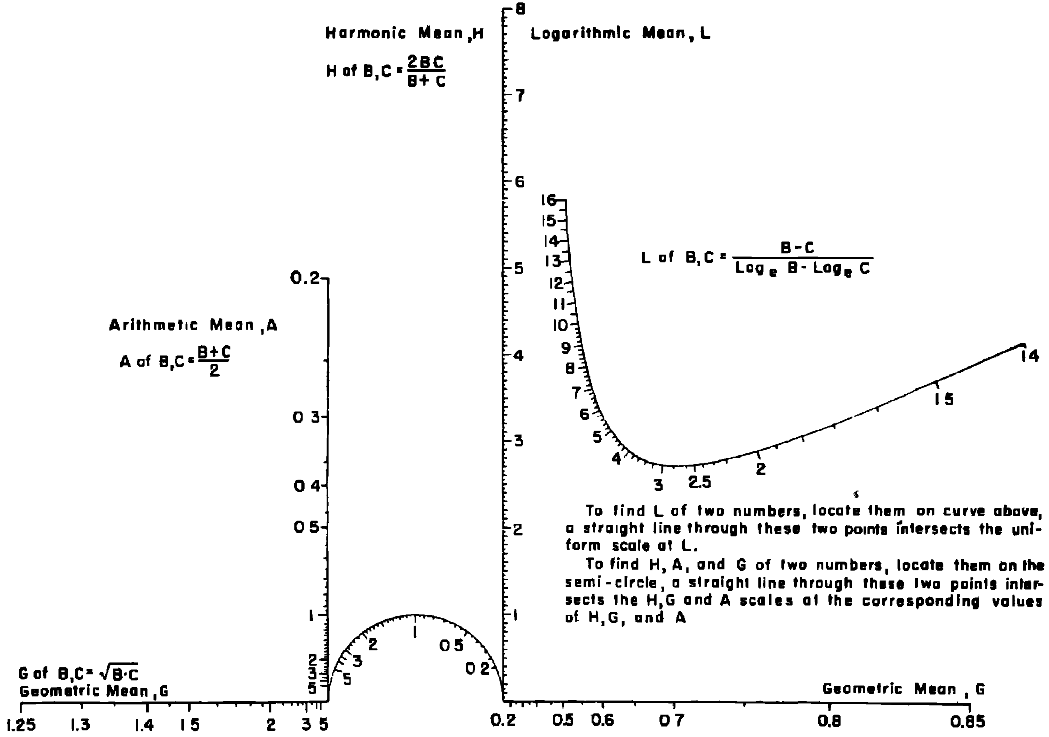

Harmonic Mean, H
H of B,C = $\frac{2BC}{B+C}$
Logarithmic Mean, L
L of B,C = $\frac{B-C}{\text{Log}_e B - \text{Log}_e C}$
Arithmetic Mean, A
A of B,C = $\frac{B+C}{2}$
G of B,C = $\sqrt{B \cdot C}$
Geometric Mean, G
Geometric Mean, G
1 2 3 4 5 6 7 8
16 15 14 13 12 11 10 9 8 7 6 5 4 3 2.5 2 1.5 1.4
0.2 0.3 0.4 0.5 1 2 3 5
5 3 2 1 0.5 0.2
1.25 1.3 1.4 1.5 2 3 5
0.2 0.5 0.6 0.7 0.8 0.85
To find L of two numbers, locate them on curve above, a straight line through these two points intersects the uniform scale at L.
To find H, A, and G of two numbers, locate them on the semi-circle, a straight line through these two points intersects the H, G and A scales at the corresponding values of H, G, and A

FIG. 29-101. Various types of means.

5. Chart for various types of mean values (Fig. 29-101)

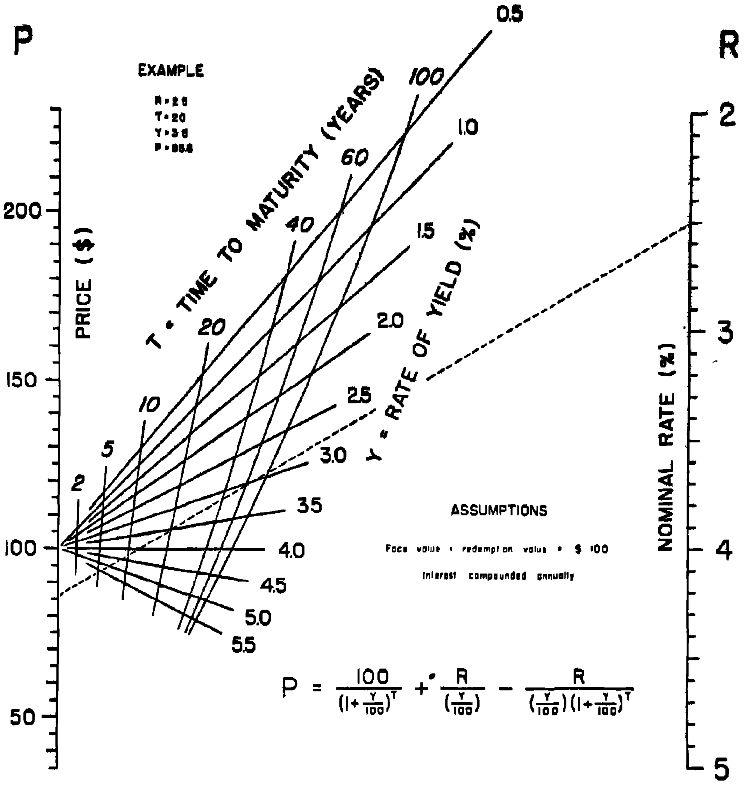


Fig. 29-102.

6. The fundamental bond equation (Fig. 29-102)

$$P = \frac{100}{\left(1 + \frac{y}{100}\right)^t} + \frac{R}{\left(\frac{y}{100}\right)} - \frac{R}{\left(\frac{y}{100}\right)\left(1 + \frac{y}{100}\right)^t}$$

7. Equation for anchored bases (Fig. 29-103)

$$Y_s^2 + \left(4d + 60\frac{M}{P}\right) Y_s = 24d\frac{M}{P} - 4d^2$$

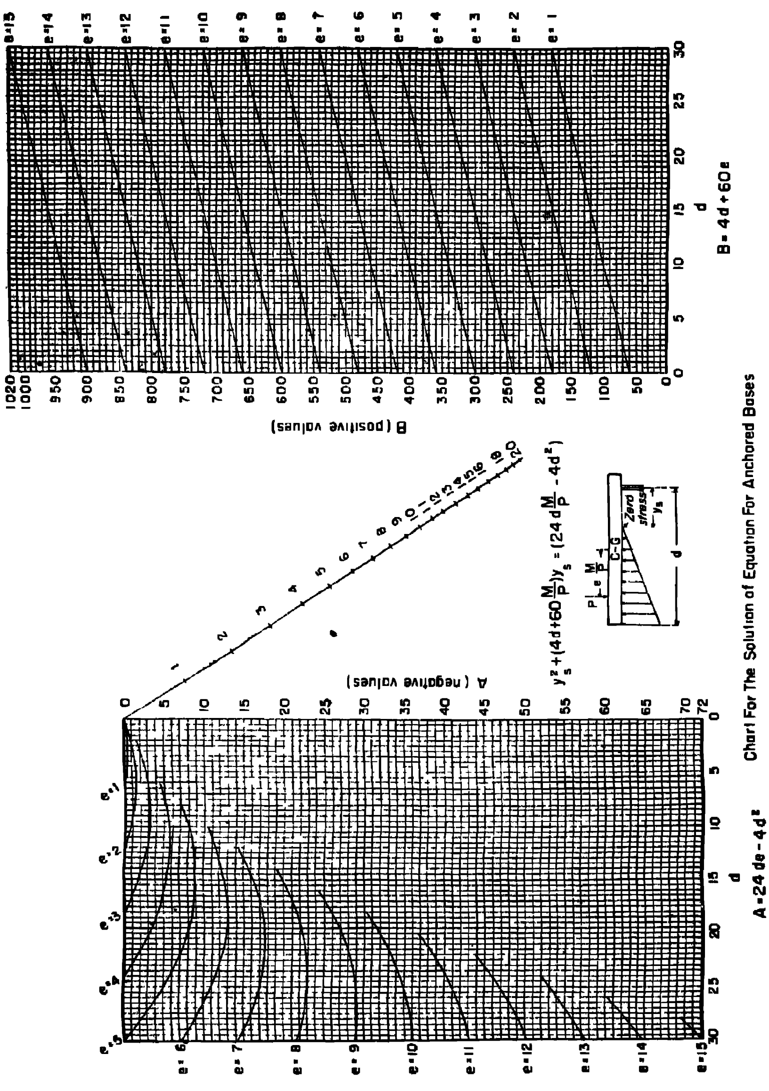


Chart For The Solution of Equation For Anchored Bases

FIG. 29-103

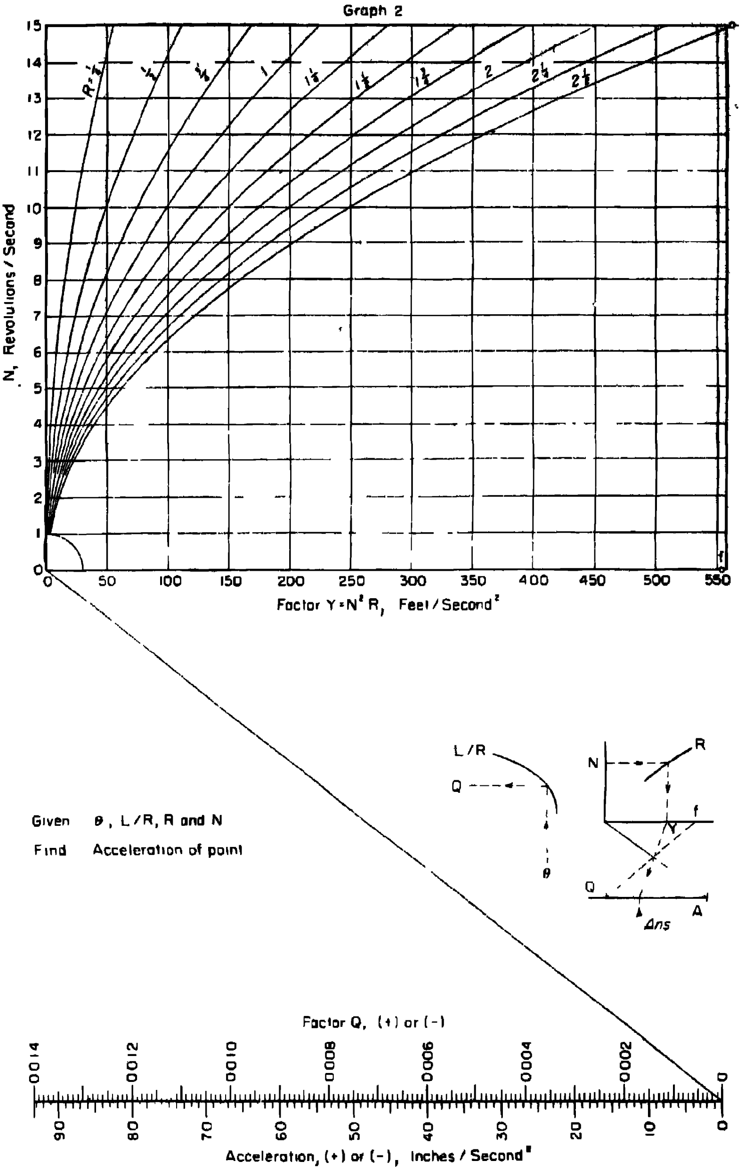


Fig. 29-104a. Crosshead acceleration.

8. Crosshead and crank-pin accelerations (Figs. 29-104a and 29-104b)

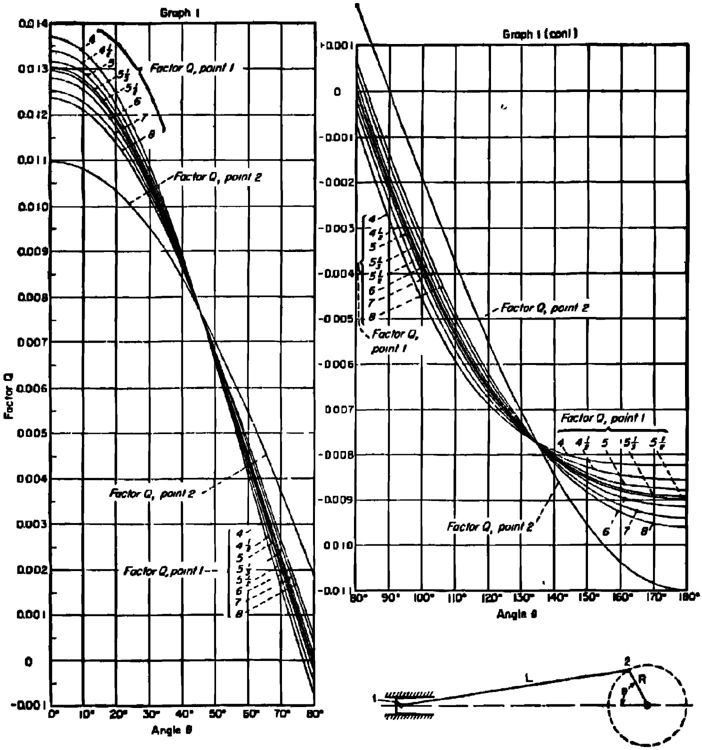


FIG. 29-104b. Crank-pin accelerations

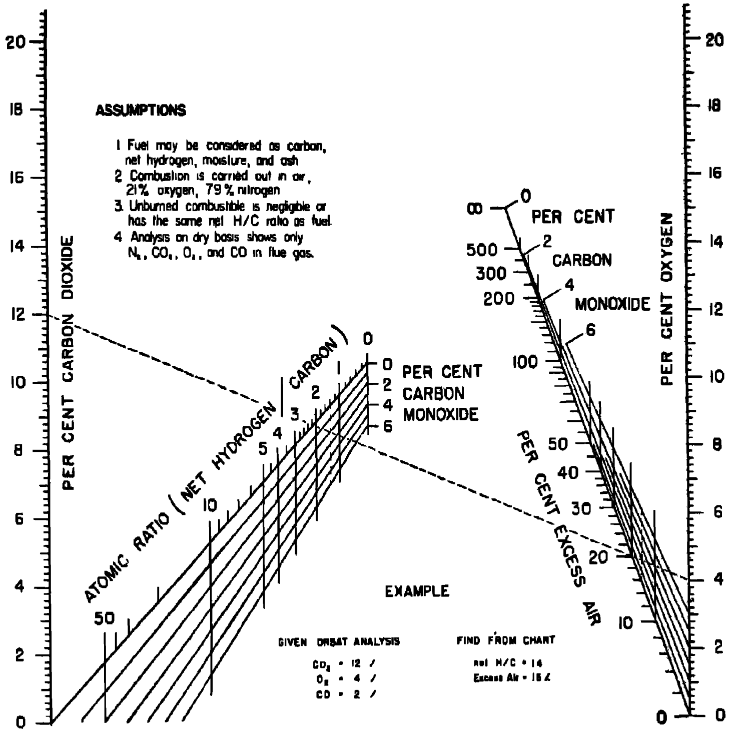

ASSUMPTIONS
1 Fuel may be considered as carbon, net hydrogen, moisture, and ash
2 Combustion is carried out in air, 21% oxygen, 79% nitrogen
3. Unburned combustible is negligible or has the same net H/C ratio as fuel.
4 Analysis on dry basis shows only N₂, CO₂, O₂, and CO in flue gas.
PER CENT CARBON DIOXIDE
20
18
16
14
12
10
8
6
4
2
0
ATOMIC RATIO (NET HYDROGEN\CARBON)
0
1
2
3
4
5
10
50
0
2
4
6
PER CENT CARBON MONOXIDE
∞
500
300
200
100
50
40
30
20
10
0
PER CENT EXCESS AIR
0
2
4
6
PER CENT CARBON MONOXIDE
PER CENT OXYGEN
20
18
16
14
12
10
8
6
4
2
0
EXAMPLE
GIVEN ORSAT ANALYSIS
CO₂ = 12 %
O₂ = 4 %
CO = 2 %
FIND FROM CHART
net H/C = 14
Excess Air = 15 %

FIG. 29-105. Stoichiometry of combustion.

9. Stoichiometry of combustion (Fig. 29-105)

Sets of Charts. Since it is possible to represent an equation graphically in numerous ways, a few sets are included in order that one may make comparisons as to usefulness or ease of construction (see Figs. 29-106 to 29-110).

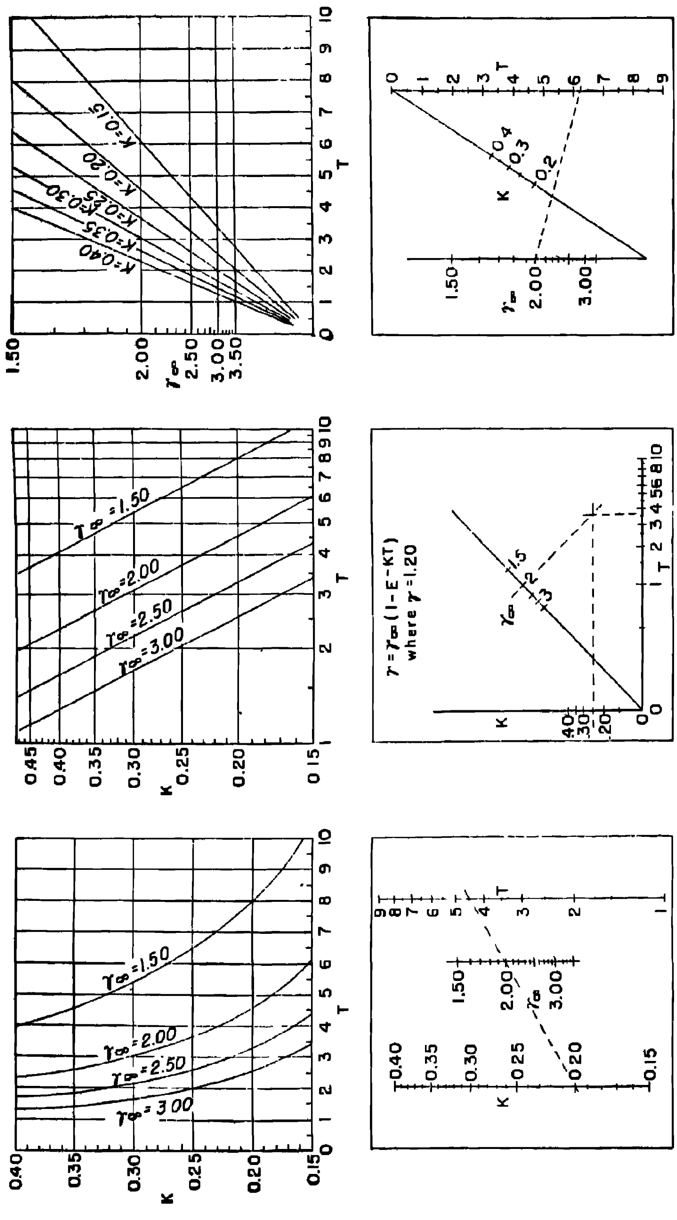

K=0.15
K=0.20
K=0.25
K=0.30
K=0.35
K=0.40
γ∞
T
γ∞=1.50
γ∞=2.00
γ∞=2.50
γ∞=3.00
K
γ = γ∞(1-E^-KT)
where γ=1.20

FIG. 29-106.

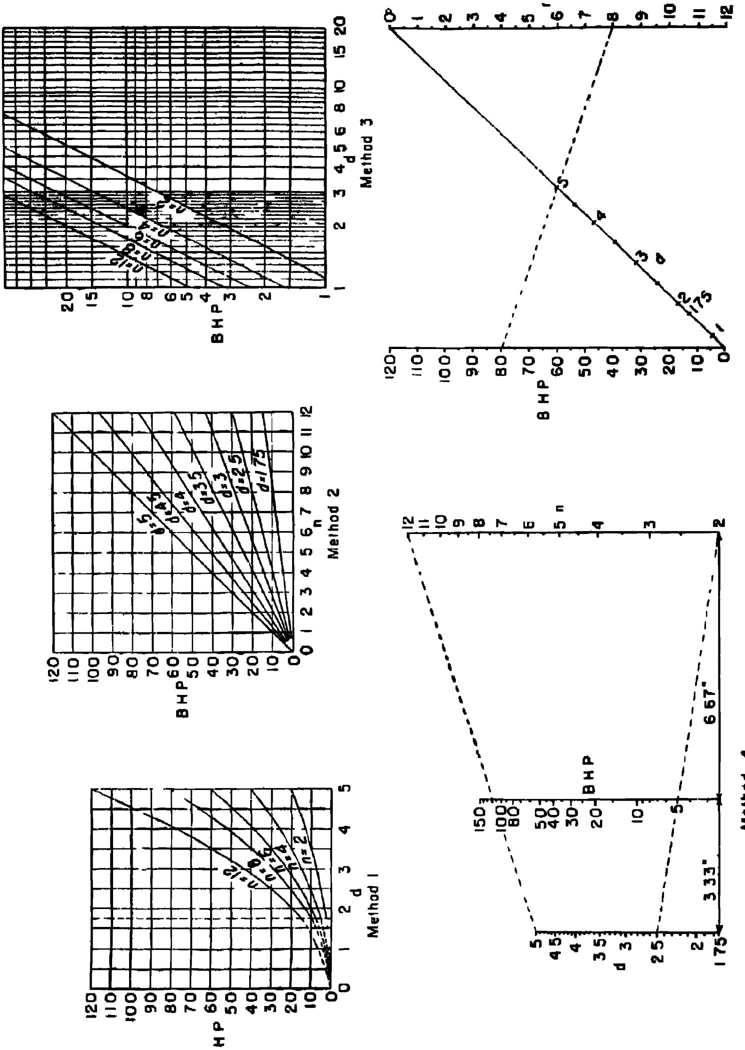


Fig. 29-107. Bhp = $d^2n/2.5$.

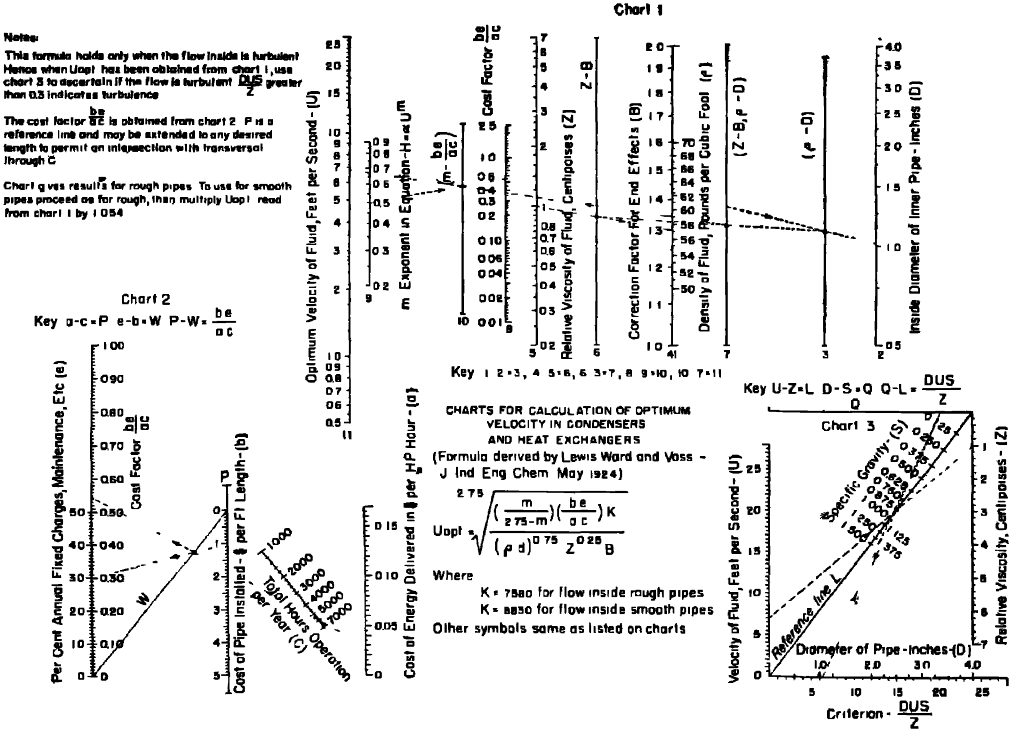


Chart 1
Optimum Velocity of Fluid, Feet per Second - (U)
m Exponent in Equation - H = αU^m
(m - be/ac)
Cost Factor be/ac
Relative Viscosity of Fluid, Centipoises (Z)
Z - B
Correction Factor for End Effects (B)
Density of Fluid, Pounds per Cubic Foot (ρ)
(Z - B, ρ - D)
(ρ - D)
Inside Diameter of Inner Pipe - Inches (D)
Key 1 2=3, 4 5=6, 6 3=7, 8 9=10, 10 7=11
Chart 2
Key a-c=P e-b=W P-W= be/ac
Per Cent Annual Fixed Charges, Maintenance, Etc (e)
Cost Factor be/ac
W
P
Cost of Pipe Installed - $ per Ft Length - (b)
Total Hours Operation per Year (C)
Cost of Energy Delivered in $ per HP Hour - (a)
Chart 3
Key U-Z=L D-S=Q Q-L= DUS/Z
Velocity of Fluid, Feet per Second - (U)
Reference line L
Specific Gravity - (S)
Diameter of Pipe - Inches - (D)
Relative Viscosity, Centipoises - (Z)
Criterion - DUS/Z

Notes:

This formula holds only when the flow inside is turbulent. Hence when Uopt has been obtained from chart 1, use chart 3 to ascertain if the flow is turbulent. $\frac{DUS}{Z}$ greater than 0.3 indicates turbulence.

The cost factor $\frac{be}{ac}$ is obtained from chart 2. P is a reference line and may be extended to any desired length to permit an intersection with transversal through C.

Chart gives results for rough pipes. To use for smooth pipes proceed as for rough, then multiply Uopt read from chart 1 by 1.054.

CHARTS FOR CALCULATION OF OPTIMUM VELOCITY IN CONDENSERS AND HEAT EXCHANGERS

(Formula derived by Lewis Ward and Voss - J Ind Eng Chem May 1924)

$$U_{opt} = \sqrt[2.75]{\frac{\left(\frac{m}{2.75-m}\right)\left(\frac{be}{ac}\right)K}{(\rho d)^{0.75} Z^{0.25} B}}$$

Where

K = 7580 for flow inside rough pipes

K = 8850 for flow inside smooth pipes

Other symbols same as listed on charts

Fig 29-108

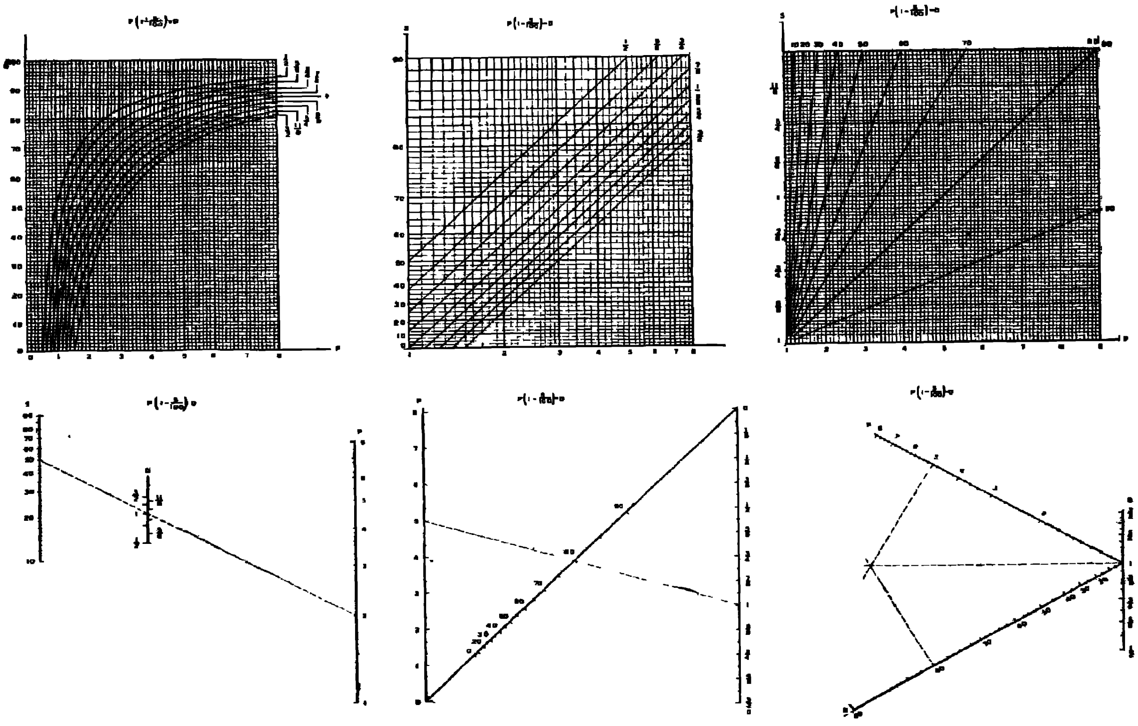


FIG. 29-109.

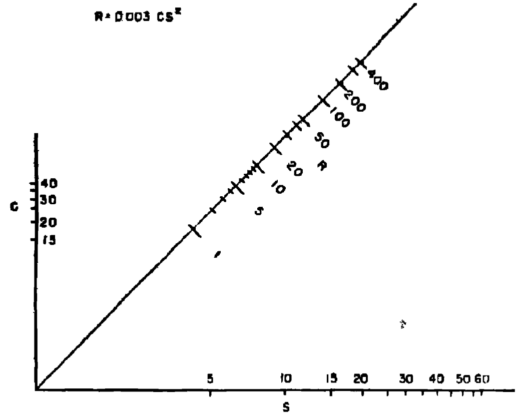


Fig. 29-110a.

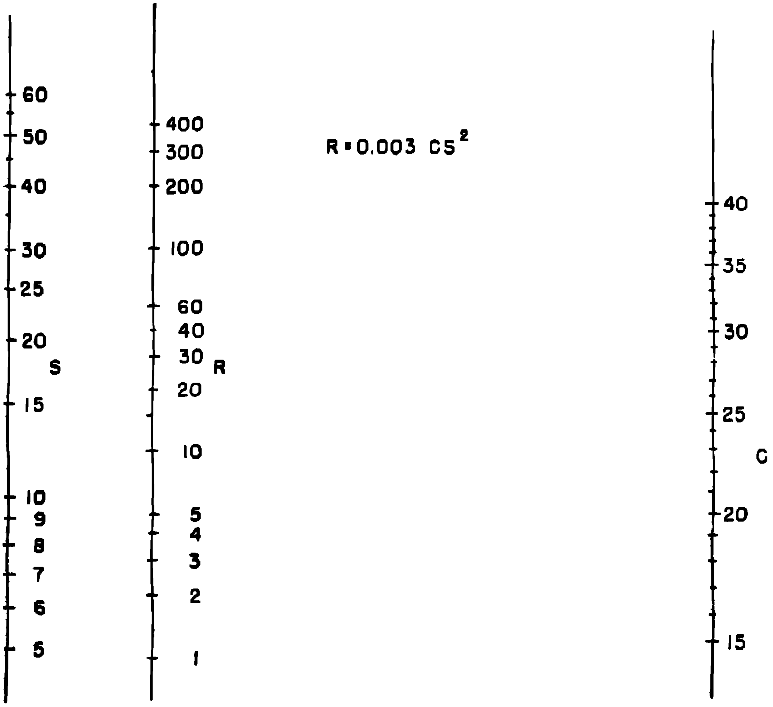


Fig 29-110b.

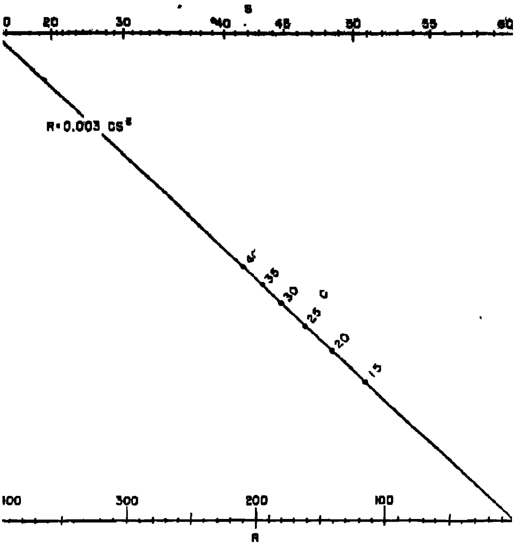


FIG. 29-110c.

DETERMINANT AS THE BASIS OF A NOMOGRAM

The condition that any three points (x_1, y_1), (x_2, y_2), and (x_3, y_3) lie on a straight line is that the determinant

$$\begin{vmatrix} x_1 & y_1 & 1 \\ x_2 & y_2 & 1 \\ x_3 & y_3 & 1 \end{vmatrix} = 0$$

In the case of an equation containing three variables u, v, and w, it is possible to express the equation in the form of the above determinant. It is also possible to construct a nomogram.

For example, the determinant

$$\begin{vmatrix} f_1(u) & g_1(u) & 1 \\ f_2(v) & g_2(v) & 1 \\ f_3(w) & g_3(w) & 1 \end{vmatrix} = 0$$

states that the three axes carry the scales

$$x_1 = f_1(u) \qquad y_1 = g_1(u) \tag{1}$$
$$x_2 = f_2(v) \qquad y_2 = g_2(v) \tag{2}$$
$$x_3 = f_3(w) \qquad y_3 = g_3(w) \tag{3}$$

and that values of u, v, and w satisfying the determinant equation lie on a straight line.

Here Eq. (1) is the parametric representation of a curve on which are marked the values of u at each point having the coordinates (x_1, y_1). Similarly, Eqs. (2) and (3) are the curves on which are marked the values of v and w, respectively.

The determinant

$$\begin{matrix} 2u & 1| \\ v-1 & 0 \end{matrix}$$

is one way of writing the equation

$$w(1 + 2u - v) + w^2(v - u) - u(v + 1) = 0$$

The equation of the curve on which the u scale is marked is $x = u$, $y = 2u$, in parametric form, or the straight line $y = 2x$ in rectangular coordinates.

The equation of the curve on which the v scale is marked is $x = v$, $y = v - 1$, in parametric form, or the straight line $y = x - 1$ in rectangular coordinates.

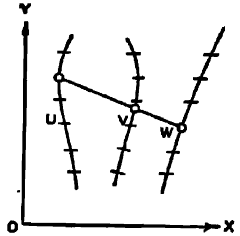


FIG. 29-111.

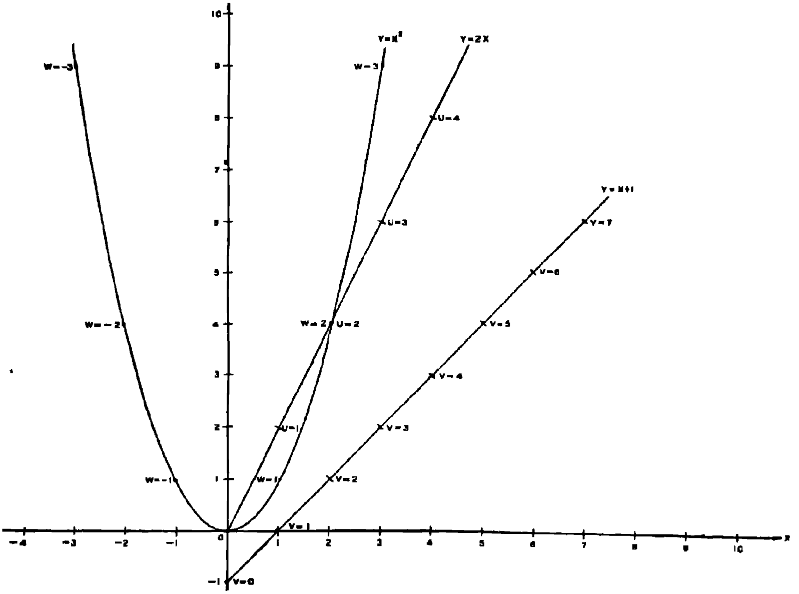


FIG. 29-112. $w(1 + 2u - v) + w^2(v - u) - u(v + 1) = 0.$

The equation of the curve on which the w scale is marked is $x = w$, $y = w^2$, in parametric form, or the parabola $y = x^2$ in rectangular coordinates. The appearance of the chart is shown in Fig. 29-112.

The general method of deriving the determinant form of an equation consists in forming three simultaneous equations of the form $Ax + By + C = 0$, from which the determinant

$$\begin{vmatrix} A_1 & B_1 & C_1 \\ A_2 & B_2 & C_2 \\ A_3 & B_3 & C_3 \end{vmatrix} = 0$$

may be derived directly by writing the coefficients of the equation in the order shown. It is then necessary to transform this determinant into the basic form where the elements occurring in one column are each equal to unity. This can be accomplished by using the well-known properties of determinants.

1. Rows may be changed into columns and columns into rows.
2. Any two rows or columns may be interchanged.
3. All the elements of a row or column may be multiplied or divided by the same quantity.
4. To the element of any row or column may be added the same multiples of the corresponding element of any one or more other rows or columns.

For example, consider the equation,

$$s = v_0t - 16t^2$$

Rewrite the equation in the form,

$$w = vu - 16u^2 \qquad \text{or} \qquad 16u^2 - uv + w = 0$$

Let $x = v$ and $y = w$; then the following three equations are obtained.

$$16u^2 - xu + y = 0$$
$$x - v = 0$$

and

$$y - w = 0$$

Each of these equations is of the form

$$Ax + By + C = 0$$

and may be written as

$$x + 0y - v = 0$$
$$0x + y - w = 0$$

and

$$-ux + y + 16u^2 = 0$$

Since the equations are simultaneously true, the following determinant

$$\begin{vmatrix} 1 & 0 & -v \\ 0 & 1 & -w \\ -u & 1 & 16u^2 \end{vmatrix} = 0$$

A new determinant with a column of ones must be found, using the properties of determinants. This may be done as follows:

1. Adding the elements of the first and second columns, we obtain

$$\begin{vmatrix} 1 & 0 & -v \\ 1 & 1 & -w \\ 1-u & 1 & 16u^2 \end{vmatrix} = 0$$

2. Dividing the third row by $1 - u$ and interchanging the first and third columns, we obtain

$$\begin{array}{cc} -v & 0 \\ -w & 1 \\ 16u^2 & 1 \\ u & \end{array} = 0$$

which is a basic form.

3. For convenience interchange the first and second columns. Then

$$\begin{array}{ccc} 0 & -v & 1 \\ 1 & -w & 1 \\ \dfrac{1}{1-u} & \dfrac{16u^2}{1-u} & 1 \end{array}$$

The equations of the scales are

$$\begin{array}{ll} x & -v \\ x & -w \\ x = \quad 1-u & \dfrac{16u^2}{1-u} \end{array}$$

Scale factors may be introduced in a variety of ways. One way is to multiply the first and second columns by d and m, respectively. The form of the determinant is

$$\begin{array}{ccc} 0 & -mv & 1 \\ d & -mw & 1 \\ d & 16mu^2 & , \\ 1-u & 1 & \cdot\, u \end{array}$$

where m and d are determined by the range of the variables and the desired size of the chart.

Another way is to introduce scale factors at the beginning.

Let $x = m_1v$ and $y = m_2w$ in the equation $16u^2 - uv + w = 0$. Then the following three equations are obtained:

$$16u^2 - (x/m_1)u + (y/m_2) = 0$$
$$x - m_1v = 0$$

and

$$y - m_2w = 0$$

which may be written

$$-(u/m_1)x + (1/m_2)y + 16u^2 = 0$$
$$x + 0y - m_1v = 0$$
$$0x + y - m_2w = 0$$

Therefore

$$\begin{vmatrix} -\dfrac{u}{m_1} & \dfrac{1}{m_2} & 16u^2 \\ 1 & 0 & -m_1v \\ 0 & 1 & -m_2w \end{vmatrix} = 0$$

By properties of determinants, the resulting determinant is obtained:

$$\begin{vmatrix} 0 & -m_1v & 1 \\ 1 & -m_2w & 1 \\ \dfrac{m_1}{-m_2u + m_1} & \dfrac{16m_1m_2u^2}{-m_2u + m_1} & 1 \end{vmatrix} = 0$$

In order to control distances between scales, multiply the elements of the first column by d. Thus, the constructional form of the determinant is

$$\begin{vmatrix} 0 & -m_1v & 1 \\ d & -m_2w & 1 \\ \dfrac{m_1d}{-m_2u + m_1} & \dfrac{16m_1m_2u^2}{-m_2u + m_1} & 1 \end{vmatrix} = 0$$

A check on the work at any time can be made by expanding the determinant at that stage of the work, and the resulting equation will be the original one. Thus a check on the final form arrived at is

$$\frac{-m_1^2dv}{-m_2u + m_1} + \frac{16m_1m_2du^2}{-m_2u + m_1} + \frac{m_1m_2dw}{-m_2u + m_1} + m_1dv = 0$$

or

$$16u^2 - uv + w = 0$$

Consider a second equation of the form

$$\frac{1}{f_2(v)} + \frac{1}{f_4(t)} = \frac{1}{f_1(u)}$$

or

$$f_1(u) \cdot f_4(t) + f_1(u) \cdot f_2(v) = f_2(v) \cdot f_4(t)$$

previously discussed

The corresponding determinant form is

$$\begin{vmatrix} f_2(v) & 0 & 1 \\ f_1(u) & f_1(u) & 1 \\ 0 & f_4(t) & 1 \end{vmatrix} = 0$$

The constructional determinant can be conveniently written

$$\begin{vmatrix} m_1f_2(v) & 0 & 1 \\ m_1f_1(u) & m_2f_1(u) & 1 \\ 0 & m_2f_4(t) & 1 \end{vmatrix} = 0$$

The scales are

$$x = m_1f_2(v) \qquad y = 0$$
$$x = m_1f_1(u) \qquad y = m_2f_1(u)$$
and
$$x = 0 \qquad y = m_2f_4(t)$$

It should be noted that, if $m_1 = m_2$, the axis carrying the u scale bisects the angle between the axes.

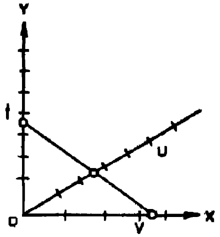


Fig. 29-113. $\frac{1}{f_2(v)} + \frac{1}{f_4(t)} = \frac{1}{f_1(u)}$.

Common Types. The basic determinant together with its constructional determinant and form of chart is shown for common types of equations occurring in practice:

1. $f_1(u) + f_2(v) = f_3(w)$

$$\begin{vmatrix} -1 & f_1(u) & 1 \\ 1 & f_2(v) & 1 \\ 0 & \tfrac{1}{2}f_3(w) & 1 \end{vmatrix} = 0 \qquad \begin{vmatrix} -d_1 & m_1f_1(u) & 1 \\ d_2 & m_2f_2(v) & 1 \\ 0 & \dfrac{m_1m_2}{m_1 + m_2}f_3(w) & 1 \end{vmatrix} = 0$$

where $m_1/m_2 = d_1/d_2$.

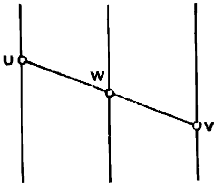


Fig. 29-114.

2. $f_1(u) + f_2(v) \cdot f_3(w) - kf_2(v) - f_1(u) \cdot f_2(v) = 0$

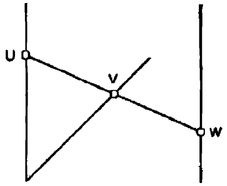


Fig. 29-115.

$$\begin{vmatrix} 0 & f_1(u) & 1 \\ f_2(v) & kf_2(v) & 1 \\ 1 & f_3(w) & 1 \end{vmatrix} = 0 \qquad \begin{vmatrix} 0 & m_1f_1(u) & 1 \\ \dfrac{dm_1f_2(v)}{(m_1 - m_2)f_2(v) + m_2} & \dfrac{m_1m_2kf_2(v)}{(m_1 - m_2)f_2(v) + m_2} & 1 \\ d & m_2f_3(w) & 1 \end{vmatrix} = 0$$

$$f_3(w) = \frac{f_1(u)}{f_1(u) + f_2(v)}$$

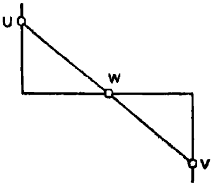


Fig. 29-116.

$$\begin{vmatrix} 0 & f_1(u) & 1 \\ f_3(w) & 0 & 1 \\ 1 & -f_2(v) & 1 \end{vmatrix} = 0 \qquad \begin{vmatrix} 0 & m_1f_1(u) & 1 \\ \dfrac{dm_1f_3(w)}{(m_1 - m_2)f_3(w) + m_2} & 0 & 1 \\ d & -m_2f_2(v) & 1 \end{vmatrix} = 0$$

4. $$f_3(w) = \frac{f_1(u)}{f_2(v)}$$

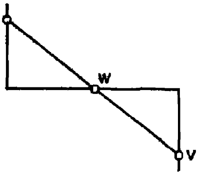


FIG. 29-117.

$$\begin{vmatrix} 0 & f_1(u) & 1 \\ \dfrac{f_3(w)}{1 + f_3(w)} & 0 & 1 \\ 1 & -f_2(v) & 1 \end{vmatrix} = 0 \qquad \begin{vmatrix} 0 & m_1 f_1(u) & 1 \\ \dfrac{dm_1 f_3(w)}{m_1 f_3(w) + m_2} & 0 & 1 \\ d & -m_2 f_2(v) & 1 \end{vmatrix} = 0$$

5. $$f_3(w) = \frac{f_1(u) \cdot f_2(v)}{f_1(u) + f_2(v)}$$

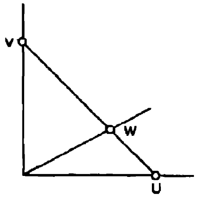


FIG. 29-118.

$$\begin{vmatrix} f_1(u) & 0 & 1 \\ 0 & f_2(v) & 1 \\ f_3(w) & f_3(w) & 1 \end{vmatrix} = 0 \qquad \begin{vmatrix} df_1(u) & 0 & 1 \\ 0 & mf_2(v) & 1 \\ df_3(w) & mf_3(w) & 1 \end{vmatrix} = 0$$

6. $$f_1(u) - f_2(v) + \phi(v)[f_3(w) - f_1(u)] = 0$$

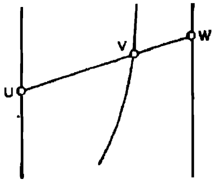


FIG. 29-119.

$$\begin{vmatrix} 0 & f_1(u) & 1 \\ \phi(v) & f_2(v) & 1 \\ 1 & f_3(w) & 1 \end{vmatrix} = 0 \qquad \begin{vmatrix} 0 & m_1 f_1(u) & 1 \\ \dfrac{km_1\phi(v)}{(m_1 - m_2)\phi(v) + m_2} & \dfrac{m_1 m_2 f_2(v)}{(m_1 - m_2)\phi(v) + m_2} & 1 \\ K & m_2 f_3(w) & 1 \end{vmatrix} = 0$$

7. $$f_1(u) \cdot f_2(v) + \phi(v) \cdot f_3(w) - f_3(w) \cdot f_1(u) = 0$$

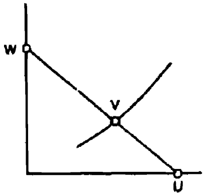


FIG. 29-120.

$$\begin{vmatrix} f_1(u) & 0 & 1 \\ \phi(v) & f_2(v) & 1 \\ 0 & f_3(w) & 1 \end{vmatrix} = 0 \qquad \begin{vmatrix} df_1(u) & 0 & 1 \\ d\phi(v) & mf_2(v) & 1 \\ 0 & mf_3(w) & 1 \end{vmatrix} = 0$$

8. $$f_3(w) = \frac{f_1(u)\cdot\phi(v) - \phi(u)\cdot f_2(v)}{\phi(v) - \phi(u)}$$

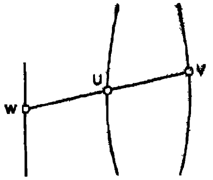


FIG. 29-121.

$$\begin{vmatrix} \phi(u) & f_1(u) & 1 \\ \phi(v) & f_2(v) & 1 \\ 0 & f_3(w) & 1 \end{vmatrix} = 0 \qquad \begin{vmatrix} k\phi(u) & mf_1(u) & 1 \\ k\phi(v) & mf_2(v) & 1 \\ 0 & mf_3(w) & 1 \end{vmatrix} = 0$$

9. $$\phi_1(u)[f_2(v) - f_3(w)] + \phi_2(v)[f_3(w) - f_1(u)] + \phi_3(w)[f_1(u) - f_2(v)] = 0$$

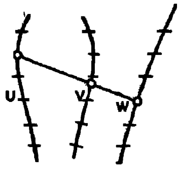


FIG. 29-122.

$$\begin{vmatrix} \phi_1(u) & f_1(u) & 1 \\ \phi_2(v) & f_2(v) & 1 \\ \phi_3(w) & f_3(w) & 1 \end{vmatrix} = 0 \qquad \begin{vmatrix} m_1\phi_1(u) & m_2f_1(u) & 1 \\ m_1\phi_2(v) & m_2f_2(v) & 1 \\ m_1\phi_3(w) & m_2f_3(w) & 1 \end{vmatrix} = 0$$

Specific Equations Expressed in Determinant Form. As illustrations of equations expressed in the form of determinants are the following:

1. Flow of steam through pipes:

$$W = 15\pi d^2VD$$

$$\begin{vmatrix} \dfrac{15\pi d^2}{W} & 1 & 1 \\ 0 & \dfrac{V}{1+V} & 1 \\ -\dfrac{1}{D} & 0 & 1 \end{vmatrix} = 0 \quad \text{or} \quad \begin{vmatrix} \dfrac{15\pi d^2}{W} & 1 & 1 \\ 0 & \dfrac{D}{1+D} & 1 \\ -\dfrac{1}{V} & 0 & 1 \end{vmatrix} = 0$$

2. Boiler horsepower:

$$\text{Boiler hp} = 3.33(A - 0.6\sqrt{A})\sqrt{H}$$

$$\begin{vmatrix} \dfrac{10A}{3(1+2\sqrt{A})} & \dfrac{2\sqrt{A}}{1+2\sqrt{A}} & 1 \\ \dfrac{B}{\sqrt{H}} & 0 & 1 \\ 1 & 1 & 1 \end{vmatrix} = 0$$

3. Shear strength of rivets:

$$\gamma_s = \frac{\pi}{4}d^2f_s$$

$$\begin{vmatrix} \dfrac{\pi d^2}{4} & -1 & 1 \\ 0 & \dfrac{f_s}{1-f_s} & 1 \\ \gamma_s & 0 & 1 \end{vmatrix} = 0$$

4. Radius of gyration of circular cylinder:

$$K \qquad \sqrt{3R^2 + h^2}$$

$$--$$

$$\begin{vmatrix} 3R^2 & 0 & 1 \\ h^2 & 1 & 1 \\ 6K^2 & 1/2 & 1 \end{vmatrix} = 0$$

5. Capacitance of two parallel cylinders:

$$C = \frac{10^{-5}K}{36 \ln (d/\gamma)}$$

$$\begin{vmatrix} 36(10)^5C & 1 & 1 \\ 0 & \dfrac{\ln d}{1 + \ln d} & 1 \\ \dfrac{1}{1 + \ln \gamma} & \dfrac{\ln \gamma}{1 + \ln \gamma} & 1 \end{vmatrix} = 0$$

6. Blast-pressure furnace:

$$P^2 - 14.7^2 = 0.0007(w^2H/D^4)$$

$$\begin{vmatrix} \dfrac{0.0007w^2}{0.0007w^2 - 1} & 0 & 1 \\ 0 & D^4(P^2 - 14.7^2) & 1 \\ 1 & -H & 1 \end{vmatrix} = 0$$

7. Brake horsepower:

$$\text{Bhp} = d^2n/2.5$$

$$\begin{vmatrix} 2.5 \text{ bhp} & 1 & 1 \\ d^2 & 0 & 1 \\ 0 & \dfrac{1}{1 - m} & 1 \end{vmatrix} = 0$$

8. Francis weir formula:

$$Q = 3.33(B - 0.2H)H^{3/2}$$

$$\begin{vmatrix} \dfrac{H^{3/2}}{H^{3/2} + 5} & \dfrac{H^{5/2}}{H^{5/2} + 5} & 1 \\ \dfrac{1}{5B} & 1 & 1 \\ 0 & \dfrac{Q}{Q - 3.33} & 1 \end{vmatrix} = 0$$

9. Lens formula:

$$\frac{1}{f} = \frac{1}{u} + \frac{1}{V}$$

$$\begin{vmatrix} \dfrac{1}{u} & 1 & 1 \\ \dfrac{1}{V} & -1 & 1 \\ \dfrac{1}{2f} & 0 & 1 \end{vmatrix} = 0$$

10. Moment of inertia of a cylinder:

$$I = \frac{W}{12}(3R^2 + h^2)$$

$$\begin{vmatrix} \dfrac{W}{12} & 1 & 1 \\ 0 & \dfrac{R^2}{R^2 - 1} & 1 \\ \dfrac{I}{h^2 + 3} & \dfrac{h^2}{h^2 + 3} & 1 \end{vmatrix} = 0$$

11. Cubic equation:

$$w^3 + nw^2 + Pw + q = 0$$

$$\begin{vmatrix} \dfrac{w}{w+1} & -\dfrac{w^3+nw^2}{w+1} & 1 \\ 0 & q & 1 \\ 1 & P & 1 \end{vmatrix} = 0$$

Examples:

1. Chordal height and length of chord:

$$d^2 = 8rh - 4h^2$$

Expressed in determinant form, the equation is

$$\begin{vmatrix} 0 & \dfrac{d^2}{40} & 1 \\ 11\ 2 & -r & 1 \\ \dfrac{22\ 4h}{2h+10} & -\dfrac{h^2}{2h+10} & 1 \end{vmatrix} = 0$$

The chart is shown in Fig. 29-123.

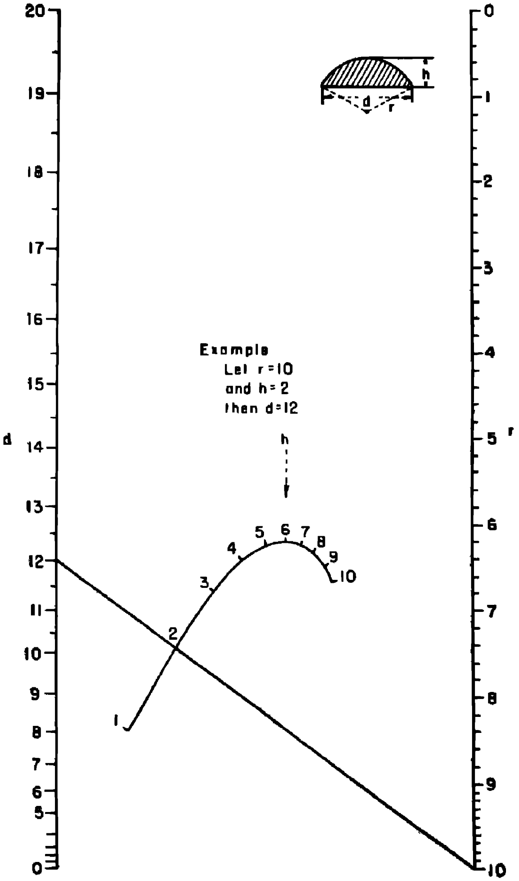


FIG. 29-123. $d^2 = 8rh - 4h^2$.

2.
$$A = \frac{4}{3} H^2 \sqrt{\frac{2R}{H} - 0.608}$$

The determinant form of the equation is

$$\begin{vmatrix} 0 & \dfrac{A^2}{180{,}000} & \\ 5 & -\dfrac{4R}{125} & 1 \\ \dfrac{5H^3}{H^3 + 180} & \dfrac{1.216H^4}{125(H^3 + 180)} & 1 \end{vmatrix} = 0$$

The completed chart is shown in Fig. 29-124.

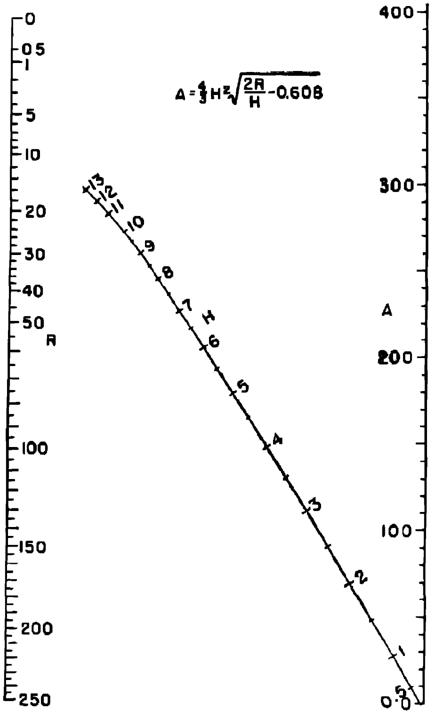


FIG 29-124.

3. Energy content for a rectangular channel with uniform flow:

$$E = D - (Q^2/2gD^2)$$

The determinant form of the equation is

$$\begin{vmatrix} 0 & \dfrac{12.2}{(100)^2} Q^2 & 1 \\ 9 & -\dfrac{E}{3} & 1 \\ \dfrac{9D^2}{D^2 + \dfrac{100^2}{73.2g}} & \dfrac{D^3}{3\left(D^2 + \dfrac{100^2}{73.2g}\right)} & 1 \end{vmatrix} = 0$$

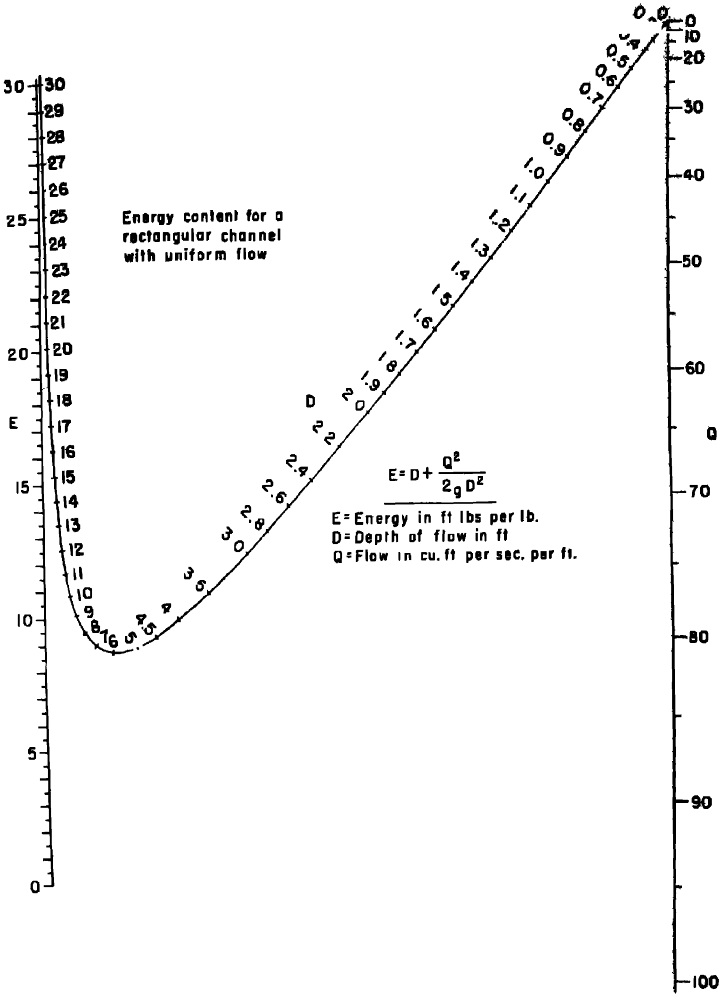


Fig. 29-125.

The completed chart is shown in Fig. 29-125.

4. Temperature in air-mixture otto cycle

$$Te^{BT} = C$$

Taking logarithms of both sides, we have

$$\log T - 0.4343BT = \log C$$

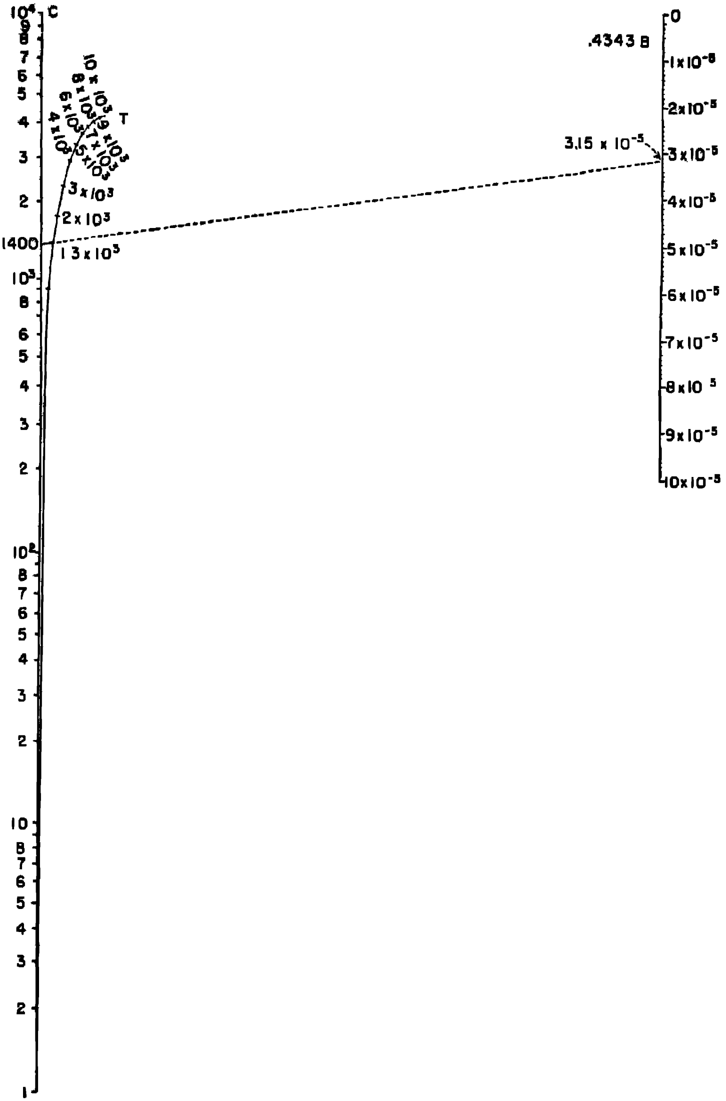


FIG. 29-126. $Te^{BT} = C$.

The determinant form and chart are shown in Fig. 29-126.

5. Mean temperature equation

$$T = \frac{T_1 - T_2}{\log_e T_1 - \log_e T_2} = \frac{T_1 - T_2}{\ln T_1 - \ln T_2}$$

Expressed in determinant form, the equation is

$$\begin{vmatrix} 0 & T & 1 \\ \dfrac{1}{\ln T_1} & \dfrac{T_1}{\ln T_1} & 1 \\ \dfrac{1}{\ln T_2} & \dfrac{T_2}{\ln T_2} & 1 \end{vmatrix} = 0$$

The chart is shown in Fig. 29-127.

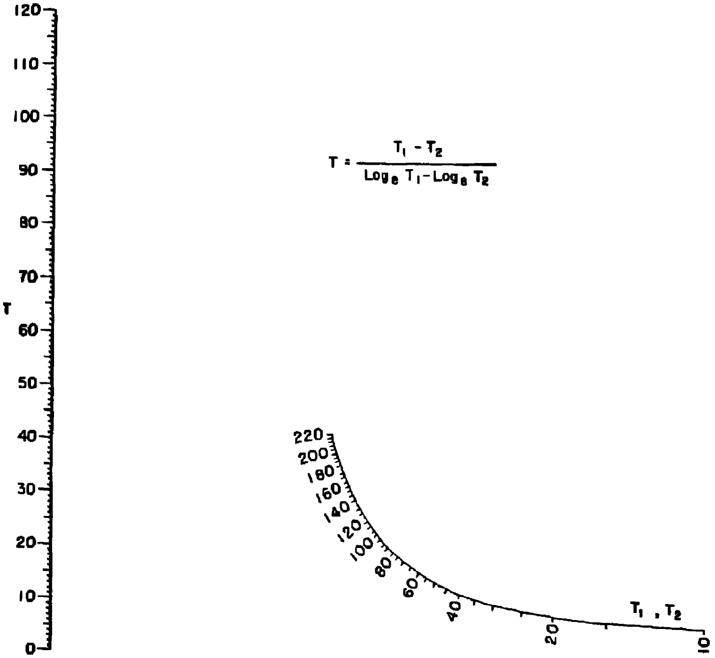


FIG. 29-127.

6. Equity debt equation

$$R = \frac{E - IP}{1 - P}$$

The determinant form of this equation is

$$\begin{vmatrix} 0 & \frac{3}{2}I & 1 \\ 12 & \frac{4}{13}R & 1 \\ \frac{468(1-P)}{39-31P} & \frac{12E}{39-31P} & 1 \end{vmatrix} = 0$$

The chart is shown in Fig. 29-128.

$$R = \frac{E - IP}{1 - P}$$

R = rate of return on equity
I = rate of interest on debt
P = proportion of debt to total capitalization
E = rate of earnings on total capitalization

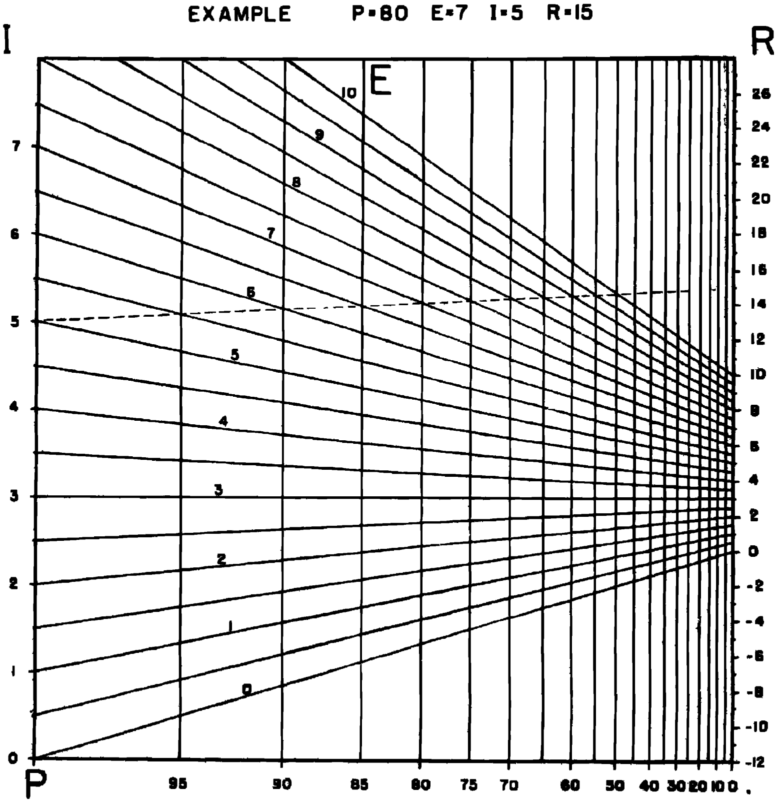


Fig. 29-128.

Circular Nomograms. The general form of the basic determinant for a circular nomogram is

$$\begin{vmatrix} \dfrac{1}{1 + [f_1(u)]^2} & \dfrac{f_1(u)}{1 + [f_1(u)]^2} & 1 \\ \dfrac{1}{1 + [f_2(v)]^2} & \dfrac{f_2(v)}{1 + [f_2(v)]^2} & 1 \\ \dfrac{1}{1 + f_3(w)} & 0 & 1 \end{vmatrix} = 0$$

from which it is seen that the u and v scales lie on circular axes having the same center and with radii equal to ½.

The constructional determinant form is

$$\begin{vmatrix} \dfrac{m_1}{1 + [f_1(u)]^2} & \dfrac{m_2 f_1(u)}{1 + [f_1(u)]^2} & 1 \\ \dfrac{m_1}{1 + [f_2(v)]^2} & \dfrac{m_2 f_2(v)}{1 + [f_2(v)]^2} & 1 \\ \dfrac{m_1}{1 + f_3(w)} & 0 & 1 \end{vmatrix} = 0$$

When $m_1 \neq m_2$, we will have arcs of ellipses instead of circles

In equations of the form $f_1(u) \cdot f_2(v) = f_3(w)$, the basic determinant is

$$\begin{vmatrix} \dfrac{1}{1 + [f_1(u)]^2} & \dfrac{f_1(u)}{1 + [f_1(u)]^2} & 1 \\ \dfrac{1}{1 + [f_2(v)]^2} & \dfrac{-f_2(v)}{1 + [f_2(v)]^2} & 1 \\ \dfrac{1}{1 + f_3(w)} & 0 & 1 \end{vmatrix} = 0$$

Consider the equation

$$AB \cos (\theta + \phi) + BU(\sin \phi - \cos \phi) + AU(\sin \theta - \cos \theta) = 0$$

The equation expressed in the form of a determinant is

$$\begin{vmatrix} A \sin \theta & A \cos \theta & 1 \\ B \cos \phi & B \sin \phi & 1 \\ U & U & 1 \end{vmatrix} = 0$$

The equations of the scales are

Circle of radius A:

$$x = A \sin \theta \qquad y = A \cos \theta$$

Circle of radius B:

$$x = B \cos \phi \qquad y = B \sin \phi$$

Straight line making an angle of 45 deg with the x-axis:

$$x = U \qquad y = U$$

Assume that $A = 5.12$ and $B = 2$.

The scale of θ is constructed on the outside, and the scale of ϕ is constructed on the inside of a circle, outside the chart, since the value of each angle is the same for each of the two basic circles along a radial line.

Consider the equation

$$AB \cos (\theta + \phi) + B(\sin \phi - \cos \phi) + A(\sin \theta - \cos \theta) = 0$$

where A, B, θ, and ϕ are variables.

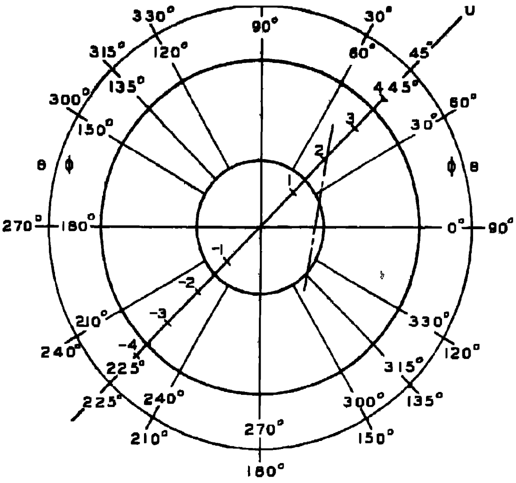


FIG. 29-129. $AB \cos(\theta + \phi) + BU(\sin\phi - \cos\phi) + AU(\sin\theta - \cos\theta) = 0$.

In determinant form, the equation is

$$\begin{vmatrix} A\sin\theta & A\cos\theta & 1 \\ B\cos\phi & B\sin\phi & 1 \\ 1 & 1 & 1 \end{vmatrix} = 0$$

The equations of the scales are

cles of radius A:

$$x = A\sin\theta \qquad y = A\cos\theta$$

cles of radius B:

$$x = B\cos\phi \qquad y = B\sin\phi$$

ed point:

$$x = 1 \qquad y = 1$$

The scales of θ and ϕ are constructed on a single circular axis as in previous stration.

Any line through the *fixed point* will cut the A and B circles in values of θ and ϕ ch satisfy the equation

$$AB\cos(\theta + \phi) + B(\sin\phi - \cos\phi) + A(\sin\theta - \cos\theta) \quad 0$$

From the construction of the chart, A and θ are paired together as are B and ϕ. For example, if $A = 5$, $\theta = 330$, and $B = 4$, find the value of ϕ.

Locate the point on the $A = 5$ circle where $\phi = 330$. (θ is graduated on the outside of the outer circle.) Lay a straightedge through this point and the fixed point. At each point where the straightedge cuts the $B = 4$ circle, draw radial lines from the common center, and read the value of ϕ on the inside of the outer circle. Two rotating dials, one at the common center and one at the fixed point, are convenient when this type of chart is used.

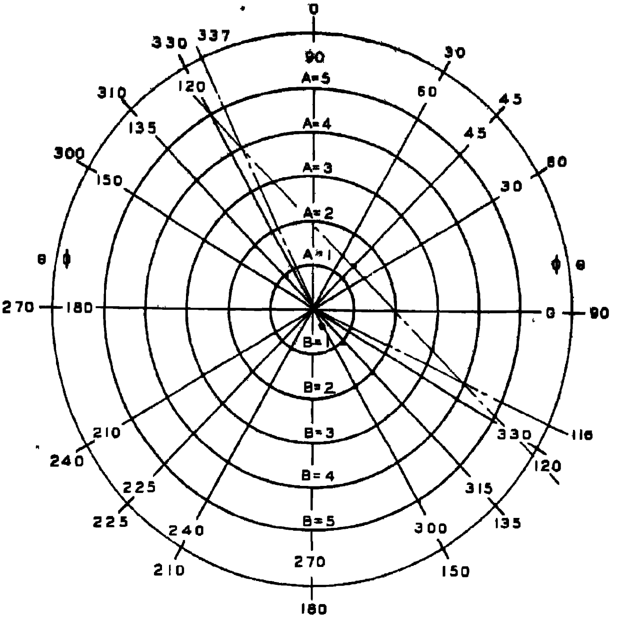


Fig. 29-130. $AB\cos(\theta+\phi) + B(\sin\phi - \cos\phi) + A(\sin\theta - \cos\theta) = 0.$

ALTERNATE GRAPHICAL METHOD

Basic Form. Another method for charting quite a general relationship between several variables can be developed from an equation of the form

$$\frac{f_1(u)}{f_2(v)} + \frac{f_3(w)}{f_4(t)} = 1 \tag{1}$$

Let OX and $O'Y$ be two parallel axes, any distance apart, and OO' a base line. Lay off the scales $m_1f_1(u)$ and $m_1f_2(v)$ on the OX axis with O as the point from which

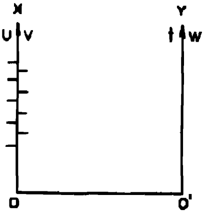


Fig. 29-131.

$$\frac{f_1(u)}{f_2(v)} + \frac{f_3(w)}{f_4(t)} = 1.$$

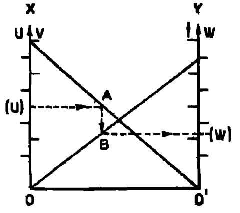


Fig. 29-132.

$$\frac{f_1(u)}{f_2(v)} + \frac{f_3(w)}{f_4(t)} = 1.$$

measurements are made. Similarly, lay off the scales $m_2f_3(w)$ and $m_2f_4(t)$ on the $O'Y$ axis measured from O' (Fig. 29-131). To find the value of one of the variables when values of the other three are given, follow the direction of the arrows in Fig. 29-132. For example, if values of u, v, and t are given, find the value of w (see Fig. 29-132).

1. Draw a line from O to t.
2. Draw a line from O' to v.

3. Draw a line from u parallel to OO' intersecting $O'v$ at A.
4. Draw a line from A parallel to the axis intersecting Ot at B.
5. Draw a line from B parallel to OO' intersecting the $O'Y$ axis at the value of w on the w scale.

It is evident that the value of any one of the variables may be found when the values of the other three are given.

$$f_1(u) + f_3(w) = f_2(v) \quad (1a)$$

If $f_2(v) = f_4(t)$, Eq. (1) takes the above form. The equations of the scales will be $mf_1(u)$, $mf_3(w)$, and $mf_2(v)$ (Fig. 29-133).

EXAMPLE: If $w = 3$ and $v = 8$, solve for u in the equation $2u + w = 10 \log v$ (Fig. 29-134). Let $m = \frac{1}{2}$. Then the scale equations will be u, $w/2$, and $5 \log v$.

The value of u is 3.01.

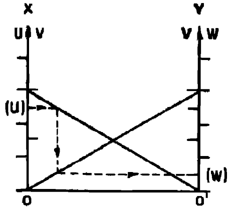


FIG. 29-133. $f_1(u) + f_3(w) = f_2(v)$.

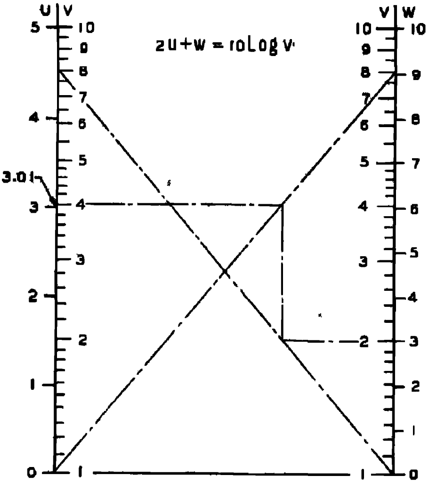


FIG. 29-134.

If lined paper is used, it is necessary only to lay off the v scale on one axis. A nomogram for this equation is shown in Fig. 29-59.

$$\frac{1}{f_2(v)} + \frac{1}{f_4(t)} = \frac{1}{f_3(w)} \qquad (1b)$$

If $f_3(w) = f_1(u)$, Eq. (1) takes this form. Since the w scale is the same on both axes, it is necessary only to lay it off on one axis.

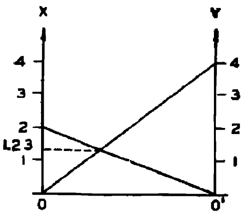


FIG. 29-135. $\frac{1}{V} + \frac{1}{T} = \frac{1}{W}$.

As an example of this type consider the equation (Fig. 29-135)

$$\frac{1}{V} + \frac{1}{T} = \frac{1}{W}$$

A chart for the equation $(1/2V) + (1/3T) = 1/U$ is shown in Fig. 29-136. When $T = 2$ and $V = 3$, $U = 3$.

A chart for the equation $[1/2(V - 1)] + [1/3(T + 1)] = 1/(U - 2)$ is shown in Fig. 29-137. When $T = 2$ and $V = 4$, $U = 5.6$.

$$\frac{f_5(R) \cdot f_1(u)}{f_2(v)} + \frac{f_6(S) \cdot f_3(w)}{f_4(t)} = 1 \tag{1c}$$

For this case $f_5(R)$ and $f_6(S)$ can be considered as scale multipliers for $f_1(u)$ and $f_3(w)$, respectively. The resulting form of chart will consist of a series of parallel scales

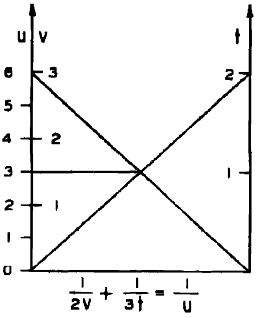


FIG. 29-136. $\frac{1}{2V} + \frac{1}{3T} = \frac{1}{U}$.

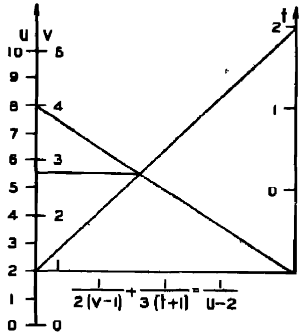


FIG. 29-137.

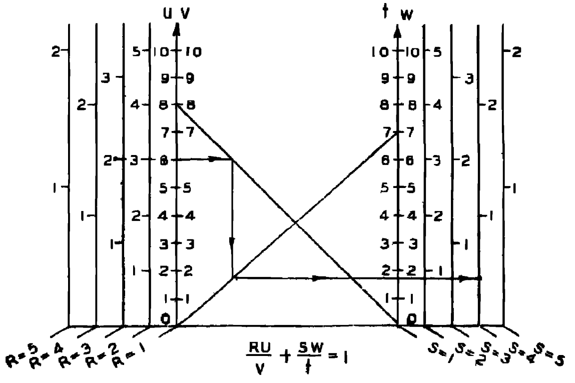


FIG 29-138

for the range of values of R and S. The series of scales may be replaced by a network. This process will be illustrated by a simple example.

Consider the equation

$$(RU/V) + (SW/T) = 1$$

On the left lay off the scales RU, at equal intervals from $R = 1, 2, 3, \ldots$, the consecutive scales being U, $2U$, $3U$, . . . (Fig. 29-138). Similarly, the scales for SW are constructed on the right-hand side. Now pass a curve (straight line in this case) through the same numbers on the scales, with a resulting network as in Fig. 29-139. When $U = 1.07$, $S = 3$, $T = 10$, $V = 8$, $R = 3$; then $W = 2$.

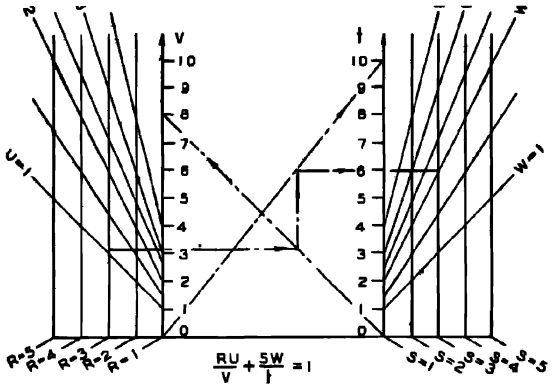


FIG. 29-139.

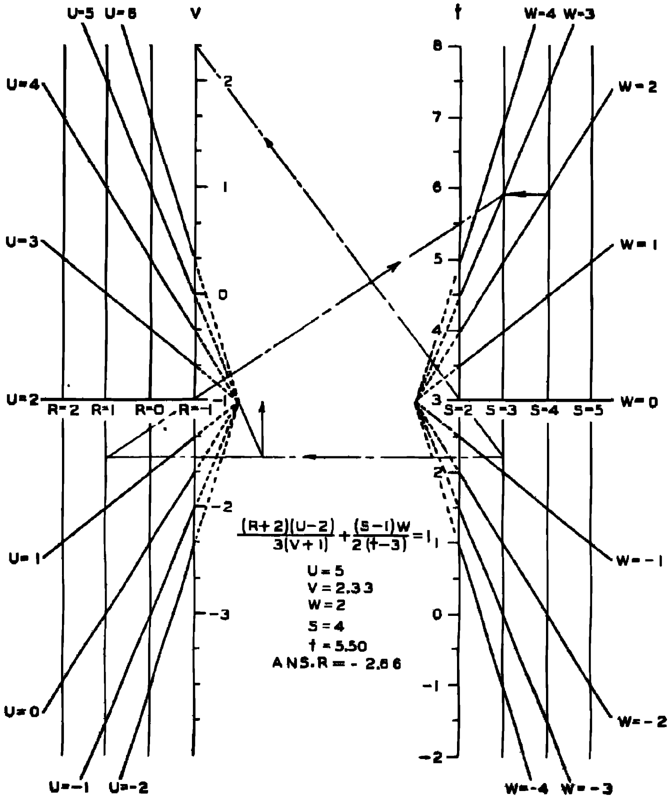


FIG. 29-140.

Two additional charts are illustrated to show the possibilities of this type of representation (Fig. 29-140):

$$\frac{(R+2)(U-2)}{3(V+1)} \qquad \frac{(S-1)W}{2(T-3)}$$

Example: Given $T = 5.5$, $V = 2.33$, $S = 4$, $W = 2$, and $U = 5$, find R.

Draw cross lines to $T = 5.5$ and $V = 2.33$. Locate point where $S = 4$ and $W = 2$. Draw horizontal line from this point to the cross line, then down to the other cross line and across until it intersects the $U = 5$ line; then project it up to the R-axis where it is estimated that $R = -2.66$.

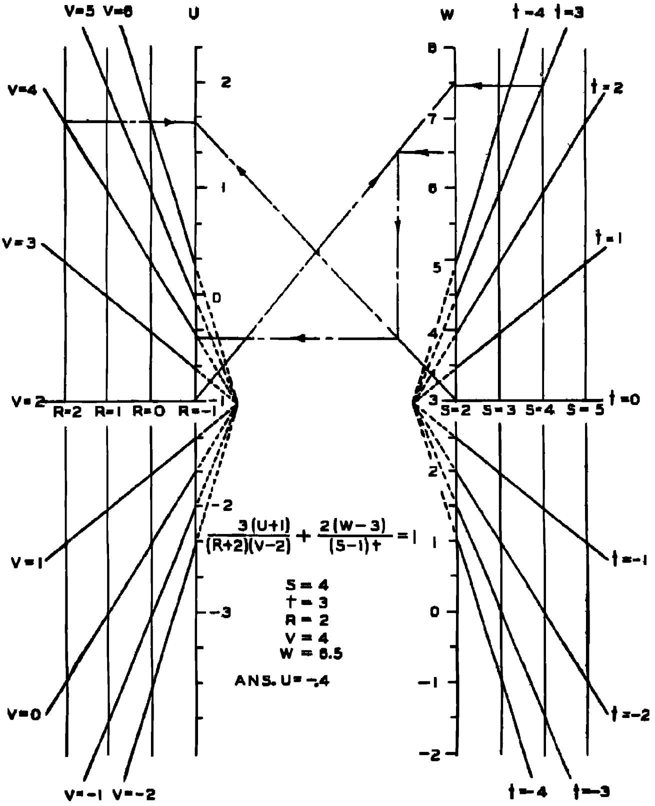


Fig. 29-141.

The R and S scales would be continued in both directions on a finished chart (Fig. 29-141).

$$\frac{3(U+1)}{(R+2)(V-2)} + \frac{2(W-3)}{(S-1)T} = 1$$

The structure of this chart is the same as the equation for Fig. 29-140 above, with the U lines replaced by V lines and the W lines by T lines.

$$f_1(u)/[f_7(Q)][f_5(R)][f_2(v)] + f_3(w)/[f_6(S)][f_4(t)] \tag{1d}$$

As an example of this type consider the equation

$$\frac{10U}{RQV} + \frac{5W}{ST} \quad 1$$

This equation may be written as

$$\frac{(10/R)U}{QV} + \frac{(5/S)W}{T} \quad 1$$

Here again consider $10/R$ and $5/S$ to be the multiplying factors of U and W, respectively. The completed chart will consist of a network of straight lines and hyperbolas on the left and a family of hyperbolas on the right (Fig. 29-142). An example is shown on the chart.

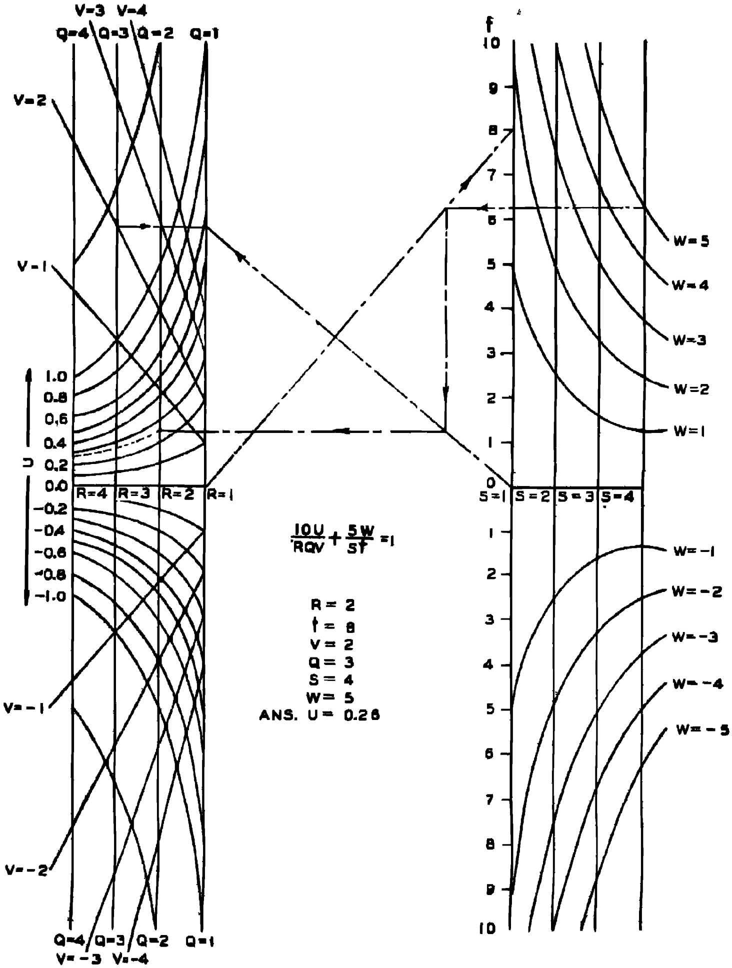


Fig. 29-142.

It is to be noticed that, if the equation is rewritten as

$$\frac{(10/R)U}{QV} + \frac{5W}{ST} = 0$$

S is a scale multiplier of T, and a family of straight lines for T replaces the W curves.

$$\frac{f_1(u)}{f_2(v)} + f_3(w) = 1 \qquad \text{or} \qquad f_1(u) + f_2(v)[f_3(w) - 1] = 0 \tag{1e}$$

This equation is a result of Eq. (1) when $f_4(t) = 1$. The t scale is replaced by a fixed point, and one of the cross lines is fixed (Fig. 29-143).

$$f_1(u) + f_3(w) = 1 \tag{1f}$$

This equation follows from Eq. (1) when $f_2(v) = 1$ and $f_4(t) = 1$. The t and v scales are replaced by two fixed cross lines (Fig. 29-144).

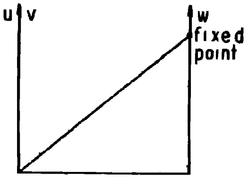


FIG. 29-143. $\frac{f_1(u)}{f_2(v)} + f_3(w) = 1.$

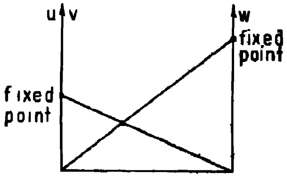


FIG. 29-144. $f_1(u) + f_3(w) = 1.$

Notice that the equation $f_5(R) \cdot f_1(u) + f_6(S) \cdot f_3(w) = 1$ could be charted by adding a network on either side.

This type of chart can be extended to an unlimited number of variables or can be used in conjunction with nomograms. For example, consider the equation

$$\frac{1}{v} + \frac{1}{t} = \frac{z}{w} \qquad \text{or} \qquad wt + vw = vt$$

Let $z/w = 1/Q$; then $(1/v) + (1/t) = 1/Q$, and $Q/w = 1/z$. The form of the chart will be as indicated in Fig. 29-145.

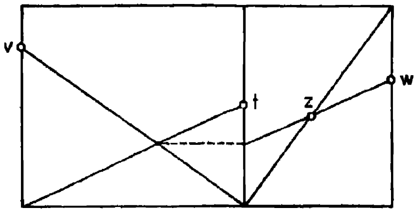


FIG. 29-145. $\frac{1}{v} + \frac{1}{t} = \frac{z}{w}.$

SIMPSON'S RULE

This is a method of finding the approximate area under a curve or finding the approximate value of a definite integral.

Given a curve, measure the ordinates spaced at intervals h apart (Fig. 29-146). The smaller the interval, the better the approximation. For an even number of intervals the area A is

$$A = (h/3)[y_0 + 4(y_1 + y_3 + \cdots + y_{n-1}) + 2(y_2 + y_4 + \qquad + y_{n-2}) + y_n]$$

where h is the distance between ordinates.

EXAMPLE 1: Find the area under the curve plotted from a table of values whose distances are measured in feet.

x	0	2	4	6	8	10	12	14	16	18	20
y	12.5	12.8	12.9	13.0	13.0	12.8	12.4	11.8	10.4	6.8	0.5

$$A = \tfrac{2}{3}[12.5 + 4(12.8 + 13.0 + 12.8 + 11.8 + 6.8) + 2(12.9 + 13.0 + 12.4 + 10.4) + 0.5] = 226.1 \text{ sq ft}$$

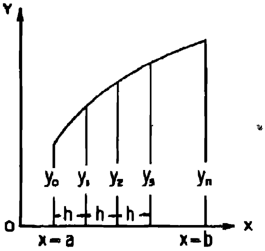


FIG. 29-146. Simpson's rule.

EXAMPLE 2: Find the value of $\int_2^{10} dx/x$. This is equivalent to finding the area under the curve $y = 1/x$ from $x = 2$ to $x = 10$. Divide the interval into eight parts. Then $h = 1$, and we obtain the following table of values.

x	2	3	4	5	6	7	8	9	10
y	$\frac{1}{2}$	$\frac{1}{3}$	$\frac{1}{4}$	$\frac{1}{5}$	$\frac{1}{6}$	$\frac{1}{7}$	$\frac{1}{8}$	$\frac{1}{9}$	$\frac{1}{10}$

$$A = \tfrac{1}{3}[\tfrac{1}{2} + 4(\tfrac{1}{3} + \tfrac{1}{5} + \tfrac{1}{7} + \tfrac{1}{9}) + 2(\tfrac{1}{4} + \tfrac{1}{6} + \tfrac{1}{8}) + \tfrac{1}{10}] = 1.6109$$

HATCHET PLANIMETER

The hatchet planimeter consists essentially of a tracing point and a convex chisel edge rigidly connected, the point and the edge being in the same plane. When the point is moved along a curve, the edge describes a curve of pursuit (Fig. 29-147). This instrument is used to approximate areas. Let the tracer point O start from a point O within the area (Fig. 29-148) (preferably near its center), move along any line to the boundary, then along the boundary and back along the same line to O. The angle θ (between the direction of departure and the direction of return) is measured and converted into radians. The approximate area is

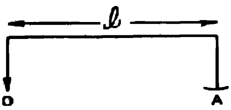


FIG. 29-147. Hatchet planimeter.

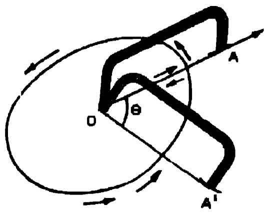


FIG. 29-148. Hatchet planimeter.

$$A = l^2\theta$$

The legs of the instrument must, at all times, be perpendicular to the horizontal plane of the paper on which the curve is drawn.

GRAPHICAL INTEGRATION

Evaluation of Integrals by Areas. It is desired to find the value of the definite integral $\int_a^b f(x)\,dx$ or the area under the curve $y = f(x)$, by a graphical method. Draw the curve $y = f(x)$ (Fig. 29-149), and along the ordinate at any point $P(x, y)$ lay off the distance y', which is the numerical value of the area bounded by the curve $y = f(x)$ and the ordinates $x = a$ and $x = x$, *i.e.*, $y' = \int_a^x f(x)\,dx$. Now for every point $P(x, y)$ there is a corresponding point $Q(x, y')$. The curve traced by the point Q is called the *integral* curve, and the curve traced by the point P is called the *derivative* curve.

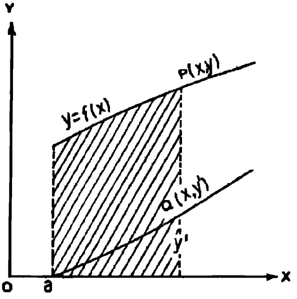


Fig. 29-149. Graphical integration.

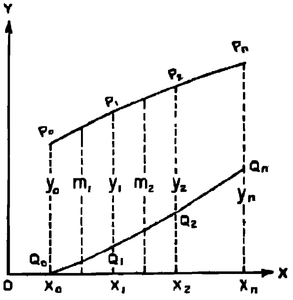


Fig. 29-150. Graphical integration.

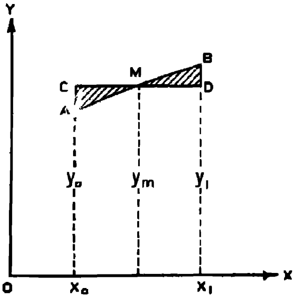


Fig. 29-151. Graphical integration.

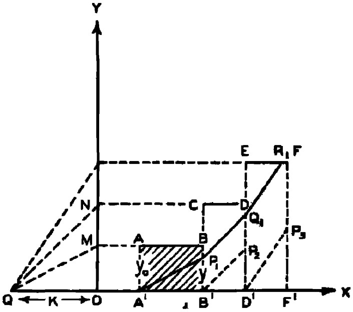


Fig. 29-152. Graphical integration.

To construct the integral curve:

1. Divide the interval from x_0 to x_n (Fig. 29-150) into n equal or unequal intervals and draw ordinates $y_0, y_1, \ldots, y_n$.

2. Measure the areas $x_0P_0P_1x_1 = x_1Q_1 = y_1'$, $x_0P_0P_2x_2 = x_2Q_2 = y_2'$, $\ldots$, $x_0P_0P_nx_n = x_nQ_n = y_n'$. These areas may be found by means of a planimeter or by the construction of mean ordinates. For example, the area $x_0P_0P_1x_1$ is equal to the area of a rectangle whose base is x_0x_1 and whose altitude is the mean ordinate m_1 within that area. Similarly, the area $x_1P_1P_2x_2$ is equal to the area of a rectangle whose base is x_1x_2 and whose altitude is the mean ordinate m_2.

The mean ordinate can be determined quite accurately by inspection since the mean ordinate y_m for the arc segment AB (Fig. 29-151) is determined by finding a

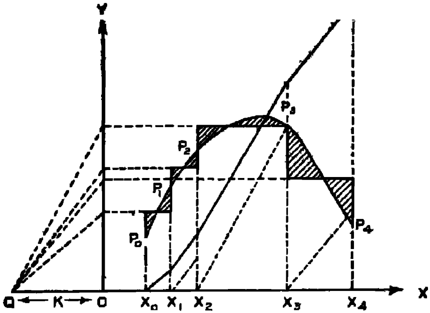
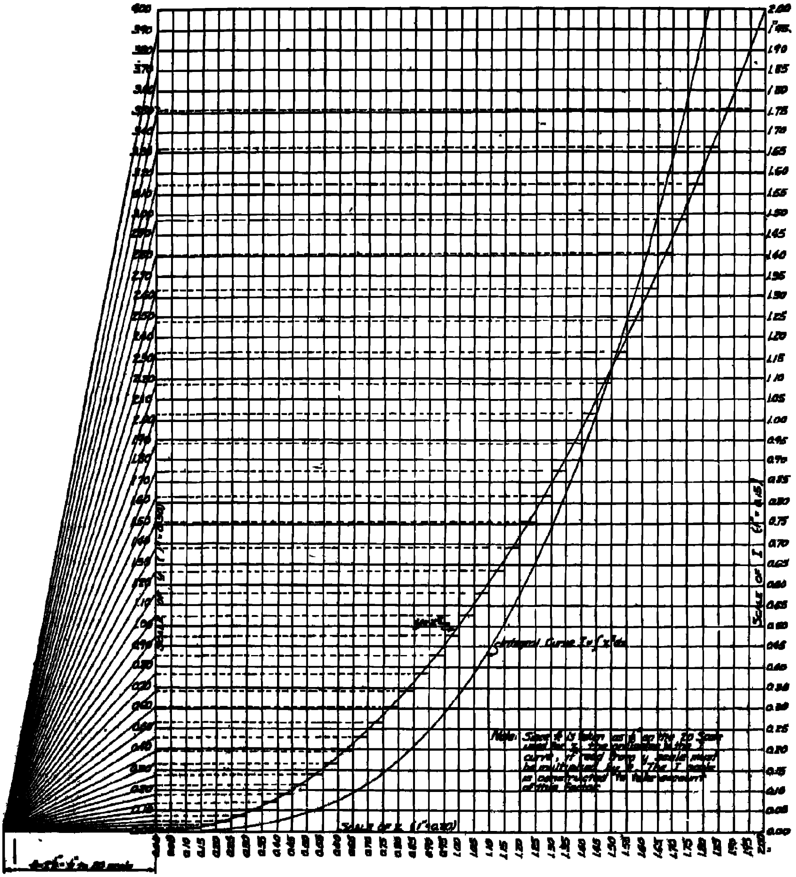


FIG. 29-153. Graphical integration.



Note: Since K is taken as ... on the 20 Scale used for y, the ordinates to the I curve, if read from y scale must be multiplied by ... The I scale is constructed to take account of this factor.

FIG 29-154. Area under curve.

point M on the curve such that the area of the rectangle x_0CDx_1 is equal to the area under the arc segment AB or such that the shaded areas ACM and MBD are equal.

First consider the case when the derivative curve consists of a series of discontinuous straight lines parallel to the x-axis (Fig. 29-152). Choose a point Q at a convenient distance K from O. Extend AB to M on the y-axis, and draw MQ. Through A', the projection of A on the x-axis, draw $A'P_1$ parallel to MQ. The line $A'P_1$ is the integral curve of the line AB. Let $B'P_1 = y'$. Then by similar triangles

$$y'/y_0 = A'B'/K \qquad \text{or} \qquad y' = y_0(A'B')/K = 1/K \cdot (\text{area under } AB)$$

Similarly, $B'P_2$ is the integral curve for the line CD, and $D'P_3$ is the integral curve for the line EF. The broken line $A'P_1Q_1R_1$ is the integral curve of the broken line $ABCDEF$.

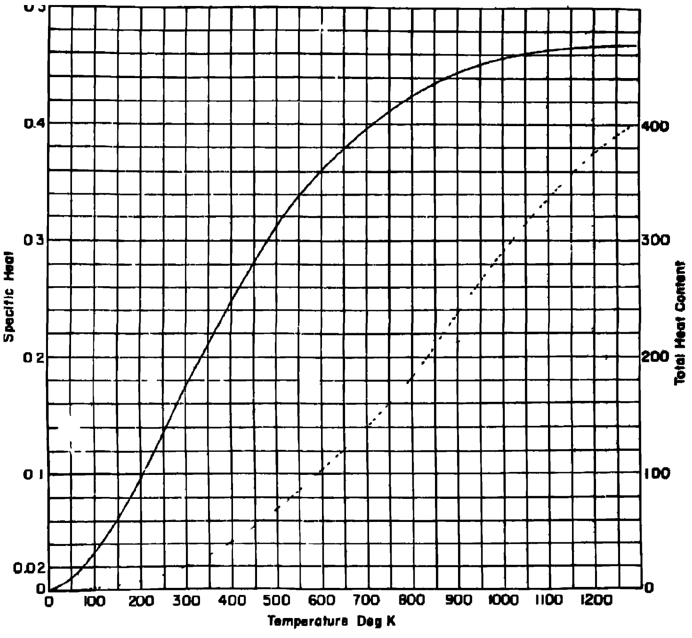


FIG. 29-155 Area under curve.

Now, consider a continuous curve (Fig. 29-153) Divide the interval from x_0 to x_1 into n parts, and draw ordinates at these points. Through $P_0, P_1, \ldots, P_n$ draw short vertical lines. Draw horizontal lines dividing each of the arcs $P_0P_1, P_1P_2, \ldots, P_{n-1}P_n$ into two parts such that the areas of consecutive shaded portions are equal (to the best approximation). Choose a suitable point Q at a distance K from O. The construction of a series of straight lines as in Fig. 29-153 represents approximately the envelope of the integral curve If the intervals x_ix_{i+1} are very small, the short sections of straight lines will approximate a continuous integral curve.

EXAMPLE 1: Find the area bounded by the curve $y = x^2$, the x-axis, and the line $x = 2$, by a graphical method. Check the result by performing the required integration.

The table of values and graphic solutions are shown in Fig. 29-154.

EXAMPLE 2. Given the graph of specific heat plotted against temperature, find the total heat content (Fig. 29-155).

Evaluation of Definite Integrals by Nomograms. If the value of the indefinite integral $\int f(x, a)\,dx$ is known, a nomogram can be made for values of the definite integral.

If the value of the indefinite integral is not known, but a table of values of the definite integral is available, then sometimes a nomogram can be made empirically.

Charts for the integrals

$$\int_{x_1}^{x_2} \tan x\,dx \qquad \int_0^x \frac{dx}{a^2 - x^2} \qquad \text{and} \qquad \int_0^{x_1} \frac{dx}{\sqrt{a + bx}}$$

are shown in Figs. 29-156*a*, 29-156*b*, 29-156*c*, respectively.

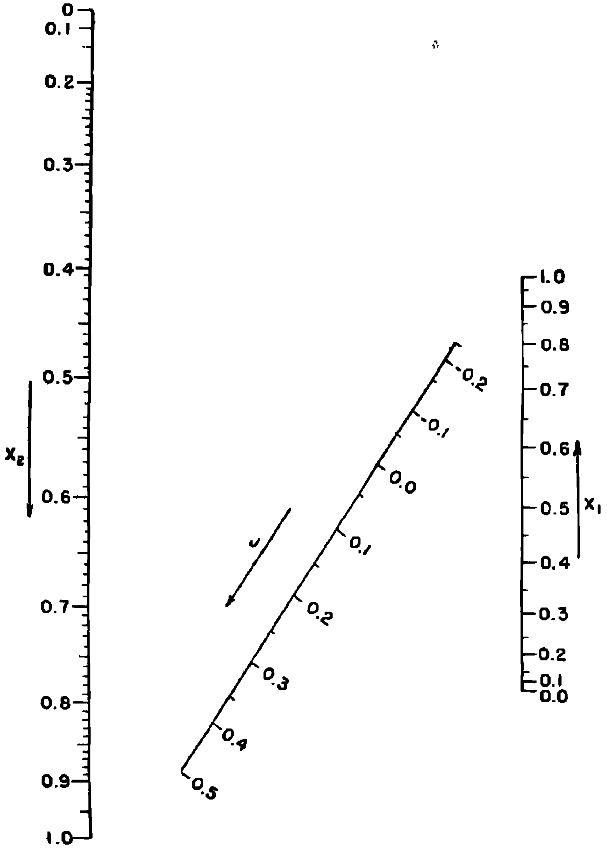
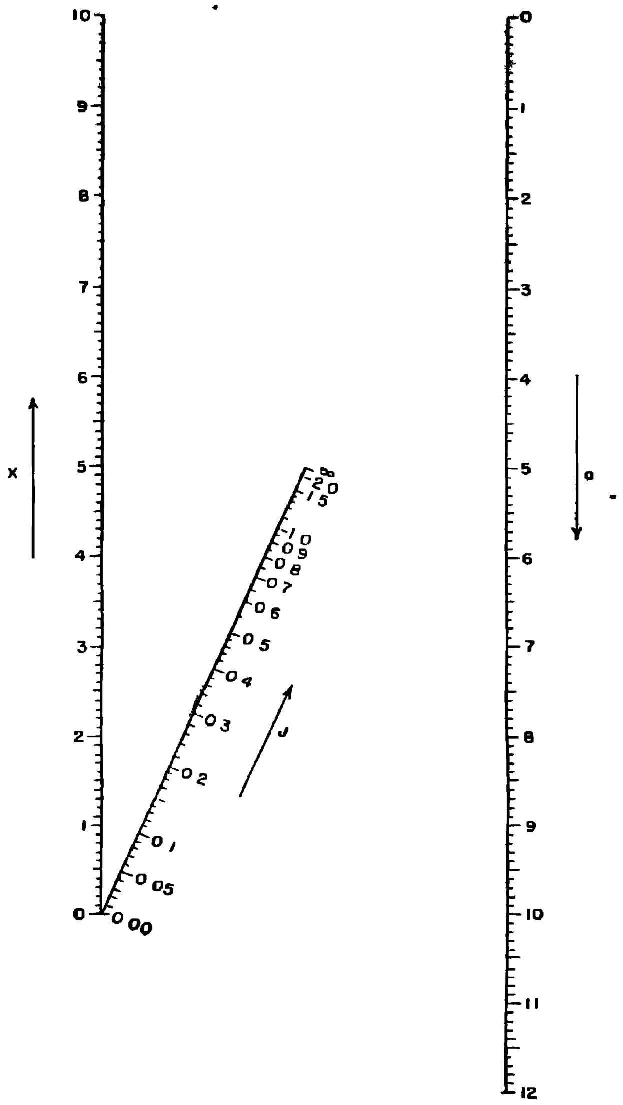


FIG. 29-156*a*. Evaluation of definite integrals.


10
9
8
7
6
5
4
3
2
1
0
X
0
1
2
3
4
5
6
7
8
9
10
11
12
a
∞
2 0
1 5
1 0
0 9
0 8
0 7
0 6
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0 4
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0 00
J

FIG. 29-156b

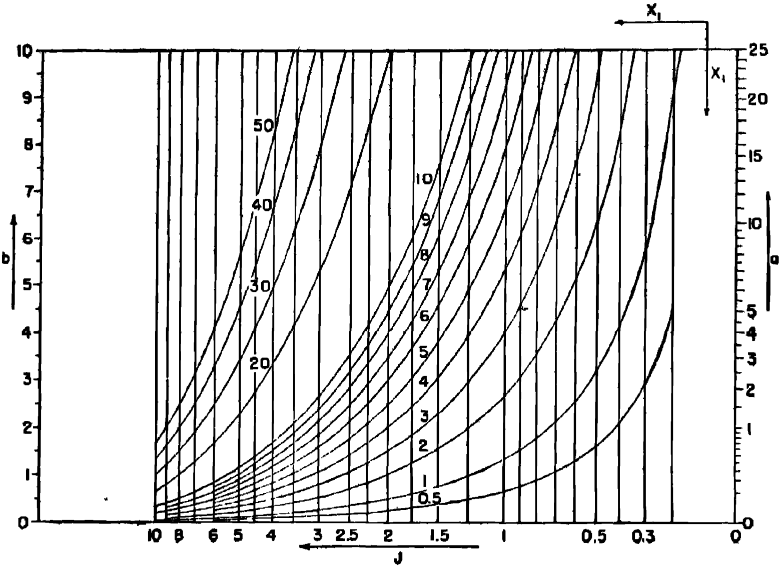
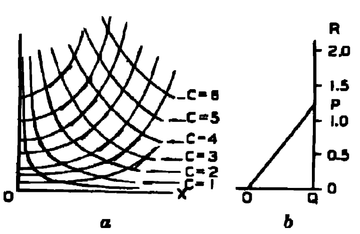


Fig. 29-156c.

GRAPHICAL SOLUTION OF DIFFERENTIAL EQUATIONS

First-order Equation $dy/dx = f(x, y)$. If (x, y) are the coordinates of a point on a curve, then dy/dx is the slope of the tangent at that point. A differential equation of the first order $[dy/dx = f(x, y)]$ assigns a definite direction to every point (x, y) in the x, y-plane. If a family of curves can be found such that, at every point of every member, the differential equation is satisfied, then that family is called the integral family of curves of the differential equation.



a b

Fig. 29-157. $dy/dx = xy$.

Example 1: Solve the equation $dy/dx = xy$.

First sketch the curves $xy = c$ (for illustrating purposes only curves in the first quadrant will be drawn) (Fig. 29-157a).

In order to show the integral curves (Fig. 29-157a), it is convenient to draw a line OQ (Fig. 29-157b) parallel to the positive direction of the x-axis and of unit length and lay off on this line a uniform scale of modulus 1. The slope of any line OP is numerically equal to PQ units. Using Fig. 29-157b, it is easy to transfer a direction at various points of each curve in Fig. 29-157a. Curves are now shown having the slopes specified by a system of short strokes. These curves are approximations to the integral curves in the first quadrant.

Example 2: A graphical solution is shown for the equation $dy/dx = x - y^2$ (Fig. 29-158). Additional aids are noted to facilitate graphing:

1. Along the curve $x = y^2$, $dy/dx = 0$
2. Along the curve $1 - 2xy + 2y^3 = 0$, $d^2y/dx^2 = 0$

Solution of $dy/dx = f(x, y)$, with the Condition $x = x_0$, $y = y_0$. Determine the slope at (x_0, y_0), and draw a short stroke from (x_0, y_0) to the line $x_0 + h$, and read off the value of y_1 for $x = x_0 + h$. Now compute the slope at $(x_0 + h_1, y_1)$, and again draw a short stroke. If this process is continued and the intervals are small, these

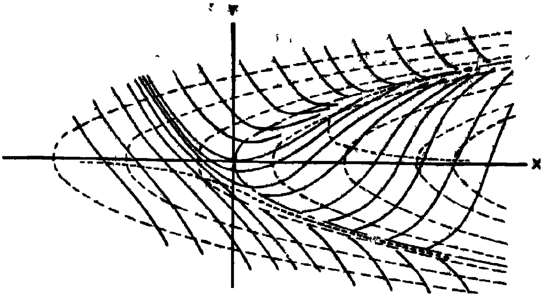


Fig. 29-158. $dy/dx = x - y^2$.

short lines will envelop a curve which is a good first approximation to the integral curve.

Graphical Solution of Two Simultaneous Equations of the First Order. Let the equations be

$$dz/dt = f_1(y, z, t) \tag{1}$$
$$dy/dt = f_2(y, z, t) \tag{2}$$

and when $t = t_0$, $y = y_0$, and $z = z_0$. Operate simultaneously with two sets of axes (Fig. 29-159).

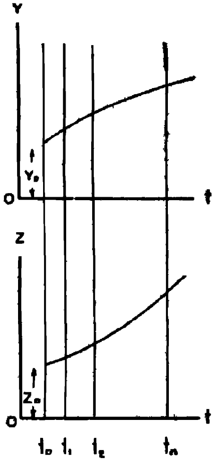


Fig. 29-159. $dz/dt = f_1(y, z, t)$, $dy/dt = f_2(y, z, t)$.

Solution of Higher Order Equations. An equation of the form

$$(d^2y/dx^2, dy/dx, y, x) = 0$$

subject to the conditions,

$$y = y_0 \qquad dy/dx = P_0 \text{ at } x = x_0$$

can be changed into the form,

$$d^2y/dx^2 = \phi(dy/dx, y, x)$$

Let $P = dy/dx$, then the original equation and its conditions become

$$dP/dx = \phi(P, y, x)$$
and
$$dy/dx = P$$

where $y = y_0$, and $P = P_0$ at $x = x_0$
Solve this pair of simultaneous equations as before.

EMPIRICAL EQUATIONS

When experimental data are obtained it is often necessary to formulate a relation between the variables in the form of an equation. From the observed data a graph is made on ordinary rectangular coordinate paper. Often the form of the curve will suggest the type of equation. However, we should choose an equation of as simple a form as possible. It is desirable to test the correctness of the form of the equation by writing the assumed equation in the form,

$$f(y) = a + b\phi(x) \qquad \text{or} \qquad Y = a + bX$$

where $Y = f(y)$ and $X = \phi(x)$ and plotting the points with (X, Y) as coordinates. If the points of this plot appear to lie on or near a straight line, then this line can be represented by the second form and hence the original curve by the first form.

Parabolic and Hyperbolic Curves $y = ax^b$ (Parabolic When $b > 0$ and Hyperbolic When $b < 0$). Parabolic curves pass through the points (0, 0) and (1, a), and as x increases, y increases. Hyperbolic curves pass through the point (1, a) and have the coordinate axes as asymptotes; and as one of the variables increases, the other decreases.

The method of verifying whether the data can be approximated by an equation of the form $y = ax^b$ is to take logarithms of both members of the equation. We obtain

$$\log y = b \log x + \log a$$

The plot of ($\log x$, $\log y$) on rectangular coordinate paper or (x, y) on logarithmic paper will be a straight line.

Consequently, *if a set of data can be approximately represented by an equation of the form* $y = ax^b$, *then the plot of* ($\log x$, $\log y$) *on rectangular coordinate paper or* (x, y) *on logarithmic paper approximates a straight line.*

Exponential Curves, $y = ae^{bx}$. Curves of this type pass through the point (0, a) and have the x-axis as asymptotes.

If a set of data can be approximately represented by an equation of the form $y = ae^{bx}$, *then the plot of* (x, $\log y$) *on rectangular coordinate paper or* (x, y) *on semilogarithmic paper approximates a straight line.*

Parabolic or Hyperbolic Curves, $y = a + bx^n$ (Where n Is Known). *If a set of data can be represented by an equation of the form* $y = a + bx^n$, *then the plot of* (x^n, y) *on rectangular coordinate paper approximates a straight line.*

Hyperbolic Curves, $y = a/(a + bx)$, $x/y = a + bx$, or $1/y = (a/x) + b$. This equation represents a hyperbola with asymptotes $x = -a/b$ and $y = 1/b$. *If a set of data can be approximately represented by an equation of the form* $y = a/(a + bx)$, *then a plot of* (x, x/y) *on rectangular coordinate paper or* (x, y) *on reciprocal paper approximates a straight line.*

Parabolic or Hyperbolic Curves, $y = ax^b + c$. Choose two points (x_1, y_1) and (x_2, y_2) on the curve sketched to represent the data. Locate a third point on the curve such that $x_3 = \sqrt{x_1x_2}$, and measure the ordinate at that point, which is y_3. Find the value of c from the equation

$$c = \frac{y_1y_2 - y_3^2}{y_1 + y_2 - 2y_3}$$

If a set of data can be approximately represented by an equation of the form $y = ax^b + c$, *then a plot of* $\log x$, $\log (y - c)$ *on rectangular coordinate paper or of* (x, $y - c$) *on logarithmic paper approximates a straight line.*

Exponential Curves, $y = ae^{bx} + c$. Choose two points (x_1, y_1) and (x_2, y_2) on the curve. Locate a third point on the curve such that $x_3 = \frac{1}{2}(x_1 + x_2)$, and measure the ordinate at that point which is y_3. Determine c from the equation used in the previous case.

If a set of data can be approximately represented by an equation of the form $y = ae^{bx} + c$, *then a plot of* x, $\log (y - c)$ *on rectangular coordinate paper or of* (x, $y - c$) *on semilogarithmic paper approximates a straight line.*

Parabola, $y = a + bx + cx^2$. *If a set of data can be approximately represented by an equation of the form* $y = a + bx + cx^2$, *then the plot of* [x, $(y - y_k)/(x - x_k)$], *where* (x_k, y_k) *are the coordinates of a point on the curve, will approximate a straight line.*

Hyperbolic Curves, $y = x/(a + bx) + c$. *If a set of data can be approximately represented by an equation of the form* $y = x/(a + bx) + c$, *then a plot of* [x, $(x - x_k)/(y - y_k)$], where (x_k, y_k) *are the coordinates of a point on the curve, will approximate a straight line.*

Logarithmic or Exponential Curves, $\log y = a + bx + cx^2$, or $y = ae^{bx+cx^2}$. *If a set of data can be approximately represented by an equation of the form* $\log y = a + bx + cx^2$, *then the plot of* $\left(x, \dfrac{\log (y/y_k)}{x - x_k}\right)$ *where* (x_k, y_k) *is a point on the curve, will approximate a straight line.*

SECTION 30

VIBRATION

BY WINSTON M. DUDLEY, SC.D.

CONTENTS

- I. Free vibrations without damping; numerical examples
- II. Effect of frictional forces
  - A. Dry friction
  - B. Velocity-squared damping
  - C. Damping proportional to velocity
  - D. Correct damping of indicating instruments
- III. Pendulum systems
- IV. Rocking chair, cylinder in trough, etc.
- V. Forced vibrations without damping
- VI. Types of unbalance in rotors
- VII. Balancing machines and other vibration-measuring instruments
- VIII. Forced vibration with viscous damping
- IX. Resilient suspension of motors and machines
- X. Spring mounting of delicate instruments for protection
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VIBRATION*

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FREE VIBRATIONS WITHOUT DAMPING

In order to be capable of executing vibrations, a physical system must contain mass and a restoring force, such as a spring. That is, if the system is moved from equilibrium in any direction, forces urging it back must be developed.

Figure 30-1 shows the simple case of a mass of weight W suspended on a coil spring whose rate is k lb per in. of deflection. The mass is supposed to be constrained so that it can move only in a vertical direction without rotation.

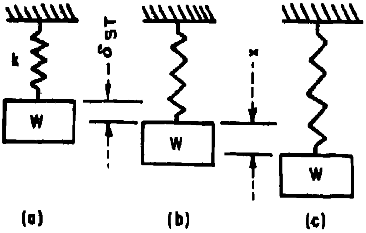


FIG. 30-1. Elementary system for free vibrations without damping.

Figure 30-1*b* shows the system at rest. The spring will be stretched an amount δ_{st} beyond its free length as indicated in Fig. 30-1*a*. This "statical deflection" $\delta_{st} = W/k$, since the weight must be supported, and equilibrium will occur when $k\delta_{st} = W$.

A vibration can be started by giving the weight initial velocity, by moving it to another position and releasing it, or both The additional deflection is denoted by x, as shown in Fig. 30-1*c*.

A mathematical solution for all possible motions can be obtained. Figure 30-2*a* shows a "free-body diagram" of the forces on the mass when it is in the deflected position, as in Fig. 30-1*c*. Since $W = k\delta_{st}$, the statical forces can be omitted, giving the simplified picture of Fig. 30-2*b*. (This is the reason for measuring x from equilibrium position.) The acceleration of the mass is d^2x/dt^2; since x is positive downward, this is also the positive direction for acceleration. Summation of vertical forces gives

$$(W/g)(d^2x/dt^2) = -kx \qquad (1)\dagger$$

the minus sign being due to the fact that the spring force acts upward for positive x. Thus, the acceleration will be negative when x is positive, indicating that the spring is trying to return the weight to equilibrium position. When the weight is above equilibrium position, x is negative and the acceleration is positive, a correct result.

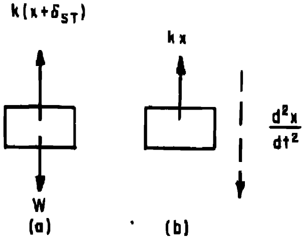


FIG. 30-2. Free-body diagram for mass in Fig. 30-1.

If initial conditions are given as $x = x_0$, $dx/dt = v_0$ when $t = 0$, the general solution of the differential equation (1) is

$$x_0 \cos \omega_n t + (v_0/\omega_n) \sin \omega_n t \qquad (2)$$

where

$$\omega_n^2 = kg/W. \qquad (3)$$

---

* Superior numbers refer to specific references listed at the end of this section.

† These numbers are for convenience in referring to the formulas.

This may be checked by substituting Eq. (2) in Eq. (1). By means of trigonometric identities, Eq. (2) can be put in the equivalent form:

$$x = A \cos (\omega_n t - \beta) \tag{4}$$

where $A = \sqrt{x_0^2 + (v_0/\omega_n)^2}$ and $\beta = \tan^{-1} (v_0/\omega_n x_0)$ (5)

The system will always require a constant time T to complete one full cycle of the vibration; T is known as the period. As the trigonometric functions in Eq. (2) repeat their values whenever $\omega_n t$ increases by 2π, we have

$$\omega_n T = 2\pi \qquad \text{or} \qquad T = \frac{2\pi}{\omega_n} = 2\pi \sqrt{\frac{W}{kg}} = 2\pi \sqrt{\frac{\delta_{st}}{g}} \quad \text{sec} \tag{6}$$

The "natural frequency" f_n is the number of cycles per second, which is the reciprocal of the period. Thus,

$$f_n = \frac{1}{T} = \frac{\omega_n}{2\pi} = \frac{1}{2\pi} \sqrt{\frac{kg}{W}} = \frac{1}{2\pi} \sqrt{\frac{g}{\delta_{st}}} \quad \text{cps} \tag{7}$$

Figure 30-3 shows a plot of Eq. (5), with the position of the weight shown vs. time. The quantity A is called the *amplitude* of the vibration, and the "peak to peak," or distance between the extreme positions, is called the *double amplitude*, or $2A$.

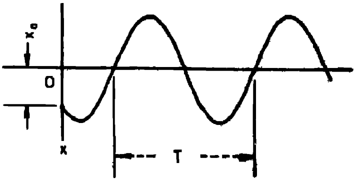


FIG. 30-3. Plot of free vibration (x vs. t).

NUMERICAL EXAMPLES: 1. A weight of 50 lb falls a distance of 2 ft from rest and then makes contact with a coil spring whose $k = 100$ lb per in. Find the greatest deflection of the spring.

SOLUTION: $\delta_{st} = W/k = 50/100 = 0.5$ in. Since the vibration starts with the weight above equilibrium position, $x_0 = -\delta_{st} = -0.5$ in. From elementary physics, the velocity developed by a 2-ft fall is found from $v_0^2 = 2gh = 2(32.2 \times 12)(2 \times 12) = 18{,}530$; hence $v_0 = 136$ in. per sec. From Eq. (3)

$$\omega_n = \sqrt{kg/W} = \sqrt{(100 \times 386)/50} = 28.2 \text{ radians/sec}$$

From Eq (5), $A = 4.85$ in. Thus the weight will reach a position 4.85 in. below *equilibrium* position, and the greatest compression of the spring will be $4.85 + 0.5 = 5.35$ in

2. An elevator car is observed to oscillate vertically with a double amplitude of ¼ in. and a frequency of 2 cps If the car weighs 6,000 lb, find the spring rate of the supports and the maximum tension developed in them.

SOLUTION: From Eq. (7), $\omega_n = 2\pi f_n = 2 \times 3.14 \times 2 = 12.57$ radians per sec. From Eq. (3),

$$k = (W/g)\omega_n^2 = (6{,}000/386)(12.57)^2 = 2{,}460 \text{ lb per in.}$$

Since the double amplitude $= 2A = 0.25$ in., $A = 0.125$ in., which is the maximum dynamic stretch in the cable. This causes a dynamic force of $kA = 2{,}460 \times 0.125 = 307$ lb. As the cable is already supporting the 6,000-lb car weight as a static load, the maximum tension is 6,307 lb.

EFFECT OF FRICTIONAL FORCES

Dry Friction. If the system of Fig. 30-1 must work against dry sliding friction, as in guides, the amplitude of a free vibration will decrease by a constant amount for each cycle, as shown in Fig. 30-4. The motion disappears entirely as soon as the weight comes to rest in a position in which the spring force is too small to overcome the friction.[1]

Velocity-squared Damping. Many actual systems are damped by fluid friction, in which the resistance is approximately proportional to the square of the velocity. Air friction, although very slight, is of this type. The mathematical treatment is difficult, resulting in infinite series representing hitherto unknown functions. The solutions, with tables complete enough for practical purposes, have been given by

Milne,[3,4,5] who had in view the problem of oscillations of water in the long pipes leading to hydroelectric plants.

With Milne's tables, any free vibration with velocity-squared damping can be plotted, one-half cycle at a time. He shows that the same mathematical results apply to all such systems, regardless of the relative values of weight, spring rate, and friction characteristic. The period of the oscillations is not constant but approaches the value for the undamped system at small amplitudes. Under such conditions, the friction force becomes insignificant, being proportional to the square of a small quantity; the vibration then continues for a long time. This occurs, for instance, in a piano string when struck; the amplitude diminishes rapidly at first, then more slowly, the string continuing to vibrate for thousands of cycles. Owing to the relatively weak friction, the pitch of the string is not changed even for large amplitudes. Another example occurs in the undulations of liquid in an open container; large amplitudes disappear rapidly, but small motions persist for a long time.

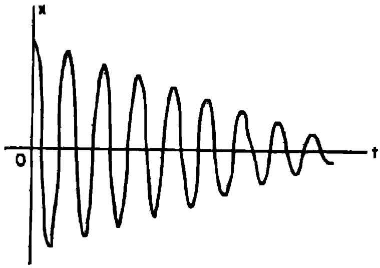


Fig. 30-4. Plot of free vibration damped by dry friction. (Amplitude decreases at constant rate with time.)

Damping Proportional to Velocity. If friction proportional to the first power of the velocity is assumed, an exact mathematical solution is easily obtained. The results have practical importance in the design of indicating instruments and vehicle suspensions. Figure 30-5 shows such a system. It is again assumed that the mass has vertical motion only, with a total spring constant k. The damping force, represented by a dashpot, equals $c\,dx/dt$. Figure 30-5*b* shows the free-body diagram, the weight again being eliminated by measuring x from the equilibrium position. The differential equation (summation of vertical forces) becomes

$$\frac{W}{g}\frac{d^2x}{dt^2} \qquad -kx - c\frac{dx}{dt} \qquad\qquad \frac{W}{g}\frac{d^2x}{dt^2} + c\frac{dx}{dt} + kx = 0 \tag{8}$$

Analysis of this equation by the usual methods shows that the system may have three different types of behavior, depending on the relative magnitude of the damping force. There is a "critical" value c_c for the damping. Letting $\omega_n{}^2 = kg/W$ as before and introducing

$$c_c = 2\sqrt{kW/g} = 2(W/g)\omega_n \tag{9}$$

there are three possible solutions, depending on whether c is greater than, equal to, or less than c_c. The equations are as follows:

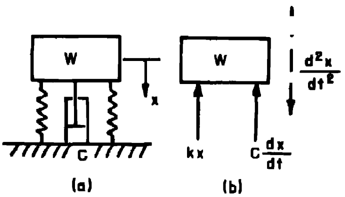


Fig. 30-5. System for free vibrations with viscous damping.

1. *More-than-critical Damping.* If $c > c_c$, there is overdamping. Let s_1 and s_2 be the two roots of the quadratic equation

$$\frac{W}{g}s^2 + cs + k = 0 \qquad\qquad s_{1,2} = \omega_n\left[\frac{c}{c_c} \pm \sqrt{\frac{c^2}{c_c{}^2} - 1}\right] \tag{10}$$

It can be seen that s_1 and s_2 will both be negative. The possible motions are given by

$$x = C_1e^{s_1t} + C_2e^{s_2t} \tag{11}$$

in which C_1 and C_2 are "constants of integration" whose values depend on the conditions assumed at $t = 0$. Thus the values of x are given by decreasing exponential functions, as shown in Fig. 30-6*a*. The system, if disturbed, does not oscillate but slowly creeps back to the central position. This is approximately the behavior of a door with a spring-and-dashpot closer attached.

2. *Critical Damping*. If $c = c_c$, there is critical damping. In this case,

$$s_1 = s_2 = -\omega_n$$

and the motion is given by

$$x = e^{-\omega_n t}(C_1 + C_2 t) \tag{12}$$

A plot of such a motion is shown in Fig. 30-6*b*. The motion is still not oscillatory, but it returns to equilibrium in less time than in Fig. 30-6*a*.

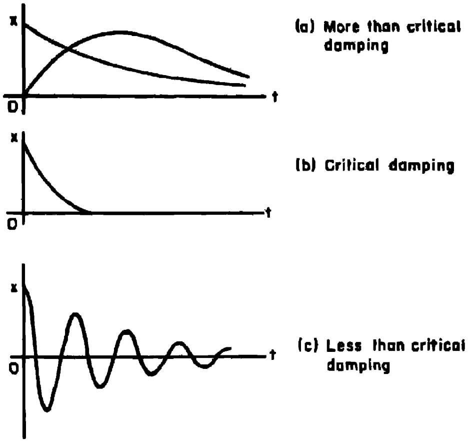


Fig. 30-6. Three types of possible motion with viscous damping.

3. *Less-than-critical Damping*. If $c < c_c$, there is less-than-critical damping. Introduce

$$q = \sqrt{\frac{kg}{W} - \frac{c^2g^2}{4W^2}} = \omega_n \sqrt{1 - \frac{c^2}{c_c^2}} \tag{13}$$

Then the possible motions are given by

$$x = e^{-(c/c_c)\omega_n t}(C_1 \cos qt + C_2 \sin qt) \tag{14}$$

As shown in Fig. 30-6*c*, a vibration of decreasing amplitude will occur. Most physical systems behave in this way; cases 1 and 2 do not usually arise unless damping is artificially increased by some means.

Correct Damping of Indicating Instruments. Proper application of the above principles is essential in the design of automatic controls or indicating instruments, such as grocery scales or electric voltmeters and ammeters. The indicating element in such cases has mass and is urged toward an equilibrium position by springs or pendulum action. When a load is applied, the location of the equilibrium position is shifted. It is seen that the most satisfactory results will occur if the system is critically damped (or slightly underdamped) as in case 2. The indicator will then go to its new position in the least possible time, without oscillations. If the system is too much underdamped, the needle will swing for some time before coming to rest, while with overdamping, it will only creep slowly toward its final reading.

Grocery scales are usually provided with an adjustable oil dashpot, so that the weight indication will be approached smoothly in the minimum time. Electric meters may have damping in the form of air paddles or a nonmagnetic metal disk moving between the poles of a magnet. In the latter case, motion of the conducting disk through the magnetic field generates eddy currents, which convert the vibration energy into heat. When the system is stationary, no force is exerted, and the indication is not affected. This is the only practical case in which damping proportional to the first power of the velocity is regularly found (except in a-c electric circuits). The same magnetic principle is used in the common automobile speedometer to give an indication of speed.

PENDULUM SYSTEMS

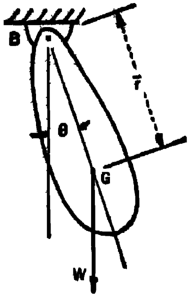


FIG. 30-7. Compound pendulum.

Figure 30-7 shows a "compound pendulum," *i.e.*, one which has distributed mass. Let W be the weight, G the center of gravity, $\bar{r}$ the radius from the swing point to G, and I_B the moment of inertia of the body around the bearing B. Taking moments around B, the equation of motion is

$$I_B(d^2\theta/dt^2) = -W\bar{r}\sin\theta \qquad (15)$$

For small angles, $\sin\theta$ may be approximated by θ (in radians). The equation is then similar to Eq. (1); $\omega_n{}^2$ becomes $W\bar{r}/I_B$, and the natural frequency is

$$f_n = (1/2\pi)\sqrt{W\bar{r}/I_B} \quad \text{cps} \qquad (16)$$

If the pendulum is reduced to a point mass W at the end of a string of length $\bar{r}$, $I_B = W\bar{r}^2/g$ and $f_n = (1/2\pi)\sqrt{g/\bar{r}}$, a familiar result in physics.

ROCKING CHAIR, CYLINDER IN TROUGH, ETC.

Figure 30-8 represents a number of possible vibratory systems, such as a rocking chair, a cylinder rolling in a trough of larger radius, etc. The moving element has a surface of radius r, and its center of gravity is at a distance h above the bottom. Its moment of inertia about a transverse axis through G is I_G. The trough radius is R. It can be shown that, if the moving element rolls without slipping, the natural frequency is

$$f_n = \frac{1}{2\pi}\sqrt{\frac{W(Rr - Rh + rh)}{(R - r)\left(I_G + \frac{W}{g}h^2\right)}} \quad \text{cps} \qquad (17)$$

If the motion takes place on a level floor, $R = \infty$ and the equation becomes

$$f_n = \frac{1}{2\pi}\sqrt{\frac{W(r - h)}{I_G + \frac{W}{g}h^2}} \quad \text{cps} \qquad (18)$$

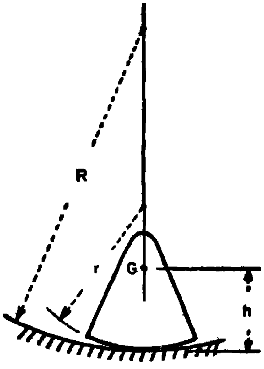


FIG. 30-8. Rocking chair, cylinder in trough, etc.

FORCED VIBRATIONS WITHOUT DAMPING

Figures 30-9, 30-10, and 30-11 show three cases in which a simple system is subject to alternating forces which compel it to vibrate at a frequency different from its natural frequency. In Fig. 30-9, the mass is acted on by a simple harmonic force $P_0 \sin\omega t$, such as might be caused by magnetic action with alternating current. In Fig. 30-10, the foundation is vibrating with an amplitude $a_0 \sin\omega t$. In Fig. 30-11, the weight carries a rotating member on which there is a small unbalanced weight W_1 at a radius r. If the rotor turns at N rpm, its rate of turning in radians per second is $\omega = 2\pi N/60$, and the centrifugal force on W_1 is $(W_1/g)r\omega^2$. The vertical component of this force is then $(W_1/g)r\omega^2 \sin\omega t$.

In all three cases, it is assumed that the mass W is guided so that only vertical motions can occur. Taking summation of vertical forces in each case, the differential equations of motion are

$$(W/g)(d^2x/dt^2) + kx = P_0 \sin \omega t \qquad \text{(Fig. 30-9)} \tag{19}$$
$$(W/g)(d^2x/dt^2) + kx = ka_0 \sin \omega t \qquad \text{(Fig. 30-10)} \tag{20}$$
$$(W/g)(d^2x/dt^2) + kx = (W_1/g)r\omega^2 \sin \omega t \qquad \text{(Fig. 30-11)} \tag{21}$$

It is seen that the equations are identical except for the meaning attached to the constants on the right-hand side.

All the systems are still capable of executing free vibrations as in Eqs. (4) and (7). However, in actual systems, such free vibrations are soon damped out by friction. Thus, the forced vibrations are the only ones of practical importance, unless transient conditions must be studied, as in the case of a spring-supported vehicle passing over a bump.

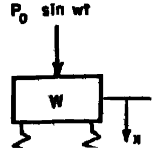


FIG. 30-9. Forced vibration without damping (constant amplitude of disturbing force).

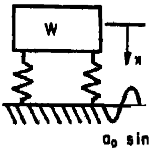


FIG. 30-10. Forced vibration caused by vibrating foundation.

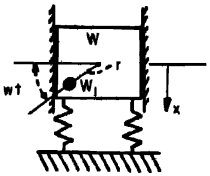


FIG. 30-11a. Forced vibration caused by rotating or reciprocating unbalanced mass (amplitude of disturbing force proportional to square of speed).

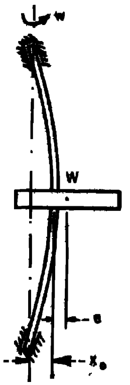


FIG. 30-11b. Deflection of flexible shaft caused by rotating unbalance.

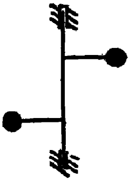


FIG. 30-11c. Dynamic unbalance.

Introduce $\omega_n^2 = kg/W$ as before. It is easily verified by substitution that the forced vibrations in the preceding cases are

$$x = \frac{P_0}{k}\frac{1}{1 - (\omega^2/\omega_n^2)} \sin \omega t \qquad \text{(Fig. 30-9)} \tag{22}$$

$$x = a_0 \frac{1}{1 - (\omega^2/\omega_n^2)} \sin \omega t \qquad \text{(Fig. 30-10)} \tag{23}$$

$$x = \frac{W_1 r}{W}\frac{(\omega^2/\omega_n^2)}{1 - (\omega^2/\omega_n^2)} \sin \omega t \qquad \text{(Fig. 30-11)} \tag{24}$$

In Fig. 30-10, let y be the amount of compression in the spring. Then

$$y = x - a_0 \sin \omega t$$

Substituting Eq. (23) for x,

$$y = a_0 \frac{(\omega^2/\omega_n^2)}{1 - (\omega^2/\omega_n^2)} \sin \omega t \tag{25}$$

Figure 30-11b shows a case closely related to Fig. 30-11a. A shaft is supported in bearings and turns with an angular velocity of ω radians per sec. It carries a disk of weight W whose center of gravity is displaced a distance e from the shaft center. The spring constant of the shaft in the lateral direction (at the location of the disk) is k lb/in. The problem is to find the amount x_0 by which unbalanced centrifugal force will deflect the shaft. As the center of gravity of the disk is moving in a circle of radius $x_0 + e$, the summation of forces gives

$$(W/g)(r_0 + e)\omega^2 = kx_0 \tag{21a}$$

Using $\omega_n^2 = kg/W$ and solving for x_0,

$$x_0 = e\frac{(\omega^2/\omega_n^2)}{1 - (\omega^2/\omega_n^2)} \tag{24a}$$

Thus, the magnitudes of important quantities in these vibrations depend on the expressions $\frac{1}{1 - (\omega^2/\omega_n^2)}$ and $\frac{(\omega^2/\omega_n^2)}{1 - (\omega^2/\omega_n^2)}$. These are known as "magnification factors" and are plotted against ω/ω_n in Figs. 30-12 and 30-13.

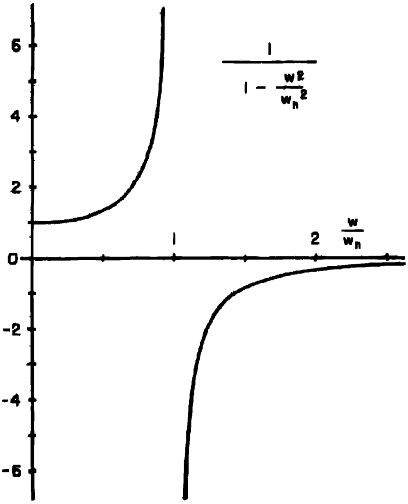


FIG. 30-12. Magnification factor for forced vibration (see Fig. 30-9).

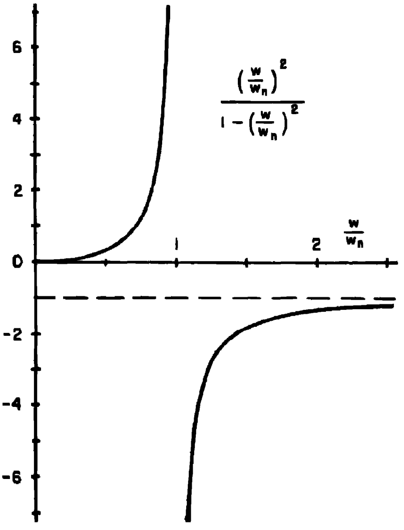


FIG. 30-13. Magnification factor for forced vibration (see Fig. 30-11a).

Figure 30-12 shows that the amount of vibration built up by the disturbing forces in Figs. 30-9 or 30-10 depends to a large extent on the ratio ω/ω_n or f/f_n of the disturbing frequency to the natural frequency. Three general regions or types of response can be noted.

1. Where the disturbing frequency is slow compared to the natural frequency, there is little inertia effect. In Fig. 30-9, a constant force P_0 would deflect the spring an amount P_0/k. At slow disturbing frequencies, the suspended mass will move slowly through cycles of this amplitude. In Fig. 30-10, at slow frequencies not much inertia force will be developed; the spring will remain of nearly constant length, and the whole system will move up and down with an amplitude a_0.

2. As the disturbing frequency approaches the natural frequency, very large vibrations can be built up; this is known as "resonance." The infinite amplitudes shown in Figs. 30-12 and 30-13 arise because friction was neglected in the analysis and because it was assumed that unlimited deflections may occur in the springs. In practice, neither of these assumptions is correct. Friction or "bottoming" of the springs may hold the amplitude to a finite value; the springs may not retain their linear force-deflection characteristic at high deflections; or breakage may occur.

3. When the disturbing frequency is well above the natural frequency, vibration is held to a small value by the inertia of the mass. That is, the weight tends to remain stationary because the disturbing force changes direction so rapidly that the weight does not have time to build up velocity in one direction.

Vibration caused by rotating or reciprocating unbalanced weight, as in Figs. 30-11a or 30-11b, occurs many times in practice. The response curve (Fig. 30-13) shows that, at slow rotational speeds, the centrifugal force is small, and not much vibration or bending of the shaft occurs. At resonance, violent motions may take place. At still higher speeds, the center of gravity tends to remain stationary in space, forcing the springs or the shaft to accommodate themselves; *i.e.*, W_1r/W and e represent the amount by which the center of gravity is displaced from the center of rotation if W_1 is small compared to W. At high speeds the magnification factor approaches -1, showing that the shaft center moves outside to permit the center of gravity to remain nearly stationary in space.

TYPES OF UNBALANCE IN ROTORS

The condition of the rotor in Fig. 30-11*b* is known as static unbalance, as it can be detected by laying the shaft on smooth horizontal rails; the heavy side will go to the bottom. Thin disks, such as turbine wheels and the balance wheels of watches, can be balanced in this way. In longer elements, such as crankshafts and electric-motor armatures, a more troublesome condition known as dynamic unbalance can occur. This is shown in Fig. 30-11c. Such a rotor is in static balance; but when run at high speed, it exerts strong shaking forces on its bearings. As the two centrifugal forces are not collinear, a force couple exists which can be balanced only by bearing reactions rotating with the shaft.

In order to correct dynamic unbalance, weights must be added or removed in at least two planes, as a couple can be counteracted only by an opposite couple. The correction locations should preferably be at the ends of the rotor and at the greatest possible radius, so as to minimize the weights required.

Some sort of balancing machine is necessary to measure the combination of static and dynamic unbalance which normally occurs in production rotors.

BALANCING MACHINES AND OTHER VIBRATION-MEASURING INSTRUMENTS

Formerly, most apparatus for measuring vibrations was mechanical. The accuracy required, and the lack of a mass market, made the equipment cumbersome and costly. In recent years, electric and electronic devices have largely taken over the field; they can be made relatively cheaply, and the use of amplifiers makes them far more sensitive than any mechanical arrangement can possibly be.

There are many such instruments now commercially available. Both old and new types are described in the general books by Timoshenko and Den Hartog.[1,2]

FORCED VIBRATION WITH VISCOUS DAMPING

In Fig. 30-14, a viscous dashpot has been added to Fig. 30-9. The frictional force (assumed proportional to the first power of the velocity) makes a considerable change in the behavior of the system. The summation of vertical forces now gives

$$\frac{W}{g}\frac{d^2x}{dt^2} + c\frac{dx}{dt} + kx = P_0 \sin \omega t \qquad (26)$$

In the steady state (no free vibrations), the motion of the mass is given by

$$x = A \sin (\omega t - \phi)$$

where

$$A = \frac{P_0}{k}\,\frac{1}{\sqrt{[1 - (\omega^2/\omega_n{}^2)]^2 + (2c\omega/c_c\omega_n)^2}} \qquad (27)$$

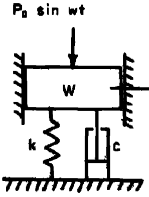


Fig. 30-14. Forced vibration with viscous damping.

ϕ is known as the phase angle, since it is that portion of 360 deg by which the x motion lags behind the disturbing force. The value of ϕ is

$$\phi = \tan^{-1} \frac{(2c\omega/c_c\omega_n)}{1 - (\omega/\omega_n)^2} \qquad (28)$$

The steps by which this solution is obtained are given in textbooks on vibration or differential equations.

Figures 30-15 and 30-16 show A and ϕ plotted against ω/ω_n. It is seen that the damping force prevents the amplitude from ever becoming infinite; also that the effect of damping on amplitude is not great except near resonance. Figure 30-16 shows the trend for the phase angle; it is small at low frequencies, passes through 90 deg at resonance, and approaches 180 deg at high frequencies. This can be demonstrated in a simple way by taking a string with a weight on one end and exciting the motion of the pendulum by moving the point of support from side to side. If the support point is oscillated at less than the natural frequency of the pendulum, the appearance will be as in Fig. 30-17, showing that the swinging of the weight is in phase with the disturbing motion of the support. Above the natural frequency, however, the string will appear crossed as in Fig. 30-18, showing that the motions at upper and lower ends are 180 deg out of phase.

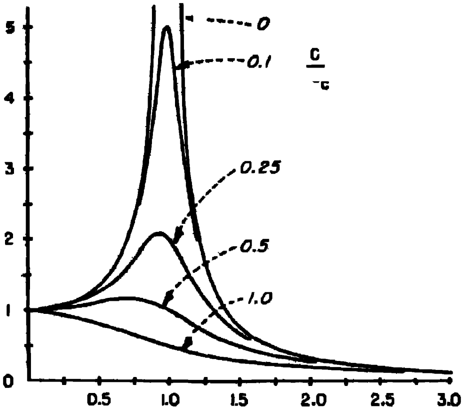


Fig. 30-15. Amplitudes of forced vibration with viscous damping.

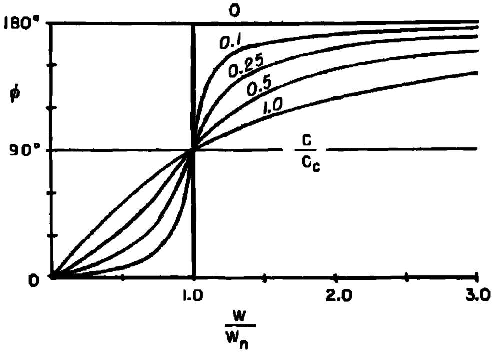


Fig. 30-16. Phase angles for forced vibration (see Fig. 30-14).

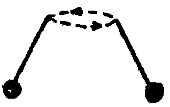


Fig. 30-17. Disturbing force and vibration in phase (below resonance).

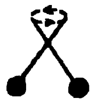


Fig. 30-18. Disturbing force and vibration 180 deg out of phase (above resonance).

The solution [Eqs. (27) and (28)] is the same for all numerical proportions of weight, damping, and spring rate; the motion is always vibratory, regardless of whether or not the damping exceeds the critical value.

RESILIENT SUSPENSION OF MOTORS AND MACHINES

Almost all power-driven devices generate vibration and noise, frequently to a disturbing extent. Reciprocating or unbalanced rotating parts create shaking forces. If impacts or explosions are involved in the working of the machine, these create noise and shock, both in the air and in the frame of the apparatus. Alternating-current electricity gives rise to magnetic hum. Gear teeth, pump and turbine blades, and electric commutators make sirenlike sounds of definite pitch, while ball or roller bearings give a confused clashing and rattling sound. The passage of water or other liquids, even at moderate speeds, is far from quiet.

If the machine is placed on a solid foundation in or near buildings, the disturbances are apt to be magnified and broadcast further. The usual types of building construction are very low in damping capacity; this is especially true of steel and reinforced concrete.

In many cases, the transmission of vibrations can be effectively prevented by interposing springs of suitable material and stiffness between the machine and its base. However, the installation must be properly designed, or matters will be worse instead of better.

If damping in the spring supports is neglected, all the necessary information can be obtained from Fig. 30-9 and Eq. (22). The maximum force applied to the machine is P_0. Let F_0 be the maximum force applied to the foundation. As the spring is the only element touching the foundation, F_0 will be equal to k times $x_{\max}$. From Eq. (22),

$$F_0 = kx_{\max} = P_0 \frac{1}{1 - (\omega^2/\omega_n^2)} \tag{29}$$

The ratio of F_0 to P_0 is known as the "transmissibility ratio" (TR), as it measures the effectiveness of the mounting in filtering out disturbances. From Eq. (29),

$$\text{TR} = \frac{F_0}{P_0} = \frac{1}{1 - (\omega^2/\omega_n^2)} \qquad \text{(Fig. 30-12)} \tag{30}$$

Figure 30-12 shows at once that spring mounting is beneficial only if the system is operating considerably above resonance, as it is only in this region that the transmissibility ratio is less than 1; anywhere below resonance, the spring mounting makes matters worse, as the force on the foundation is greater than that on the machine. Physically, the desired principle is to mount the machine in a soft or "floating" manner and thus allow the disturbing force to dissipate itself against the inertia of the machine instead of being transmitted into the supporting structure.

A question arises as to how low the transmissibility ratio should be made in order to secure good isolation. Hull[6,7,8] recommends that, for unbalance vibrations, ω/ω_n should be at least 3, while to secure satisfactory reduction of sound and noise frequencies, ω/ω_n must be 8 or more.

The spring rate necessary to produce a given transmissibility ratio is easily computed in a numerical case. It should be noted from Fig. 30-12 that, in the useful region (above resonance), the transmissibility ratio is a negative quantity, since the disturbing force and the motion are 180 deg out of phase. However, this negative sign is of no physical importance, and in the formulas below, transmissibility will be taken as a *positive* quantity. If the disturbance has a frequency of N cycles per min, $\omega = 2\pi N/60$. Substituting $\omega_n^2 = kg/W$ in Eq. (30) and solving for k gives

$$k = \frac{W}{g}\,\frac{4\pi^2N^2}{3{,}600[1 + (1/\text{TR})]} \tag{31}$$

the sign of the TR being changed. Combining the values of the numerical constants with $g = 386$ in. per sec per sec gives

$$k = 2.84 \times 10^{-5}\,\frac{WN^2}{1 + (1/\text{TR})} \tag{32}$$

EXAMPLE: The machinery of a refrigerator weighs 50 lb and contains a reciprocating compressor which operates at 400 rpm. Find the spring rate of the foundation for 10 per cent transmissibility ratio.

SOLUTION: $W = 50$ lb, $N = 400$ rpm, TR $= 0.10$. Substituting gives $k = 20.6$ lb per in. If the machine is mounted on four springs, each spring should have a rate of 5.2 lb per in. The center of gravity of the springs should be under the center of gravity of the suspended parts. Such an arrangement can vibrate in other ways besides simple vertical motion; these other modes are also analyzed by Hull.

EFFECT OF DAMPING IN SPRING SUSPENSIONS

If the foundation contains viscous damping as well as elasticity, as in Fig. 30-14, the expression for the transmissibility ratio under such conditions is[2]

$$\mathrm{TR} = \frac{\sqrt{1 + (2c\omega/c_c\omega_n)^2}}{\sqrt{[1 - (\omega^2/\omega_n^2)]^2 + (2c\omega/c_c\omega_n)^2}} \tag{33}$$

This function is plotted in Fig. 30-19. The results show that, in the useful range (well above resonance), damping is undesirable as it increases the force on the foundation. The damping does have the property of limiting the amplitude at resonance, an important point since the machine must pass through resonance every time it starts or stops. This is noticeable as a thumping noise in small electric refrigerators when the mechanism shuts down. In starting up, most machines pass through resonance so rapidly that large vibrations cannot build up. To provide protection against excessive motion at resonance, it is usually possible to provide bumpers which do not come into play at the regular running speed.

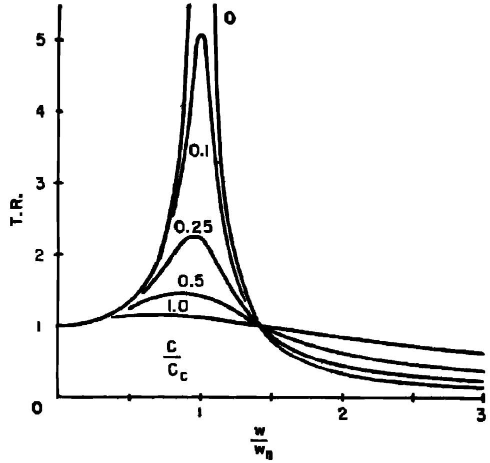


FIG. 30-19. Transmissibility ratio (spring mounting with damping).

In actual spring materials, such as rubber, the amount of damping is not usually known, nor is it of the simple viscous type. If a material is known to have considerable damping capacity, this can be counteracted by using a somewhat smaller value for the transmissibility ratio in applying Eq. (32) or (33).

SPRING MOUNTING OF DELICATE INSTRUMENTS FOR PROTECTION

It should be noted that the same "transmissibility factor" applies for transmission in either direction through the mounting. That is, if a disturbing force $P_0 \sin \omega t$ acts on the machine, the maximum dynamic force on the foundation will be $P_0 \times$ (TR). Also, if the support vibrates with a motion $a_0 \sin \omega t$ as in Fig. 30-12, the amplitude of vibration of the suspended mass will be $a_0 \times (TR)$. Both situations occur regularly in design work: in one case, a violent machine is spring-mounted to prevent radiation of shock and vibration in a structure or neighborhood; in the other, a delicate instrument is mounted on springs or rubber pads to protect it from vibration due to nearby machinery.

The lowest natural frequencies of building frames are likely to be in a range of 10 to 20 cps. An instrument mounting designed to filter out these frequencies will thus require a natural frequency of perhaps 3 to 6 cps. From Eq. (7), this means the instrument must deflect the mounting about half an inch. This may be too unstable for some applications. The building frame is also likely to carry high-frequency vibrations from shocks of various kinds. A somewhat stiffer mount may still be very beneficial against sound frequencies.

DESIGN OF VEHICLE SUSPENSIONS

Figure 30-19 can be applied directly to the design of vehicle springs and shock absorbers. Here the problem is to protect the body, load, and passengers from shocks due to waviness or bumps in the road.

The natural frequencies of modern cars are on the order of 1 cps or less. Thus the suspension will operate mostly above resonance, as very few bumps have a "wavelength" as great as the distance the car moves in 1 sec. (Railroad crossings are a notable exception.) In this region, any reduction of spring rate is beneficial, as it reduces ω_n and increases ω/ω_n, giving lower transmissibility. This is accomplished as far as possible by softer springs, softer tires, and better weight distribution.

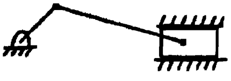
The shock absorbers are usually of the dashpot type. Figure 30-19 shows that, for best results, the damping constant must be variable. If the body is moving up and down at its natural frequency, the damping should be in the critical range so as to bring the body to rest quickly and smoothly. This action can be observed on automobiles by raising the bumper a few inches and dropping it. However, if the road is giving high-frequency bumps, as on cobblestones or "washboard" gravel, Fig. 30-19 shows that high damping makes the riding quality definitely worse. To take care of this, shock absorbers commonly have spring-loaded orifices. At body frequencies the damping is approximately critical, but at higher frequencies the increased oil pressure forces the orifices open, and the effective damping constant is reduced.

Many shock absorbers offer little resistance to closing up but employ some sort of check valve to exert a strong resistance to lengthening. So far as riding quality is concerned, this practice is of dubious value. It does not protect the body sufficiently against "striking bottom"; and on a continuously rough road, the result is merely to "rack the springs down" to a point where the average excess spring force compensates for the unbalanced action of the shock absorber. The purpose can perhaps be explained as protection for the usual flimsy shock absorber against bursting when the wheels are forced suddenly upward; but a stronger double-acting shock absorber would probably result in better spreading of the shock when the road profile requires rapid upward motion of the body.

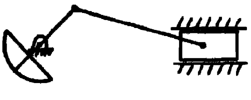
DESIGN DIFFICULTIES OF RESILIENT SUSPENSIONS FOR MACHINERY

There are a number of practical difficulties to be overcome in arriving at a successful resilient mounting. First is that of ignorance. Persons unfamiliar with vibration theory, when confronted with shaking machinery, are prone to conclude that a more massive foundation will dispose of the problem. Usually it will not, as it adds little damping capacity and does not lower the natural frequency enough to be helpful.

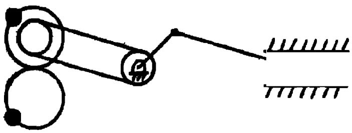
Sometimes it is possible to cure the trouble at the source by improving the balance or redesigning the troublesome mechanism. Single-cylinder engines or reciprocating pumps are frequent offenders in creating vibration. Such a mechanism cannot be inherently balanced (see Fig. 30-20). In the elementary form, the inertia force of the reciprocating mass creates a disturbance (Fig. 30-20*a*). If a counterweight of the correct amount is added to the opposite side of the crank, as in Fig. 30-20*b*, this will cancel the horizontal disturbing force (except for higher frequency components due to angularity of the connecting rod). However, there remains an unbalanced vertical force. In many machines, including steam locomotives, the practice is to make a compromise between these two extremes. A still better solution, however, is to add an auxiliary device known as a "Lanchester balancer."[9] As shown in Fig. 30-20c, this consists of two counterrotating shafts, geared together and geared in a positive manner to the crank. Each of the counterrotating shafts carries an unbalanced weight. These cancel each other in vertical motion, but they add together horizontally to oppose the inertia of the reciprocating parts. The cost of such devices is not great, they consume little power, and they frequently permit installing small steam engines in apartments, office buildings, hospitals etc., where vibration cannot be tolerated.



(a) No counterbalancing



(b) Rotating counterweight



(c) Lanchester balancer

Fig. 30-20. Methods of balancing single-cylinder mechanism.

If a machine is inherently disturbing, owing to nonuniform torque or some other condition, then spring mounting may be the only recourse. In designing such an installation, the most frequently encountered difficulties are as follows:

1. The design of suitable springs.
2. When designed for good isolation, the foundation may be too soft or "wobbly."
3. The machine may contain several disturbances at various frequencies. In designing to filter out vibrations of one frequency, care must be taken to avoid resonance with another frequency which may be present.
4. The machine on its mounting has more than 1 deg of freedom; *i.e.*, it can vibrate in different ways at different frequencies.
5. There may be conflicting rigidity requirements: soft for isolation, stiff for driving through gears or belts.
6. There may be several paths through which disturbances can escape from the machine, such as bearings, pipes, conduits, controls etc. All these should also be flexible enough to avoid fatigue failure from the vibrating motions of the spring-suspended machine.
7. Air-borne noise.

These will be discussed individually below.

1. Spring Design and Materials. Hull[8] gives a good summary of the problem of spring design and materials. A spring material should be easy to apply (as to existing machines); it must occupy little space, be low in cost, resistant to injury, have a long life, and have elastic properties which are constant and predictable. It should not "creep" or become hard with the passage of time. He lists as possible choices: (1) cork, felt, and gelatinous substances; (2) metals, especially steel; (3) rubber.

Type (1) materials tend to be expensive, uncertain in dynamic behavior, have high damping, flow plastically under load, and tend to lose elasticity with time. The latter is important, as it may be very inconvenient to remove a machine and a large foundation block in order to replace a resilient material underneath.

Metal springs can be successfully designed in many cases.[2,30] They do not deteriorate except for fatigue and corrosion, which are both preventable. However, they tend to be bulky; they are not easily applied to existing machines, and it is difficult to vary the stiffness in different directions.

Hull concludes that rubber is likely to prove the best spring material in many cases. It is cheap, takes little space, can be made in any size or shape (except that maximum thickness is limited), and is resistant to cutting or injury. It must be protected from sunlight, ozone, heat above 150°F or extreme cold, and oil.

Oxidation used to spoil rubber in 3 to 5 years. However, chemical additives have been found which greatly retard this process. Oxidation attacks only the surface layer of rubber to a depth of $\frac{1}{64}$ in. or so. Thus, rubber bands, gloves, raincoats, and bathing caps become useless in a few months or years; but thick pads will retain their elastic properties for many years.

Rubber was for a long time a proprietary or mystery material. Recently, however, much valuable information has been published on the elastic properties and other data important in spring design.[10] At present, it appears possible to design a rubber spring with as much assurance of life and elastic behavior as for metals.

At one time, application of rubber to mountings was held up by the difficulty of bonding it to metal. However, it was found that rubber would stick to brass. Steel fittings can be brass-plated and then joined to rubber with a bond strength which is many times the allowable working stress in the rubber itself. Likewise, in vulcanization with steam or electric heat, rubber could not be made more than about an inch thick, as rubber's heat conductivity is low, and the surface rubber would be damaged before the interior was properly "cooked." High-frequency electric heating has removed this restriction, as heat can now be generated uniformly throughout any thickness. As a result, application of rubber springs to streetcars and other large machines is becoming common.

2a. Conflicting Rigidity Requirements. It has been seen that, in order to secure good isolation, the natural frequency of the machine on its base must be low compared to the disturbing frequency. The low natural frequency implies soft springs. If the disturbing frequency is itself low, it may not be practical to design a sufficiently soft suspension.

Care should be taken to determine just what portions of a machine's noise or vibration are really objectionable. Everyone is familiar with the reduction of noise from a typewriter when it is placed on rubber feet or a felt pad. In such a case, the natural frequency of the typewriter in vertical motion is much *higher* than the frequency of striking the keys. Theory indicates that the mounting should then make matters worse, but experience shows the opposite. The explanation is that it is not the fundamental frequency of key striking which creates objectionable noise. When a key is struck, the frame and other parts of the machine start to vibrate at their own natural frequencies, which may be very high. These latter are easily filtered out by the padding; hence a noticeable reduction in noise is attained. The mounting theory for impact machines is discussed below.

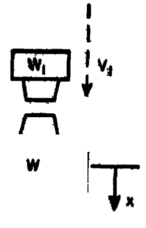


FIG. 30-21. Dynamic parts of drop-forging hammer.

2b. Elastic Mounting of Drop-forging Hammers. In the past, drop-forging plants have frequently been the cause of complaints and lawsuits because of the noise and heavy vibration transmitted to nearby buildings. In addition, in many cases, the hammer causes rapid deterioration of its own foundation, necessitating frequent and expensive repairs. It is easy to show that these unpleasant characteristics can be mitigated by two simple steps: (1) mount the machine on springs; (2) if necessary, add more mass to the anvil.

Figure 30-21 shows the three dynamic parts of the forging hammer: the striking mass W_1, the anvil W, and the elasticity k of the foundation (even a "rigid" foundation has some springiness).

Suppose the hammer, weighing W_1 lb, falls and strikes the work on the anvil with a velocity V_1 in. per sec. Usually, the hammer will not rebound, as its energy is used up in plastically deforming the metal being worked. If the hammer and anvil remain together, they will start downward with a common velocity V in./sec, which can be found from the well-known principle of the conservation of momentum:

$$W_1V_1 + W(O) = (W_1 + W)V \tag{34}$$

(It is assumed that this common velocity is reached before the anvil moves appreciably.)

The maximum amount x_0 by which the spring k will be compressed can now be computed by conservation of energy. As W and W_1 begin their motion, they have kinetic energy $E_k = \frac{1}{2}[(W + W_1)/g]V^2$. Since $V = W_1V_1/(W + W_1)$ from Eq. (34), $E_k = \frac{1}{2}[W_1^2/(W + W_1)g]V_1^2$. When all this kinetic energy of motion has been converted into potential energy E_p in the spring, the weights will be stopped. Since $E_p = \frac{1}{2}kx_0^2$, conservation of energy requires that

$$\frac{1}{2}kx_0^2 = \frac{1}{2}\frac{W_1^2}{(W + W_1)g}V_1^2 \qquad (35)$$

Solving, the maximum compression x_0 of the spring is

$$x_0 = W_1V_1\sqrt{1/[k(W + W_1)g]} \qquad (36)$$

The maximum dynamic force F_0 transmitted to the foundation is

$$F_0 = kx_0 = W_1V_1\sqrt{k/[(W + W_1)g]} \qquad (37)$$

The results of any change in the system can be easily seen from Eqs. (36) and (37). W_1V_1 represents the momentum of the hammer as it strikes. This probably should not be reduced if the hammer is to continue to do its work; such a question is beyond the present scope. W and k are the quantities which can be varied. By simultaneously increasing W and reducing k, the transmitted force F_0 can be reduced, while the action of the hammer will remain as effective as before. That is, if the product $(W + W_1)k$ is kept constant, Eq. (36) shows that the hammer-stopping distance will not change; hence the available inertia force and power to work the metal will remain the same. However, it can be seen in Eq. (37) that, if k is reduced and W increased, both these changes will reduce the transmitted disturbance F_0. As W_1 is only a small fraction of W in actual hammers, it can be concluded in any given case that, if the anvil weight W is doubled while the foundation stiffness k is cut in half, the disturbing force F_0 will be reduced to half its previous value, etc. If the stopping distance x_0 is permitted to increase, a still further reduction in F_0 can be had.

If the above principles are observed in designing and installing forging hammers, it appears quite practical to prevent the hammer from shaking up its neighborhood without reducing the effectiveness of the machine itself. Both the larger anvil and the softer springs may increase the cost of the installation, but not out of reason.

In forging slanted and unsymmetrical articles such as propeller blades, there is frequently a tendency for the hammer to move sidewise off its base. It may be necessary to provide horizontal springs and abutments to control this effect.

3. Frequency Disturbances. [See (3), page 1810.]

4. Many Types of Motion Possible. Thus far, the discussion has been confined to vibrations in which the mass moved back and forth in one direction only without rotations. While such a system is adequate to explain certain phenomena of vibration, such as resonance and isolation, there are many actual systems whose motions are more complicated. The "number of degrees of freedom" of a system is the number of independently variable displacements which must be given in order to specify the positions of the moving elements.

In general, a rigid body supported on springs has six degrees of freedom; it can move in three directions, and it can rotate about each of these axes. The number of degrees of freedom is equal to the number of natural frequencies at which the system can vibrate.[1,2]

In most cases, the calculation of the natural frequencies is a problem of considerable difficulty, as the motions are "impure," involving simultaneous translations and rotations. Hull[6] gives formulas for the six natural frequencies of a symmetrical mechanism mounted on a rectangular plate supported by springs at the corners. Crede and Walsh[11] have designed a vibrating-table installation in which all six of the natural frequencies are the same. The general methods for attacking such problems are discussed on page 1829.

5. Maintaining Alignment of Power Drives. In the case of motors, it occurs frequently that the torque is nonuniform or pulsating, although the rotational parts are in balance and do not generate lateral vibration. This is true of single-phase electric motors and also automobile engines. Various mountings have been designed which filter out the torsional vibration while maintaining lateral rigidity so that the motor remains in alignment and can drive its load.[2] Small electric motors for washing machines, refrigerators, etc., are frequently mounted in a U-shaped cradle with rubber rings around the bearings at each end. The soft rings filter out torsional impulses but are quite rigid against lateral deflection. Similar arrangements for mounting automobile engines on rubber pads are in nearly universal use.

6. Alternative Paths for Vibration Transmission. If power-driven machinery is to be effectively isolated by resilient mountings, all attachments to such machinery must be made sufficiently flexible. Bearings can be mounted in rubber. Pipes and conduits can be provided with spirals, horseshoe bends, etc., if necessary, to decrease their rigidity. Control levers can be made flexible, passed through rubber grommets, etc.

7. Noise Measurement and Reduction. Engineering investigations of noise were formerly hampered by the lack of suitable methods of impersonal measurement. However, there are now several companies marketing sound meters, and the American Standards Association has established standards for their performance and calibration.[12] Of the available instruments, the "sound-level meters" and "sound analyzers" of the General Radio Co. are probably the best known and most widely used.[13]

As a result of fundamental research by telephone engineers, sounds are now measured on the "decibel scale." This is a logarithmic scale measuring sound pressures or powers above a standard reference level. The human ear, like the other senses, has a somewhat logarithmic sensitivity, described by the well-known Weber-Fechner law of sensation. This states that, if F is the quantity of something being sensed, such as amount of weight lifted or loudness of a sound heard, and ΔF is the least detectable change in the quantity being sensed, then $\Delta F/F$ = constant. Thus if a man can detect a 5-lb change in 100 lb of weight, he should also detect $\frac{1}{20}$-lb change in a 1-lb weight. If the man in lifting any weight has a "sensation of weight" S by which he can judge whether the amount is 2, 20, 200 lb, etc., then

$$\Delta S = \Delta F/F \qquad \text{or} \qquad S - S_0 = \log (F/F_0) \tag{38}$$

If F_0 is a "threshold" or minimum quantity which can be detected, then $S_0 = 0$. Because of the logarithmic characteristic, a tremendous range of intensities can be safely experienced. All the human senses operate roughly in this way, although precise experiments will show distinct deviations from the simple logarithmic law.

A "pure tone" is a sound of definite pitch due to constant-frequency sinusoidal oscillations of pressure in the air. If E is the amplitude of pressure variation in such a tone, then the loudness of the tone on the decibel scale is given by

$$\text{db} = 20 \log_{10} (E/E_0) \tag{39}$$

where E_0 is a standard reference level of 0.0002 dynes per sq cm at 1,000 cps.

Within the audible frequency range, the amount of power P necessary to maintain a pure tone will be very nearly proportional to the square of the pressure. If Eq. (39) is written in terms of power instead of pressure,

$$\text{db} = 10 \log_{10} (P/P_0) \tag{40}$$

where P_0 corresponding to E_0 = 0.0002 dynes per sq cm is about 10^{-16} watts per sq cm. A "sound-level meter" is thus a device for measuring the power of sounds striking its microphone and giving readings in decibels.

A sound of 130 db will cause physical pain in the ear, and continued exposure to 100 db or more will eventually cause deafness. Three decibels is about the smallest change in a given sound which is detectable. At the other end of the scale, the threshold of hearing for average persons is around 20 db. A completely quiet location is almost nonexistent; there are few places where the average noise level is below 40 db. Table 30-1 gives representative readings on sounds commonly experienced:

Table 30-1. Loudness of Common Sounds, Decibels

Circumstances	*Decibels*
Airplane-motor test cell	140
Pneumatic rock drill, airplane motor at 20 ft	130
Auto horn	120
Punch press, boiler factory (riveting)	110
Platform of elevated or subway station, newsboy shouting	100
Heavy street traffic, noisy factory, airplane cabin	90
Noisy office, loud radio, noisy restaurant	80
Average factory, typewriting office, traffic on boulevard	70
Normal conversation, "noiseless" typewriter, inside railroad train	60
Inside quiet automobile	50
Quiet home	40
Broadcasting studio	30
Country road, whisper in quiet room	20

Men and machinery can create a great diversity of noise problems, but there are certain fundamental situations of frequent occurrence. Among these are

1. Two similar machines running at once give a sound level only 3 db more than one machine running alone. This is obtained from Eq. (40) since

$$10 \log_{10} (2P/P_0) - 10 \log_{10} (P/P_0) = 10 \log_{10} 2 = 3 \text{ db}$$

Thus if a machine contains two sources of noise of about equal power, it does little good to reduce only one of them.

2. Two similar machines placed in proximity will create "wave patterns" of noise. If identical pure tones are generated at two points of a room with completely sound-absorbing walls, the sound pressure at a series of neighboring locations may vary from double (+6 db) to nothing. In a more practical situation, machines do not generate pure tones, and there is a certain level of background noise. However, sound readings in the neighborhood of two similar machines will show fluctuations with position. Likewise, a single machine in a room with hard walls will create wave patterns by reflection. It is always advisable to measure sound at several positions and take an average if the wave-pattern condition exists.

3. Human psychology is frequently important. People will tolerate a great deal of noise if they feel the cause is useful and necessary. At the same time, they will complain against a very moderate noise if created by persons whom they dislike for any reason.

4. Whether a certain noise is important depends on its relation to other background noises. A sound whose individual power is more than 10 db below the general level will probably be inaudible, although it might sound very loud in a quiet place.

A valuable paper by Abbott[14] discusses 12 cases in which a great variety of machinery was successfully treated to reduce noise. Both theory and experimental methods are clearly described. The following general principles may be mentioned:

1. Frequently, noise vibrations do not originate in the component which radiates them to the air. If flexible parts or "mechanical filters" can be interposed between a motor and a sheet-metal housing, for example, the noise level can be greatly reduced.

2. If one part of a machine excites resonant vibration in another part, a slight change in the mass or stiffness of the latter is beneficial. If this situation is suspected, it can be studied by varying the speed of the driving motor by some means.

3. In either (1) or (2), if more than one path of vibration transmission exists, it will probably do little good to cut down only one channel (see 1 above).

4. If a machine is complicated and emits a variety of sounds, it may be necessary to analyze the total noise into its components at various frequencies. In this way, noise of a certain frequency can usually be traced to the part causing it. Likewise, it may be helpful to study instantaneous noise vs. rotation of the shaft of the machine, using an oscillograph.

5. The "noise spectrum" of a machine part may be altered by the structure to which it is later attached. In one case, a rear axle, when tested on a production stand, had a strong fundamental-frequency noise and a weak second harmonic. When

installed in an automobile, however, the fundamental was suppressed, and the harmonic became predominant. In acceptance testing of production axles, the second harmonic was therefore of paramount importance, but it could not even be heard on the original testing stand.

SYSTEMS WITH TWO DEGREES OF FREEDOM

It has already been seen that a system with one degree of freedom may consist of a weight suspended on a spring (Fig. 30-1) or any kind of rigid pendulum (Fig. 30-7). Thus the vibratory motion may be linear or angular; but with 1 deg of freedom there can be only one natural frequency at which vibrations occur.

Figure 30-22 shows one of the simplest cases in which there are two degrees of freedom. It is assumed that the masses are constrained to move only in a vertical direction. Since the position of both masses must be known in order to specify the configuration of the system at a given instant, there are two degrees of freedom. There will also be two natural frequencies of vibration: a slower mode in which the masses move up and down in unison (though the amplitudes of the masses may be different) and a faster mode in which the masses always move in opposite directions. This conclusion, based on experience or experiment, can also be worked out mathematically. Figures 30-23a and 30-23b show free-body diagrams of the masses, from which the following equations are obtained:

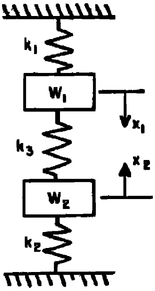


FIG. 30-22. Two vibrating systems coupled together.

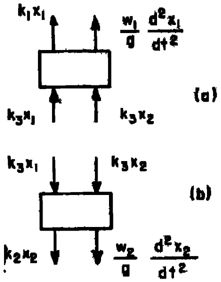


FIG. 30-23. Free-body diagrams for Fig. 30-22.

$$\frac{W_1}{g}\frac{d^2x}{dt^2} + (k_1 + k_3)x_1 + k_3x_2 = 0 \tag{41}$$

$$\frac{W_2}{g}\frac{d^2x}{dt^2} + (k_2 + k_3)x_2 + k_3x_1 = 0 \tag{42}$$

(As usual, the weights have been eliminated by measuring x_1 and x_2 from the equilibrium positions.) Introduce the symbols

$$s_1{}^2 = \frac{(k_1 + k_3)g}{W_1} \qquad s_2{}^2 = \frac{(k_2 + k_3)g}{W_2} \qquad s_3{}^2 = \frac{k_3 g}{\sqrt{W_1W_2}} \tag{43}$$

Equations like (41) and (42) are readily solved by assuming

$$\left.\begin{aligned} x_1 &= a_1 \sin \omega t \\ x_2 &= a_2 \sin \omega t \end{aligned}\right\} \tag{44}$$

and substituting. It is found that a_1, a_2, and ω can always be adjusted to satisfy the equations of motion, and there will always be two such solutions. The substitution gives

$$\left.\begin{aligned} a_1(s_1{}^2 - \omega^2) + \frac{k_3 g}{W_1}a_2 &= 0 \\ a_2(s_2{}^2 - \omega^2) + \frac{k_1 g}{W_2}a_1 &= 0 \end{aligned}\right\} \tag{45}$$

If Eqs. (45) are regarded as simultaneous equations in a_1 and a_2, there is no constant term in either equation. Hence, a_1 and a_2 can have values different from zero only if each equation gives the same value for the ratio a_1/a_2. This furnishes a third relation from which the unknown ω can be found. Thus

$$\frac{a_1}{a_2} = \frac{-k_3g}{W_1(s_1^2 - \omega^2)} = \frac{-W_2(s_2^2 - \omega^2)}{k_3g} \tag{46}$$

Using the second and third parts and clearing fractions gives the following quadratic for ω^2:

$$(s_1^2 - \omega^2)(s_2^2 - \omega^2) = \frac{k_3^2g^2}{W_1W_2} = s_3^4$$

or

$$\omega^4 - (s_1^2 + s_2^2)\omega^2 + s_1^2s_2^2 - s_3^4 = 0 \tag{47}$$

Hence

$$\omega^2 = s_1^2 + s_2^2 \pm \frac{\sqrt{(s_1^2 + s_2^2)^2 - 4s_1^2s_2^2 + 4s_3^4}}{2} \tag{48}$$

The quantity under the radical will always be positive, since it can be written as,

$$(s_1^2 - s_2^2)^2 + 4s_3^4 \tag{49}$$

which is a sum of squares. Also, the value of the radical is always less than $s_1^2 + s_2^2$, since the quantity underneath is

$$(s_1^2 + s_2^2)^2 - 4(s_1^2s_2^2 - s_3^4) \qquad \text{and} \qquad s_1^2s_2^2 - s_3^4 = \frac{(k_1k_2 + k_2k_3 + k_3k_1)g^2}{W_1W_2} \tag{50}$$

which cannot be negative. Thus, for any real system as in Fig. 30-22, Eq. (48) will always give two positive but different values for ω^2. When one of these values is selected, Eq. (46) determines the value of a_1/a_2 associated with this frequency of vibration.

EXAMPLES: 1. The simplest case occurs when $W_1 = W_2 = W$ and $k_1 = k_2 = k$. If ω_a^2 and ω_b^2 represent the two natural frequencies, it is found that

$$\omega_a^2 = kg/W$$

where $a_1/a_2 = -1$

and $\omega_b^2 = [(k + 2k_3)g]/W$

where $a_1/a_2 = 1$

The a-type vibration can be recognized as one in which the masses move up and down in unison through the same amplitude, with a frequency $f_a = \omega_a/2\pi$ cps per sec. Since the coupling spring k_3 does not change its length during such a motion, it is like a rigid connection and has no effect on the frequency. The b-type motion is one in which the masses move in opposite directions with equal amplitude at a frequency $f_b = \omega_b/2\pi$. In such a motion, the center point of k_3 remains stationary. If this is regarded as a fixed point, the stiffness of one-half of k_3 is $2k_3$, which explains the result for f_b since each mass is vibrating on springs equivalent to $k + 2k_3$.

2. As a general numerical example, let $W_1 = 30$ lb, $W_2 = 50$ lb, $k_1 = 20$, $k_2 = 30$, $k_3 = 40$ lb per in. Substituting in the equations, it is found that the lower frequency is 2.93 cps, with $a_1/a_2 = -1.137$, while the faster frequency is 7.10 cps, with $a_1/a_2 = 0.431$.

3. A case of some practical importance occurs with k_1 and k_2 zero. This can be visualized as a pair of railroad cars on a track, connected by a spring, friction being neglected. Solving the equations, it is found that one frequency is zero (corresponding to motion of the system along the track at uniform speed, a "vibration" of zero frequency), while the other is

$$f = (1/2\pi)\sqrt{[k_3g(W_1 + W_2)]/W_1W_2} \tag{51}$$

It is also found that, for the vibrating mode, $a_1/a_2 = W_2/W_1$, or $W_1a_1 = W_2a_2$, showing that the center of gravity of the system does not move during a vibration, since no external forces are acting. The position of the "node" or stationary point of the spring can be determined from this relation; the node divides the system into two parts, each of which must have the same natural frequency.

4. In Example 1, the symmetrical case, an interesting type of behavior occurs when the coupling spring k_3 is made very weak compared to the others. This makes the two natural frequencies fall very close together. Figure 30-24 shows a pair of equal pendulums connected by a weak spring, giving a system in which one natural frequency (with the pendulums moving in phase) is slightly less than the other frequency in which the motions are opposed. This occurs because in one type of motion the coupling spring is not stretched,

while in the other it is. If the system is disturbed by moving one pendulum aside and releasing it, the first pendulum will swing with a slowly decreasing amplitude until it comes to rest. During the same time, the second pendulum gradually builds up a vibration equally large. A typical record is shown in Fig. 30-25. From this point on, the procedure continues cyclically, with the motion transferring from one pendulum to the other and back again, until friction brings the system to rest. A vibration with variable amplitudes of this kind is known as "beating." It is a familiar effect in music when two instruments of slightly different pitch are sounded together. As one vibration gradually changes phase with respect to the other, the sound intensity varies from double to nothing. The "beat frequency" is the difference in the two primary frequencies: thus, when tones of 256 and 258 cps are sounded, a beat frequency of 2 per sec will occur.

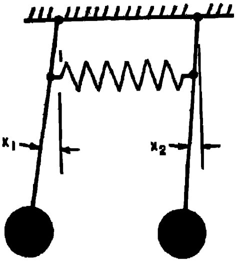


FIG. 30-24. System with two nearly equal natural frequencies (illustrates "beating").

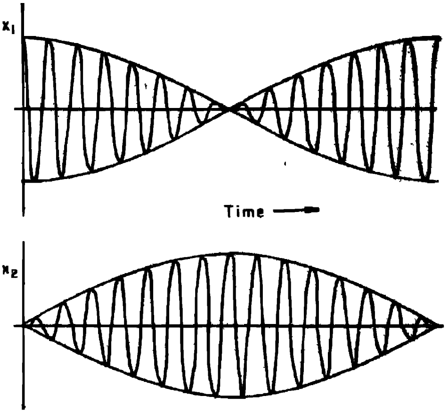


FIG. 30-25. "Beating" motions possible with Fig. 30-24.

Such motions can be demonstrated mathematically by using the complete solution of the differential equations. This is

$$\left.\begin{aligned} x_1 &= A_1 \cos \omega_1 t + B_1 \sin \omega_1 t + A_2 \cos \omega_2 t + B_2 \sin \omega_2 t \\ x_2 &= A_1 \cos \omega_1 t - B_1 \sin \omega_1 t + A_2 \cos \omega_2 t + B_2 \sin \omega_2 t \end{aligned}\right\} \tag{52}$$

Suppose that when $t = 0$, $x_1 = x_0$, $x_2 = 0$, and the velocities are zero. The constants are found to be

$$A_1 = A_2 = x_0/2 \qquad B_1 = B_2 = 0$$

giving

$$\begin{aligned} x_1 &= x_0/2(\cos \omega_1 t + \cos \omega_2 t) \\ x_2 &= x_0/2(-\cos \omega_1 t + \cos \omega_2 t) \end{aligned} \tag{53}$$

By trigonometric identities, these become

$$\left.\begin{aligned} x_1 &= x_0 \cos\left(\frac{\omega_1 + \omega_2}{2} t\right) \cos\left(\frac{\omega_1 - \omega_2}{2} t\right) \\ x_2 &= x_0 \sin\left(\frac{\omega_1 + \omega_2}{2} t\right) \sin\left(\frac{\omega_1 - \omega_2}{2} t\right) \end{aligned}\right\} \tag{54}$$

Curves like Fig. 30-25 can be obtained from these equations.

5. Figure 30-26 shows a side view of the suspended parts of an automobile. Considering only motions parallel to the plane of the paper, the system has two degrees of freedom, as it can move vertically and also rotate about a transverse axis. The natural modes and frequencies can be found by the same procedure as used above. Let x be the downward motion of the center of gravity and ϕ be the counterclockwise rotation. Then, as shown in Fig. 30-27, the compression of the front and rear springs will be $x + a\phi$ and $x - b\phi$, respectively. It is necessary to know the moment of inertia I of the body about a transverse axis through G in order to write the force equations. Using the free-body diagram in Fig. 30-28, two equations are obtained, one for summation of vertical forces, the other for moments about G. These are

$$\frac{W}{g}\frac{d^2x}{dt^2} + k_1(x + a\phi) + k_2(x - b\phi) = 0 \tag{55}$$

$$I\frac{d^2\phi}{dt^2} + k_1a(x + a\phi) - k_2b(x - b\phi) = 0$$

Assuming $x = x_0 \sin \omega t \quad \text{and} \quad \phi = \phi_0 \sin \omega t$ (56)

gives
$$\frac{-\phi_0}{x_0} = \frac{k_1 + k_2 - (W/g)\omega^2}{k_1a - k_2b} = \frac{k_1a - k_2b}{k_1a^2 + k_2b^2 - I\omega^2} \tag{57}$$

In a numerical case, the two natural frequencies are easily found from the quadratic equation for ω^2. Each of these in turn determines the ratio of x_0/ϕ_0, which gives the distance d to the apparent center of rotation for that mode, as shown in Fig. 30-29.

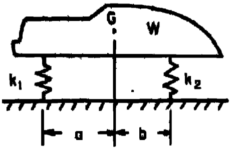


Fig. 30-26. Side view of dynamic parts of automobile suspension.

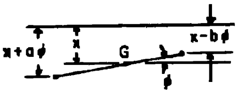


Fig. 30-27. Displacement and pitching rotation coordinates for Fig. 30-26.

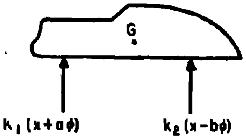


Fig. 30-28. Free-body diagram for Fig. 30-26.

It is possible to adjust the locations of the springs in such a way that in one mode of vibration, the system pivots about the front spring, while in the other it pivots about the rear spring. Let k be the radius of gyration of the body, defined by $I = Wk^2/g$. It can be shown that the above condition will occur if $k^2 = ab$. That is, if $k^2 = ab$, the vertical motion at one spring is independent of vertical motion at the other spring. This may be shown as follows: if a vibration mode is to exist with no motion at the front spring, then it is necessary that

$$x_0 + a\phi_0 = 0 \qquad \text{or} \qquad -\phi_0/x_0 = -1/a$$

Substituting this in Eq. (57), it is found that

$$\omega^2 = [k_2(a + b)g]/Wa = [k_2b(a + b)]/I \tag{58}$$

from which $k^2 = ab$. If we begin by assuming no motion at the rear spring, then

$$x_0 - b\phi_0 = 0$$

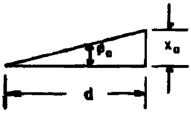


Fig. 30-29. Locating apparent centers of rotation for automobile-body motions (see Fig. 30-26).

When this is substituted, it also gives $k^2 = ab$.

The result is independent of the spring rates k_1 and k_2, although these control the frequencies in their respective modes. The condition may be described by saying that each spring is located at the center of percussion of the other spring. This is thought to be the best arrangement for riding comfort, since, when the front wheels strike a bump, only the front springs are compressed; and the same is true for the rear. If the condition is not satisfied, then a bump at the front wheels may compress both front and rear springs. If the rear springs are already compressed when the rear wheels strike the same bump, the probability of bottoming and general roughness is increased.

The above analysis was not available until 1923 and was not applied until several years thereafter. In the early days, it was thought that a long wheel base promoted riding comfort, hence the axles were placed at the extreme front and rear of the chassis. In such an arrangement, ab will be much greater than k^2, and a very rough ride results. In modern cars, the axles have been moved toward the center of the car. The front axle is now partly under the engine, while the rear axle has moved forward under the rear seat. Heavy bumpers, trunks, grills, etc., are used at both ends of the car in the attempt to get as much mass as possible beyond the axles. The result more nearly approaches the desired $k^2 = ab$ condition, but it is not possible to reach it completely without having the axles so far under the car that steering control at high speeds is impaired, with the added hazard of having the projecting portion of the car strike the ground when moving over uneven pavement such as steep driveway entrances, etc. Thus the final design is a compromise, although soft springs and tires are also used to lower the natural frequencies and promote riding comfort.

DYNAMIC VIBRATION ABSORBER

Figure 30-30 shows another important case having two degrees of freedom. W_1 and k_1 indicate a piece of machinery or structure subject to a disturbing force $P_0 \sin \omega t$. W_2 and k_2 represent a "damper" which will entirely prevent any forced vibration of W_1, provided the damper is tuned so that $k_2g/W_2 = \omega^2$. This can be shown as follows: considering each mass as a free body and summing vertical forces gives the two equations

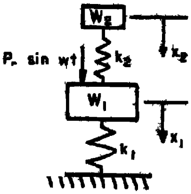


FIG. 30-30. Use of Dynamic Vibration Absorber to prevent vibration.

$$\frac{W_1}{g}\frac{d^2x_1}{dt^2} + (k_1 + k_2)x_1 - k_2x_2 = P_0 \sin \omega t \tag{59}$$

$$\frac{W_2}{g}\frac{d^2x_2}{dt^2} + k_2x_2 - k_2x_1 = 0 \tag{60}$$

The system is, of course, capable of executing free vibrations. There are two natural frequencies, which can be computed as shown previously. However, any such free vibrations in an actual system would soon be damped out by friction, so that only the forced vibrations at a frequency $\omega/2\pi$ are important. Assuming $x_1 = a_1 \sin \omega t$, and $x_2 = a_2 \sin \omega t$ and substituting gives two simultaneous equations for a_1 and a_2:

$$[k_1 + k_2 - (W_1/g)\omega^2]a_1 - k_2a_2 = P_0 \tag{61}$$

$$-k_2a_1 + [k_2 - (W_2/g)\omega^2]a_2 = 0 \tag{62}$$

For simplicity, let $k_1g/W_1 = \Omega_1^2$ and $k_2g/W_2 = \Omega_2^2$ (63)

The results are found to be

$$a_1 = \frac{P_0}{k_1}\,\frac{1 - (\omega^2/\Omega_2^2)}{\left(1 - \frac{\omega^2}{\Omega_2^2}\right)\left(1 + \frac{k_2}{k_1} - \frac{\omega^2}{\Omega_1^2}\right) - \frac{k_2}{k_1}} \tag{64}$$

$$a_2 = \frac{P_0}{k_1}\,\frac{1}{\left(1 - \frac{\omega^2}{\Omega_2^2}\right)\left(1 + \frac{k_2}{k_1} - \frac{\omega^2}{\Omega_1^2}\right) - \frac{k_2}{k_1}} \tag{65}$$

Equation (64) shows that, if $k_2g/W_2 = \Omega_2^2 = \omega^2$, then a_1 will be zero, thus establishing the condition stated at the beginning. Equation (65) shows that a_2 then equals $-P_0/k_2$, or $-k_2a_2 = P_0$. This shows the physical behavior of the tuned damper; it vibrates at exactly the right phase and amplitude so that the variable force in k_2 neutralizes the disturbing force, thus eliminating any vibration of the main mass W_1.

Such a damper is, in many cases, ideally suited for preventing vibration. Large vibrations of W_1 will not build up unless the disturbing frequency ω is close to the natural frequency $\Omega_1 = \sqrt{k_1g/W_1}$. In such a case, even a small disturbing force $P_0 \sin \omega t$ can build up a large vibration by resonance. Hence, if only a small disturbing force is involved, the vibration can be cured with a damper mass which is a very small fraction of the main weight W_1. Since the disturbing force is opposed by inertia rather than friction, no energy is converted by the damper. Thus the efficiency of the original machine is not affected, nor is there a problem of maintaining a correct frictional adjustment in the damper.

Whether such a damper is practical in a given case depends on the magnitude and frequency of the disturbing force. The damper spring k_2 must be able to stand up indefinitely when subjected to an alternating load of $\pm P_0$. Also, when $a_1 = 0$, Eqs. (63) and (65) show that $P_0 = k_2a_2 = W_2\omega^2a_2/g$. This determines the necessary size of damper weight W_2 for a specific application, as a_2 will usually be limited by the design of the spring. If P_0 is small and the disturbing frequency $\omega/2\pi$ is high, a small damper weight will suffice. As ω is reduced, however, the required damper eventually reaches cumbersome and impractical proportions.

Dampers of this type have been successfully applied in such diverse situations as electric hair clippers[16] and bending vibrations of the hull structure of a merchant ship.[17]

If the disturbing frequency $\omega/2\pi$ is variable, the damper may be worse than useless, as it creates two natural frequencies instead of one. Large vibrations will then build up if the disturbing frequency coincides with either of these natural frequencies. In one case, the masses vibrate in phase; in the other, their motions are opposed.

In the hair clipper, the disturbance was due to constant-frequency (60-cycle) electromagnetic impulses. In the motorship, the vibration was caused by impulses from the propeller blades. The engine speed varied over a certain range. However, since the ship would usually run for hours or days at one speed, it was practical to adjust the damper for the speed being used by running water to the proper level in a tank which was part of the damper mass.

By an ingenious extension of the dynamic-absorber principle, it has been applied to the reduction of torsional vibration in airplane engines.[18] The disturbing frequency varies with the running speed of the engine. The damper is made in the form of a "torsional pendulum" whose "spring" is provided by centrifugal forces. As these vary with the square of the speed, a relation like Eq. (63) can be made to hold at all speeds, thus removing the constant-frequency limitation.

Another possibility is to introduce dry or viscous friction in the damper in addition to spring action. Such a device will no longer prevent vibration entirely, but it holds it to a safe value even in passing through resonance. The theory has been completely worked out,[2,19] and some successful applications have been made.[20]

SYSTEMS OF SEVERAL DEGREES OF FREEDOM

Previously, systems with one or two degrees of freedom have been considered. It was seen that the number of natural frequencies equals the number of degrees of freedom and that for each frequency the amplitudes of vibration of the masses are in a fixed ratio with each other. All amplitudes may be increased or decreased proportionally, depending on initial conditions, but the relative amplitudes for each natural frequency are fixed by whatever mass and stiffness arrangement exists in the system.

FIG. 30-31. Vibrating system with many uniform parts (the "string-of-beads" problem).

When more than two degrees of freedom must be considered, the general results are the same as above, but the labor of computing them increases rapidly with the complexity of the system, and more advanced mathematical methods are required. A completely general case of six degrees of freedom, with nonuniform masses and with each mass elastically connected to most of the other masses, may require 12 to 15 hr of computing if all six of the natural modes and frequencies must be found.

Systems with Uniform Parts. Many problems of practical importance have especially simple structural forms so that shorter methods can be used. In some cases, formula solutions exist for the natural frequencies and proportional amplitudes. Figure 30-31 shows the classical "string-of-beads" problem, first solved by Lagrange ("Mecanique Analytique," 1787). A weightless string is stretched with a tension T. It carries uniform "beads" of mass m at equal intervals L along the string. The problem is to find the natural frequencies and proportional amplitudes for small lateral vibrations. If there are n beads, it can be shown by the so-called "calculus of finite differences"[21] that the n natural frequencies are given by

$$f_s = \frac{1}{\pi}\sqrt{\frac{T}{mL}}\sin\left[\frac{s(180°)}{2(n+1)}\right] \tag{66}$$

where $s = 1, 2, \ldots n$. If the masses are numbered from left to right, the mass number being r, the proportional amplitudes are

$$a_{rs} = C_s \sin\left[\frac{sr(180°)}{n+1}\right] \tag{67}$$

where C_s is an arbitrary constant, and s and r each range independently from 1 to n. That is, if $s = 3$, the third natural frequency is being considered; and if r is given values from 1 to n in succession, the relative amplitudes of the masses are found.

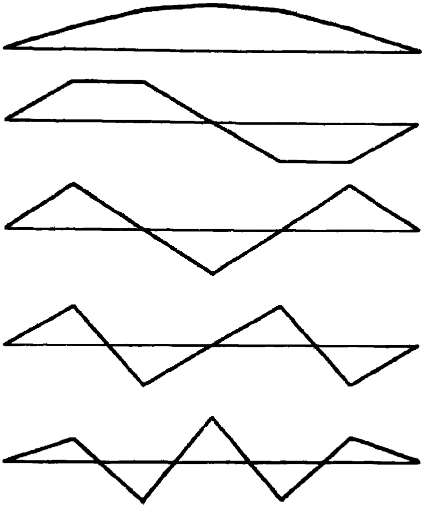


FIG. 30-32. The five possible types of free vibration in one plane for Fig. 30-31.

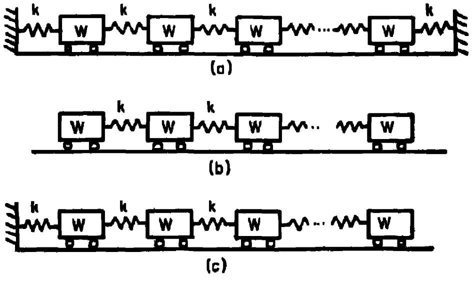


FIG. 30-33. Analysis of longitudinal vibrations in trains of cars.

EXAMPLE: Let $n = 5$, $T = 100$ lb, $L = 20$ in., and the weight of each mass $= 0.2$ lb. Then $m = W/g = 0.2/386$. The natural frequencies are found to be 8.11, 15.7, 22.2, 27.2, and 30.2 cps. The proportional amplitudes are shown in Fig. 30-32.

Figures 30-33*a*, *b*, and *c* show three similar problems. In each case, n cars of weight W rest on a frictionless track and are connected by frictionless springs whose rate is k lb per in. Horizontal vibrations in the train are to be considered, with varying end conditions. The general procedure is well explained by Karman and Biot.[21] The following results can be obtained:

1. When both ends are fixed to immovable supports by springs k, there are n natural frequencies, given by

$$f_s = \frac{1}{\pi}\sqrt{\frac{kg}{W}}\sin\left[\frac{s(180^\circ)}{2(n+1)}\right] \quad \text{where } s = 1, 2, \ldots, n \tag{68}$$

The proportional amplitudes are given by Eq. (67).

2. When both ends are free, there are only $n - 1$ vibration frequencies, as the entire train can move at uniform speed, a "vibration" of zero frequency.

$$f_s = \frac{1}{\pi}\sqrt{\frac{kg}{W}}\sin\left[\frac{s(180^\circ)}{2n}\right] \tag{69}$$

$$a_{rs} = C_s \cos\left[\frac{s(2r-1)(180^\circ)}{2n}\right] \tag{70}$$

where $s = 1, 2, \ldots, n - 1$, and $r = 1, 2, \ldots, n$.

3. When one end is free, and one end is fastened to fixed support by spring k,

$$f_s = \frac{1}{\pi}\sqrt{\frac{kg}{W}}\sin\left[\frac{(2s-1)(180^\circ)}{2(2n+1)}\right] \tag{71}$$

$$a_{rs} = C_s \sin\left[\frac{(2s-1)r(180^\circ)}{(2n+1)}\right] \tag{72}$$

where $s = 1, 2, \ldots, n$, and $r = 1, 2, \ldots, n$.

If the mass weights or spring constants, or both, are nonuniform, the method of finite differences can no longer be applied, except for the case where the nonuniformities are confined to one unit at each end of the train.

Longitudinal Motions of Nonuniform Masses and Springs in a Straight Line. The case of two unequal masses connected by a spring has already been solved [Eq. (51)].

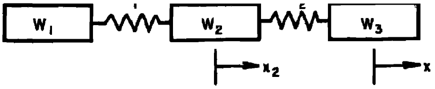


FIG. 30-34. Analysis of nonuniform trains (three masses).

Figure 30-34 shows the next step, with weights W_1, W_2, and W_3 connected by springs k_1 and k_2. Taking summation of forces of each mass in turn gives the equations

$$(W_1/g)(d^2x_1/dt^2) + k_1x_1 - k_1x_2 = 0 \tag{73}$$

$$(W_2/g)(d^2x_2/dt^2) - k_1x_1 + (k_1 + k_2)x_2 - k_2x_3 = 0 \tag{74}$$

$$(W_3/g)(d^2x_3/dt^2) - k_2x_2 + k_2x_3 = 0 \tag{75}$$

Assume $x_1 = a_1 \sin \omega t$, $x_2 = a_2 \sin \omega t$, etc., and substitute in the equations. This eliminates the time by cancellation of the factor $\sin \omega t$. Adding the equations together gives

$$(\omega^2/g)(W_1a_1 + W_2a_2 + W_3a_3) = 0 \tag{76}$$

By using Eqs. (73) and (75), a_1 and a_3 can be eliminated from (74) giving

$$\omega^4 - \omega^2 g\left[k_1\left(\frac{1}{W_1} + \frac{1}{W_2}\right) + k_2\left(\frac{1}{W_2} + \frac{1}{W_3}\right)\right] + \frac{g^2k_1k_2(W_1 + W_2 + W_3)}{W_1W_2W_3} = 0 \tag{77}$$

The two natural frequencies can be found by solving this quadratic for ω^2.

A system with four masses would result in a cubic equation for ω^2, etc. For systems with four or more masses, however, it is quicker to find the roots by an iteration method. An estimated value is obtained for ω^2 (see numerical example on page 1823), and a_1 is taken as 1. Equation (73) gives

$$a_2 = a_1 - (\omega^2/k_1g)W_1a_1 \tag{78}$$

Adding Eqs. (73) and (74) gives

$$a_3 = a_2 - \frac{\omega^2}{g}\frac{1}{k_2}(W_1a_1 + W_2a_2) \tag{79}$$

In a general problem, for the $(r + 1)$th amplitude

$$a_{r+1} = a_r - \frac{\omega^2}{g}\frac{1}{k_r}(W_1a_1 + W_2a_2 + \qquad + W_ra_r) \tag{80}$$

Thus, all the amplitudes can be found successively. It is convenient to arrange the computations in a table. When all amplitudes have been found, the results can be checked in Eq. (76). If there are n masses, the equation becomes

$$\frac{\omega^2}{g}\sum_{r=1}^{r=n} W_ra_r = 0 \tag{81}$$

Unless the estimated value used for ω^2 was exactly equal to one of the natural frequencies, Eq. (81) will not give a zero result. The indicated sum may be called the "remainder force," as it represents an external force which would have to be applied to the system to maintain the assumed vibration.

If the assumed value of ω is between zero and the first natural frequency, the remainder force is positive because of the positive value assigned to a_1. For values of ω between the first and second natural frequencies, the remainder force is negative, etc. Thus the sign of the remainder force indicates whether ω should be increased or decreased for the next trial. An experienced computer can often arrive at the correct result for the first natural frequency in two or three trials. The higher frequencies can be found by exactly the same procedure, but the number of trials tends to increase because the rate of variation of the remainder force with ω^2 is much greater.

Example: Consider the system shown in Fig. 30-35. In the fundamental mode of vibration, there can be only one "node" (stationary point). Since this will probably occur near the center of the train, a simplified two-mass system (Fig. 30-36) is formed by combining W_1 with W_2, W_3 with W_4, and connecting with a spring somewhat softer than k_2. From Eq. (51), the natural ω^2 of Fig. 30-36 is

$$\omega^2 = \frac{kg(W_a + W_b)}{W_aW_b} = \frac{30 \times 386(500 + 600)}{500 \times 600} = 42.4 \tag{82}$$

Table 30-2 shows the rest of the computation.

Table 30-2

W	$\frac{W\omega^2}{g}$	a	$\frac{W\omega^2a}{g}$	$\sum\frac{W\omega^2a}{g}$	k	$\frac{1}{k}\sum\frac{W\omega^2a}{g}$
First trial: $\omega^2 = 42.4$						
300	32.9	1.000	32.9	32.9	50	0.659
200	22.0	0.341	7.5	40.4	70	0.577
100	11.0	−0.236	−2.6	37.8	40	0.945
500	54.9	−1.281	−70.4	−32.6		
Fifth trial: $\omega^2 = 32.0$						
300	24.9	1.000	24.9	24.9	50	0.498
200	16.6	0.502	8.34	33.24	70	0.475
100	8.3	0.027	0.22	33.46	40	0.837
500	41.4	−0.810	−33.5	0		

Hence the lowest natural frequency is $\sqrt{32}/2\pi = 0.900$ cps, or 54 cycles per min.

Torsional Vibrations. In most machinery, power is transmitted by rotating shafts. Torsional (rotary) vibrations are likely to occur, especially if the motor or the load has nonuniform characteristics.

Figure 30-37 shows the simplest system of this kind, in which a shaft is fixed at one end and carries a flywheel at the other. The moment of inertia of the flywheel is I: in engineering units it is usually measured in pounds-inches-seconds squared. As an example, the moment of inertia of a homogeneous cylinder about its axis is $\frac{1}{2}mr^2$, where m is the mass of the cylinder and r is the radius. In vibration work, as in machine design generally, lengths are usually measured in inches. Thus, mass = weight/g, with weight in pounds and g = 386 in. per sec per sec. This gives the engineering unit of mass as pound-(seconds)^2 per inch; and with r in inches, the units of I will be pound-inch-(seconds)^2.

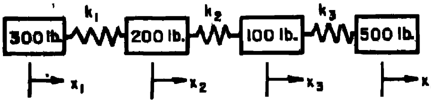


FIG. 30-35. Analysis of nonuniform trains (more than three masses).

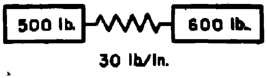


FIG. 30-36. Approximation for fundamental mode of Fig. 30-35.

k_t is the torsional spring constant of the shaft, measured in inch-pounds per radian of twist. For a solid cylindrical shaft of length L and diameter d, $k_t = \pi d^4 G/32L$, where G is the shear modulus of the material. For steel, G is often taken as 12×10^6 lb per sq in. although the accurate value is usually between 10.8 and 11.5.

For a hollow shaft, $\pi d^4/32$ is replaced by $\pi(d_0^4 - d_i^4)/32$. Formulas for noncircular shafts can be found in books on strength of materials or theory of elasticity. Much work has been done in testing and developing approximate formulas for torsional stiffness of crankshafts.[1,2,22] If found from a test, k_t = (torque, inch-pounds divided by angle of twist, radians).

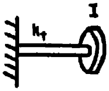


FIG. 30-37. Torsional vibration (one degree of freedom).

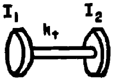


FIG. 30-38. Torsional vibration (shaft with two disks).

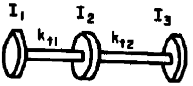


FIG. 30-39. Torsional vibration (shaft with three disks).

If ϕ is the angle of twist from equilibrium position in Fig. 30-37, the equation of motion may be found by considering the moments which act on the flywheel. This gives

$$I(d^2\phi/dt^2) + k_t\phi = 0 \tag{83}$$

This is exactly the same as Eq. (1), with W/g replaced by I and k by k_t. Thus, the natural frequency for torsional vibrations is

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k_t}{I}} \tag{84}$$

All the methods and results developed for linear vibrations can now be applied to torsional cases. For a shaft with two disks (Fig. 30-38), the natural-frequency formula is

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k_t(I_1 + I_2)}{I_1 I_2}} \quad \text{cps} \tag{85}$$

For a three-disk arrangement (Fig. 30-39), there are two values of $f_n = \omega/2\pi$, with ω^2 being found from the quadratic:

$$\omega^4 - \omega^2\left[k_1\left(\frac{1}{I_1}+\frac{1}{I_2}\right)+k_2\left(\frac{1}{I_2}+\frac{1}{I_3}\right)\right]+\frac{k_1k_2(I_1+I_2+I_3)}{I_1I_2I_3}=0 \qquad (36)$$

For more than three disks, the tabular method of iteration can be used as in Table 30-2, with W/g replaced by moments of inertia I and spring constants k by torsional constants k_t. Such tables are known as "Holzer tables."

Installations of internal-combustion or diesel engines, or reciprocating steam engines, are especially subject to trouble from torsional vibration because of the pulsations inherent in using cylinder pressures to create turning moments. Large vibrations can build up when the operating speed coincides with a natural frequency. Also, since the torque is not a pure sine wave, it has harmonic components of two, three, etc., times the fundamental frequency. Thus, vibrations can occur not only when running at the natural frequency (often called the "critical speed") but at many fractions of this speed. For a steam engine, "2-cycle" engine, or compressor, there is 1 cycle of torque per revolution; for 4-cycle gasoline engines, the cycle of torque requires two revolutions.

If there are several cylinders, the combined effect of all the torques must be considered. For some harmonic orders, the torques may cancel provided running conditions are the same in all cylinders. However, there will always be some orders where the effect on vibration is cumulative. Consider, for example, a 6-cylinder 4-cycle engine installation in which one natural frequency for torsional vibration is 2,100 cycles per min. For each cylinder, the torque cycle requires two revolutions. The sixth harmonic will then have 6 cycles in two revolutions or 3 cycles in one revolution. This is therefore known as the "third-order" component. The third-order components of all the cylinders in the above engine will be in unison, regardless of firing order, because there are also three firings per revolution. Hence, heavy torsional vibration is likely to occur with the engine running at 2,100/3 = 700 rpm since the third-order torques will give a disturbing frequency of 2,100 cycles per min at this speed. Vibration is also likely at speeds which are multiples of 700 rpm.

The main objection to torsional vibration is, of course, the likelihood of breaking crankshafts or other components owing to high fatigue stresses. If there are gears in the drive, the "chattering" caused by torsional vibration can be very destructive. Many breakdowns caused by such vibrations have occurred in the past.

Textbooks on vibration[1,2] give extended consideration to this problem. Methods of reducing a multicylinder engine to an idealized system of shafts and disks are discussed, and examples of natural-frequency calculations are shown. Fairly accurate methods are available for analyzing the torque into its harmonic components, estimating the amount of damping (from elastic hysteresis, propeller action, etc.), and thus computing the maximum amplitude of torsional vibration which will occur at a given critical speed.

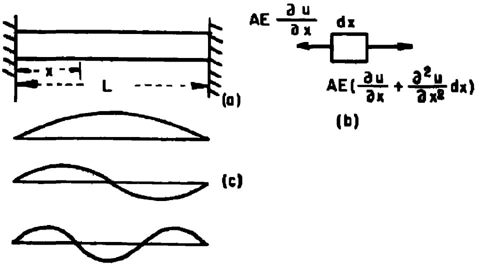
In most cases, the inherent damping in the system is too small to prevent dangerous fatigue stresses from being built up by vibration at one or more of the critical speeds. If the engine is required to operate only over a restricted range of speeds or at a constant speed, it is possible to adjust the elastic and mass relations so that no important critical speed falls within the operating range. On existing installations, it may be possible to rearrange the severity of several of the critical speeds by changing the firing order of the engine. However, if the engine must operate over a wide speed range, the only effective method of controlling vibration amplitudes is by the use of some sort of damper. Many devices have been tried for this purpose. One of the oldest is the "Lanchester damper," consisting of two flywheels mounted on the engine shaft and coupled to it only through friction on spring-loaded pads. When the engine runs smoothly, the disks run with it. If a vibration occurs, superposed on regular running, the disks tend to maintain constant speed owing to their inertia. The relative motion absorbs energy by a rubbing action, thus reducing the amplitude of vibration. While the device is effective when properly designed and adjusted, it has the weakness that the friction must be held within fairly narrow limits. If the disks are too loose or too tight, little damping can be secured. Thus an adjustment and maintenance problem exists. The classic example of this was a near disaster to the dirigible Graf Zeppelin: on one attempted oceanic flight, most of the airship's engines suffered broken crankshafts because an uninformed mechanic had excessively tightened the vibration dampers.

As mentioned in the section on the dynamic vibration absorber, a "torsional pendulum" or floating counterweight can be designed to eliminate vibration due to a specified torque harmonic at any speed of the engine. This has been extensively applied in aircraft engines.[2]

In the automobile industry, dampers using rubber in shear between a pair of disks and the crankshaft are very popular. The rubber supplies both spring and damping action. Again, proper design and adjustment are essential for correct functioning of such a damper.

VIBRATIONS OF BODIES WITH DISTRIBUTED MASS AND ELASTICITY

Previously, systems with concentrated masses and weightless springs have been the only types considered. A system with distributed mass can be approximated by dividing it into a number of parts, lumping the mass and spring factors, and treating it as a problem in several degrees of freedom. However, by means of partial differential equations, the completely distributed case can be solved for many cases where the geometric form is sufficiently simple.



x
L
(a)
$AE\frac{\partial u}{\partial x}$
dx
$AE\left(\frac{\partial u}{\partial x}+\frac{\partial^2 u}{\partial x^2}dx\right)$
(b)
(c)

Fig. 30-40. Longitudinal vibration of uniform bar with fixed ends.

Longitudinal Vibrations in Bars, Coil Springs, and Gas or Liquid Columns. As an example of longitudinal vibrations, consider longitudinal vibrations in a uniform elastic bar. The solution, when obtained, can be applied immediately to axial vibrations in coil springs.

Figure 30-40 shows a uniform elastic bar with fixed ends and coordinates. L = length, A = cross-sectional area, E = modulus of elasticity, w = density of material, x = distance from one end to any point. Let u = displacement of the cross section at x from its position of static equilibrium. During a vibration, each cross section of the bar moves through a certain amplitude, which varies along the length of the bar. Thus u is a function of both x and time t.

If such a bar is subjected to an axial pull P, the elongation is obtained from the formula $\delta = PL/AE$. Thus, $P = AE\delta/L$. δ/L is the unit strain (inches per inch); in terms of u, $\delta/L = \partial u/\partial x$. (If u = constant, all sections of the bar move the same distance, and there is no strain.) Thus the elastic force P in the bar equals $AE\partial u/\partial x$. Figure 30-40*b* shows the forces acting on an element of the bar of length dx. The elastic force to the left is $AE\partial u/\partial x$. The force to the right is $AE[(\partial u/\partial x) + (\partial^2 u/\partial x^2)\,dx]$, reflecting the change in u with x. Applying the principle that force equals mass times acceleration gives

$$AE\frac{\partial^2 u}{\partial x^2}\,dx = \frac{wA\,dx}{g}\,\frac{\partial^2 u}{\partial t^2}$$

Removing common factors,

$$\frac{\partial^2 u}{\partial x^2} = \frac{w}{Eg}\,\frac{\partial^2 u}{\partial t^2} \tag{87}$$

Partial differential equations of this sort are solved by assuming that u is a product of two functions, one containing x only and the other t only. Thus, let

$$u = U \sin \omega t$$

where U is a function of x. Substituting in Eq. (87) and canceling the common factor $\sin \omega t$,

$$(d^2U/dx^2) + (w\omega^2/Eg)U = 0 \tag{88}$$

Letting $\Omega = w\omega^2/Eg$, the solution of this ordinary differential equation is known to be [see Eqs. (1) and (2)]

$$U = C_1 \cos \Omega x + C_2 \sin \Omega x \tag{89}$$

where C_1 and C_2 are constants which must be adjusted to fit the end conditions of the bar. In Fig. 30-40, it is necessary to have $U = 0$ when $x = 0$ and when $x = L$. The first condition makes $C_1 = 0$. For the second, $C_2 = 0$ would give $U = 0$, a nonvibrating state of the bar. The alternative is to take $\sin \Omega L = 0$. This gives $\Omega L = \pi, 2\pi, 3\pi$, etc., from which the unknown frequency factor ω can be found. Thus, the natural frequencies for longitudinal vibration of the bar are

$$f_s = \frac{\omega}{2\pi} = \frac{s}{2L}\sqrt{\frac{Eg}{w}} \qquad \text{where } s = 1, 2, 3, \ldots . \tag{90}$$

Figure 30-40c shows the mode or amount of vibration at points along the length of the bar. Since $U = C_2 \sin s\pi x/L$, the constant C_2 gives the maximum vibrational amplitude (which is determined when the vibration is started). For the fundamental mode, U is given by half of a sine wave. At the second frequency, U is a full sine wave; hence there is a node at the center of the bar, and the two ends always have displacements in opposite directions.

As an example, when Eq. (90) is applied to the case of a steel bar 6 in. long, $E = 30 \times 10^6$ psi, $g = 386$ in. per sec per sec, $w = 0.284$ lb per cu in., giving $f = 16{,}800$ cps for the fundamental frequency. This is about six octaves above middle C on the musical scale.

The practical applications of Eq. (90) occur most often in two cases: the "organ pipe," where the natural frequency of a column of gas or liquid is desired, and the coil spring. For a gas, the modulus of elasticity may be determined from the gas laws. If the cross section A of the pipe is constant, the unit strain

$$dx/L = A\,dx/AL = dv/v$$

where v is the volume. For a small change dp in the pressure, the modulus of elasticity or "spring constant" may be found from

$$E = \text{change in stress/change in strain} = -v\,dp/dv$$

The minus sign is used to make the modulus a positive quantity, since dp and dv will always have opposite signs. Since the alternations of pressure in the organ pipe occur very rapidly, the process will be adiabatic rather than isothermal. Using pv^n = constant, $E = -v\,dp/dv = pn$. For air at atmospheric pressure, $n = 1.4$ and $p = 14.7$ psi, giving $E = 20.6$ psi.

For the coil spring, let d = wire diameter, D = mean coil diameter, G = shear modulus, N = number of active turns, w = density of wire material, lb per cu in.

$$\text{Spring constant } k \text{ of the coil spring} = \text{load/deflection} = Gd^4/8ND^3 \tag{91}$$

Let W = weight of active turns of the spring. Then (approximately)

$$W = (\pi d^2/4)(\pi DN)(w)$$

Equation (90) must be put in terms of k and W for the uniform bar. Since $k = AE/L$ for the straight bar and $W = wAL$, Eq. (90) can be written as

$$f = \frac{s}{2}\sqrt{\frac{AEg}{AwL^2}} = \frac{s}{2}\sqrt{\frac{kg}{W}} \tag{92}$$

Introducing k and W for the coil spring gives

$$f = \frac{s}{2\pi}\frac{d}{ND^2}\sqrt{\frac{Gg}{2w}} \qquad \text{where } s = 1, 2, 3, \ldots, \tag{93}$$

as the natural frequencies for a coil spring fixed at both ends. The frequency is independent of any initial load (compression or tension) which may exist in the spring. As most springs are made of steel, let $G = 11.5 \times 10^6$ psi, $g = 386$ in. per sec per sec, $w = 0.284$ lb per cu in. The lowest natural frequency of a steel coil spring is thus found to be

$$f_1 = 14{,}100d/ND^2 \qquad \text{cps} \tag{94}$$

When springs are used in high-speed machinery, malfunctioning may occur due to "surges," or axial vibrations at the natural frequency of the spring. If the spring is used to maintain pressure on a cam follower, as in poppet valves for internal-combustion engines, the cam lift curve is ordinarily not simple harmonic. It will therefore contain harmonics at multiples of the fundamental cam frequency. In such a case it is considered good practice to have the natural frequency of the spring from 15 to 20 times the maximum operating frequency of the cam. This may require working the spring material at the maximum shearing stress allowable under the given working conditions, as can be readily shown. The maximum shearing stress in the spring is

$$s_{\max} = (8PD/\pi d^3)C \tag{95}$$

where C is the "Wahl factor," or stress-concentration factor. For well-designed springs, C does not vary greatly. The spring rate of the spring is given by Eq. (91). In most applications, the values of k and the maximum load P will be set by the duty which the spring must perform. Using Eq. (91) to rewrite Eq. (95) gives

$$s_{\max} = \frac{PCG}{\pi k}\frac{d}{ND^2} \tag{96}$$

Equation (94) shows that to raise the natural frequency of a steel spring requires an increase in the quantity d/ND^2. If P and k are held at constant values, it follows from Eq. (96) that the maximum stress will go up in the same proportion as the natural frequency.

Many other cases of free vibration of elastic bodies have been solved. Results for the commonly encountered shapes are tabulated at the end of this section (pages 1831 to 1835).

General Methods for Difficult Problems. Mathematicians have solved many cases in addition to those recorded above, such as movement of rigid bodies suspended in various ways, vibrations in elastic spheres, cylinders, plates, disks, shells and tubes, and vibration of air or gases in containers of simple shapes.[34 to 37] In general, the results are not presented in readily available formulas.

Energy Method. Rayleigh's Method. In many cases, the "energy method" is used. A vibration represents a continual interchange of energy between moving masses and elastically strained members. If friction is neglected, the kinetic energy at mid-position of a vibration must equal the strain (potential) energy at the end positions. By assuming a simple harmonic motion, calculating the two energies and equating them, the natural frequency can be found.

"Rayleigh's method" consists in extending this procedure to elastic bodies in general. In such a case, all parts of the body do not vibrate with the same amplitude, but the true shape or mode is not known in advance. Rayleigh[34] developed the procedure of assuming some reasonable shape, calculating the energies, and in this way finding the natural frequency. If several shapes are tried, the one giving the lowest frequency is most nearly correct. This is because the true shape is the one in which the body is distorted with the least amount of strain energy. Ritz added the refinement of assuming the trial shape in the form of an equation with a parameter, then varying the parameter until a minimum natural frequency is obtained.

"Critical Speed" for Lateral Vibration of Rotors. Consider a case in which it is desired to find the lowest natural frequency for lateral vibration of a rotor, as in a steam turbine. Usually, the stiffness varies along the shaft. The latter is divided into a sufficient number of sections, and the weight and stiffness factors of each section are computed. The total deflection at each section is then found, this being a problem in strength of materials. If it is assumed that the shape of the lowest-frequency vibration is the same as this statical deflection curve, the frequency formula is

$$f_1 = \frac{1}{2\pi}\sqrt{\frac{g\Sigma W_i\delta_i}{\Sigma W_i\delta_i^2}} \qquad \text{cps} \qquad \text{where } i = 1, 2, \ldots, n \tag{97}$$

When converted to cycles per minute, this is frequently called the "critical speed."

Dynamically Equivalent Systems. Frequently, complicated elements can be reduced to simpler ones which are dynamically equivalent. In Fig. 30-1, if a coil or an axially loaded bar is used for the spring, its weight can be included in the calculations by adding one-third the spring weight to w. If the spring action is obtained by bending, the factor varies from about one-quarter to one-half, depending on the shape assumed for the vibration. In analyzing torsional vibrations, the reduction of rotating machinery and engines to a system of simple shafts and disks has been mentioned. Kemler and Freberg[34] give numerous examples of reduction to dynamically equivalent systems. When machinery contains shafts geared at different speeds, an equivalent system all on one shaft can be set up.

MORE COMPLICATED CASES OF SEVERAL DEGREES OF FREEDOM

There are many practical problems of vibration in systems of several degrees of freedom in which the foregoing simplified methods of analysis cannot be applied. The following list is representative of this class:

1. Higher frequency vibrations of a nonuniform beam, as in ship hulls, steam turbine rotors, etc.
2. Combined bending and torsional vibrations in airplane propellers, wings, fuselages, etc.
3. Two- or three-dimensional vibrations of an automobile structure, with sprung and unsprung weights and passengers on spring seats

Powerful mathematical methods have been developed and solutions obtained for typical problems of the above types. The treatments are too lengthy to reproduce here, but References 1, 2, and 23 to 26 listed on pages 1836 and 1837 give the best available procedures. In some cases, punched-card computing machines can be used.

As an example of an unsolved problem, the two- or three-dimensional motions of a chain or cable suspended at two points may be mentioned.[27,28]

SELF-INDUCED VIBRATIONS

Forced vibrations previously discussed have been caused by rotating or reciprocating unbalance, magnetic action, etc. If such a system is clamped, it still experiences sinusoidal disturbing forces. However, there are many vibrating systems in which, deliberately or accidentally, the forces driving the motion are caused to fluctuate by the motion itself. When the motion stops, there is no independent periodic force. Such cases are called self-induced vibrations. There are many familiar examples:

1. The bowing of a violin string, the "chattering" of brakes and cutting tools, and the screech of dry rubbing surfaces are cases of vibration due to friction. In each case, a driving member is caused to move at fairly uniform speed over the vibrating element with dry friction between the two. Such a friction force has the characteristic that it varies inversely with the rubbing speed. When the two elements move in the same direction, the relative rubbing speed is less, and a large force exists to help the motion. When the vibrating element reverses, the relative velocity is greater; and while friction now opposes the motion, its magnitude is less. Thus, over a complete cycle there is a net input of energy to the vibrating system, and the motion maintains itself. In preventing such noises, lubrication is effective because liquid friction has an opposite characteristic: it varies directly with velocity.

2. The waving or flapping of a flag and the humming noise of telephone wires exposed to wind are due to formation of eddies in the stream of air. These are discharged alternately from the two sides of the object, giving an alternating force which maintains the vibration. Such motions are occasionally serious,[4] as in the "fluttering" of airplane wings and tails, or the collapse of the Tacoma Narrows Bridge.

3. The flow of liquids or gases through spring-loaded valves will nearly always cause vibrations of the valve member to occur. Figure 30-41 shows air being forced at a uniform rate into a closed chamber, the only outlet being partially closed by a reed at B. Assume the reed is vibrating, causing a variation in the size of the opening through which air escapes. When the reed is passing through center in an outward or opening direction, the pressure in the chamber is higher than average, as the outlet has recently been restricted. When the reed returns inward, the pressure is lower as the opening has been larger. Thus, the total effect is to maintain the vibration of the reed. In order to suppress the vibration, it is necessary to apply some sort of damping resistance to motion of the reed.

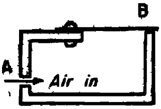


FIG. 30-41. Vibration of musical reed or spring-loaded valve.

Vibration can also occur with air flowing in the opposite direction. In this case, air flowing under the reed causes a suction by Bernoulli's principle. This draws the reed down and closes the passage. The pressures on the two sides of the reed then equalize; the reed moves upward, and the cycle repeats.

The same effect can be created by a loose obstruction. The stream carries the object into a position in which it cuts off the flow. Under static conditions, pressures equalize, and the obstruction falls away, permitting the cycle to repeat. Noisy vibrations of this sort often occur in faucets with loose washers.

4. Automatic control systems on temperature, speed of rotation, etc., frequently "hunt" or fluctuate about the desired control point rather than settling to a steady value. This is because there is a time lag between the controller and the quantity controlled, leading to overshooting and overcorrection. Performance can be improved by decreasing the time lags and by introducing damping at the proper point of the system.

All the above matters are more thoroughly treated by Den Hartog.[2]

NONLINEAR SYSTEMS

Frequently, vibration problems involve a spring whose force cannot be adequately represented by k times the displacement. k may be a function of time, as in torsional vibrations of the connecting-rod drive systems of locomotives; or the spring may become relatively softer or harder with deflection. Forced vibration of systems of the latter type can best be analyzed by the methods of Rauscher.[32] Figure 30-42 shows a resonance curve for a system with a "hard" spring. Free vibrations are given by the curve AB; frequency will increase with amplitude because of increasing stiffness in the spring. Forced vibrations under increasing speed will follow the curve CG to D. For further increase in speed, the amplitude drops suddenly to E, and the system becomes relatively quiet. Under decreasing speed, the amplitude follows EF, then suddenly jumps to G, then moves down to C. It never becomes as large as the peak values for increasing speed.

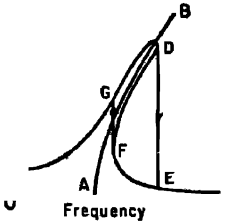


FIG. 30-42. Forced-vibration characteristics of system in which spring rate increases with deflection.

For mathematical treatments of the time-variable and nonlinear systems, the general books of Timoshenko and Den Hartog[1,2] should be consulted.

NOMENCLATURE

NOTE: In vibration work, lengths are usually in inches, weights and forces in pounds, and time in seconds. Mass = weight/g = pounds/386, since g = 386 in. per sec per sec

A = cross-sectional area of a bar or surface area of a membrane
a, b = dimensions of rectangular membrane or plate
c = damping constant = force/velocity = lb-sec/in.
c_c = critical damping = $2\sqrt{kw/g} = 2(W/g)\omega_n$
db = decibels (measurement of sound intensity)
δ = deflection of elastic element
δ_{st} = statical deflection = W/k
D = plate constant = $Eh^3/[12(1-\mu^2)] = 2.75 \times 10^6 h^3$ for steel
E = modulus of elasticity in tension or compression. For a gas, $E = pn$, where p = pressure, and n is the adiabatic exponent in pv^n = const. E = 20.6 lb/sq in. for atmospheric air
f = frequency, cps
f_n = natural frequency of a vibrating system, cps
f_1 = lowest natural frequency
F_0 = maximum dynamic force transmitted to foundation by vibrating system
g = acceleration of gravity = 386 in./sec/sec
G = modulus of elasticity in shear
h = thickness of a plate, in.
k = spring rate, lb/in. of deflection
k_t = torsional spring rate = in.-lb/radian
L = length of bar or string
N = revolutions per minute of a rotor
P_0 = maximum value of disturbing or shaking force on a machine
r = radius of a circle or mean radius of a ring
$\bar{r}$ = radius from swing point to center of gravity of a pendulum
s = 1, 2, 3, etc. = successive modes of vibration
TR = transmissibility ratio of a suspension = force on foundation/force on machine = F_0/P_0
t = time, sec
T = period of a vibration, sec, or tension in a cord, lb
T_1 = tension, lb/lin in. around the boundary of a membrane
v = velocity = dx/dt; velocity of sound in a material = $\sqrt{Eg/w}$
W = weight, lb
w = density of a material, lb/cu in. = 0.284 for steel
w_1 = weight of string or bar, lb/lin. in.
w_u = unit weight of a membrane or plate, lb/sq in.
x = displacement of a vibration; velocity = dx/dt and acceleration = d^2x/dt^2
μ = Poisson's ratio = 0.3 for steel
ω = angular velocity, radians/sec; $\omega = 2\pi N/60$
$\omega_n = \sqrt{kg/W}$

NATURAL FREQUENCIES OF ELASTIC BODIES

(Natural frequencies of assembled systems are given in the preceding text.)

Stretched Strings.

String of length L, tension T:

$$f_n = (s/2L)\sqrt{Tg/w_1} \qquad \text{where } s = 1, 2, 3, \ldots \tag{98}$$

Chain or Cord Suspended by One End.

Uniform chain or flexible cord of length L, suspended by one end, hanging vertically (Fig 30-43):

$$y = CJ_0(z) \qquad f_n = (z_s/4\pi)\sqrt{g/L} \qquad z = z_s\sqrt{x/L} \tag{99}$$

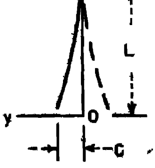


FIG. 30-43. Free vibration of chain or cord suspended by one end.

$J_0(z)$ is Bessel's function of order zero, and z_s represents the successive roots of $J_0(z) = 0$.[27,31] The first four values of z, are 2.4048, 5.5201, 8.6537, and 11.7915. For higher modes, z_s approaches $\pi(s - 0.25)$.

Longitudinal Vibrations in Uniform Bars or Gas Columns of Length L. Ends both fixed or both free:

$$f_n = (s/2L)\sqrt{Eg/w} \qquad \text{where } s = 1, 2, 3 \ldots \tag{100}$$

One end fixed, one free:

$$f_n = (s/4L)\sqrt{Eg/w} \qquad \text{where } s = 1, 3, 5, \ldots \tag{101}$$

One end fixed, other end attached to concentrated weight W_2. If weight of bar $wAL \leq W_2$,

$$f_n = \frac{1}{2\pi}\sqrt{\frac{AEg}{L[W_2 + (wAL/3)]}} \tag{102a}$$

If $wAL > W_2$, use

$$f_n = (\beta/2\pi L)\sqrt{Eg/w} \tag{102b}$$

where β is a root of $\beta \tan \beta = wAL/W_2$ (see following table from Reference 1).

$\frac{wAL}{W_2}$	0.01	0.10	0.30	0.50	1.0	2.0	5.0	10.0	∞
β	0.10	0.32	0.52	0.65	0.86	1.08	1.32	1.42	$\pi/2 = 1.57$

Torsional Vibration of Uniform Bars.

Torsional vibration of uniform bars of circular cross section and length L. (103) Replace E by G in cases (100) to (101).

Lateral Vibration of Uniform Bars of Length L:

$$f_n = (\alpha^2/2\pi L^2)\sqrt{EIg/w_1}$$

Both ends simply supported (hinged-hinged):

$$\alpha^2 = \pi^2 s^2 = 9.87, 39.48, 88.83, 157.9, 246.7, \ldots \tag{104}$$

Both ends free; bar lying on frictionless surface or supported at nodal points. $\cos \alpha \cosh \alpha = 1$:

$$\alpha^2 = 22.37, 61.67, 120.91, 199.8, 298.6, \ldots \tag{105}$$

If bar is floated in liquid, as a ship hull, some allowance should be made for weight of water moved by the vibration.

Both ends built in (fixed-fixed). Same results as Eq. (105). (106)

Cantilever (fixed-free). $\cos \alpha \cosh \alpha = -1$:

$$\alpha^2 = 3.516, 22.03, 61.70, 120.9, 199.8, 298.6, \ldots \tag{107}$$

One end fixed, other end hinged. $\tan \alpha = \tanh \alpha$:

$$\alpha^2 = 15.42, 49.97, 104.24, 178.3, 272.0, \ldots \tag{108}$$

One end hinged, other end free. Same results as Eq. (108). (109)

Wedge (Fig. 30-44)

Lateral, in plane of paper:

$$f_1 = \frac{0.2442d}{L^2}\sqrt{\frac{Eg}{w}} = 49{,}300\frac{d}{L^2} \qquad \text{for steel} \tag{110}$$

Perpendicular to plane of paper:

$$f_1 = \frac{0.3266t}{L^2}\sqrt{\frac{Eg}{w}} = 65{,}950\frac{t}{L^2} \qquad \text{for steel} \tag{111}$$

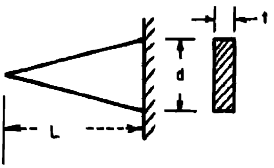


FIG. 30-44. Vibrations of wedge (large end fixed).

Lateral vibration of conical bar, length L, base diameter d, built in at base:

$$f_1 = \frac{0.3469d}{L^2}\sqrt{\frac{Eg}{w}} = 70{,}000\frac{d}{L^2} \qquad \text{for steel} \tag{112}$$

Circular Ring (Fig. 30-45)
Pure radial vibration:

$$f_n = \frac{1}{2\pi r}\sqrt{\frac{AEg}{w_1}} = \frac{1}{2\pi r}\sqrt{\frac{Eg}{w}} = \frac{32{,}100}{r} \quad \text{for steel} \tag{113}$$

Pure torsional vibration:

$$f_1 = \frac{1}{2\pi r}\sqrt{\frac{EgI_x}{w(I_x + I_y)}} = \frac{22{,}700}{r} \quad \text{for steel, circular cross section} \tag{114}$$

Flexural vibrations in plane of ring:

$$f_n = \frac{1}{2\pi r^2}\sqrt{\frac{EgI_y}{wA}\,\frac{s^2(s^2-1)^2}{s^2+1}} \tag{115}$$

where $s = 2, 3, 4, \ldots$

Flexural vibrations perpendicular to plane of ring:

$$f_n = \frac{1}{2\pi r^2}\sqrt{\frac{EgI_x}{wA}\,\frac{s^2(s^2-1)^2}{s^2+1+\mu}} \tag{116}$$

μ = Poisson's ratio, and $s = 2, 3, 4, \ldots$

Free-free cut rings.[30] (117)

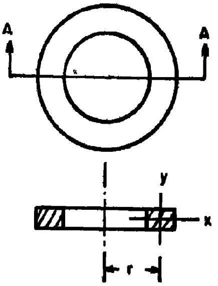


Fig. 30-45. Circular ring.

Membranes

Circular membrane. Let d = number of nodal diameters and c = number of nodal circles, including boundary.

$$f_n = (F_{dc}/r)\sqrt{gT_1/w_a} \tag{118}$$

where values of F_{dc} appear in Table 30-3.

Table 30-3. Values of F_{dc}

c	$d = 0$	$d = 1$	$d = 2$	$d = 3$	$d = 4$	$d = 5$
1	0 3826	0 6099	0 8173	1 0153	1 2074	1 3974
2	0 8785	1 1167	1 3396	1 5534	1 7609	1 9639
3	1 3773	1 6191	1 8494	2 0717	2 2875	2 4987
4	1 8767	2 1204	2 3548	2 5821	2 8037	3 0211
5	2 3763	2 6213	2 8584	3 0892	3 3147	3 5364
6	2 8761	3 1220	3 3609	3 5942	3 8226	4 0475
7	3 3760	3 6224	3 8627	4 0081	4 3290	4 5563
8	3 8759	4 1226	4 3642	4 6010	4 8337	5 0632

Rectangular membrane:

$$f_1 = \frac{F}{a}\sqrt{\frac{gT_1}{w_a}} \qquad f_n = \frac{1}{2}\sqrt{\frac{gT_1}{w_a}\left(\frac{m^2}{a^2} + \frac{n^2}{b^2}\right)} \tag{119}$$

where $m = 1, 2, 3, \ldots$; $n = 1, 2, 3, \ldots$.

Dimensions	F
$a \times a$	0 7071
$1\ 5a \times a$	0 6008
$2a \times a$	0 5589
$3a \times a$	0 5270
$\infty \times a$	0 5000

Membranes of other shapes:

$$f_1 = H\sqrt{gT_1/w_aA} \tag{120}$$

Shape	*H*
Circle	0.6782
Semicircle	0.7644
Quadrant of circle	0.7243
60-deg sector of circle	0.7348
Square	0.7071
Rectangle 3 × 2	0.7359
Rectangle 2 × 1	0.7905
Rectangle 3 × 1	0.9129
Equilateral triangle	0.7598
Isosceles right triangle	0.7906

Plates

Rectangular plate, simply supported edges:

$$f_n = \frac{\pi}{2}\left(\frac{m^2}{a^2} + \frac{n^2}{b^2}\right)\sqrt{\frac{gD}{wh}} \qquad \text{where } m = 1, 2, 3, \ldots, n = 1, 2, 3, \ldots \tag{121}$$

Square plate, free edges:

$$f_n = (F/a^2)\sqrt{gD/wh} \qquad \text{where } F = 2.244, 3.272, 3.805, \ldots \tag{122}$$

Circular plate, clamped boundary, in vacuo:

c = number of nodal circles, including boundary

d = number of nodal diameters

$$f_n = (F/a^2)\sqrt{gD/wh} \tag{123}$$

Values of F

c	$d = 0$	$d = 1$	$d = 2$
1	1.625	3.377	5.545
2	6.331		
3	14.15		

In case plate is vibrating in a liquid of density w_L, the first frequency becomes

$$f_1 = \frac{1.625}{a^2\sqrt{1+\beta}}\sqrt{\frac{gD}{wh}} \qquad \text{where } \beta = 0.6689(w_L/w)\,(a/h)$$

For a circular plate with free edges, Eq. (123) may be used. The values of (124) are shown in the following table:

c	$d = 0$	$d = 1$	$d = 2$	$d = 3$
0	......	.....	0.836	1.946
1	1.4445	3.266	5.609	8.421
2	6.131	9.527		

For a circular plate, fixed at center, Eq. (123) may again be used. The (125) lues of F are shown in the following table:

c	$d = 0$	$d = 1$	$d = 2$	$d = 3$
0	0.597	0	0.836	1.946
1	3.328	3.266	5.609	8.421
2	9.658	9.527		
3	19.05			

At large amplitudes of vibration, frequencies increase slightly because of stretching in the middle plane of the plate.

Turbine Disks.

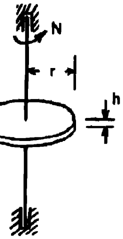


FIG. 30-46. Turbine disk.

Turbine disks, fixed at center (Fig. 30-16):

$$\left.\begin{aligned} f_n &= \sqrt{f_1^2 + f_2^2} \\ f_1^2 &= (F_1/r^4)(gD/wh) \quad \text{due to elastic force} \\ f_2^2 &= KN^2/10{,}000 \quad \text{due to rotation} \end{aligned}\right\} \tag{126}$$

	F_1				K			
c	$d = 0$	$d = 1$	$d = 2$	$d = 3$	$d = 0$	$d = 1$	$d = 2$	$d = 3$
0	0.357	0	0.734	3.95	0	2.78	6.53	11.25
1	11.09	10.69	30.65	71.94	9.17	16.53	24.86	34.17

SPRING CONSTANTS

Linear Spring Constants. Formulas for linear spring constants, pounds per inch deflection, and equivalent masses are listed in the following material.

Any linear spring:

$$k = \text{load, lb/deflection, in.} \tag{127}$$

Springs in parallel:

$$k_e = \Sigma k \tag{128}$$

Two springs in series:

$$k_e = k_1k_2/(k_1 + k_2) \tag{129}$$

Uniform bar axially loaded. Area A, length L:

$$k = AE/L \tag{130}$$

Add one-third weight of bar at each end.

Coil spring, axial deflection. Wire diameter d, mean coil diameter D_c, n active turns:

$$k = d^4G/8nD_c^3 \tag{131}$$

Add one-third weight of spring at each end.

Air under piston in cylinder. Length L, area A, pressure p:

$$k = 1.4Ap/L \tag{132}$$

Cantilever beam:

$$k = 3EI/L^3 \tag{133}$$

Add 23 per cent of beam weight at free end.

Beam loaded at center. Simple supports (hinged-hinged).

$$k = 48EI/L^3 \tag{134}$$

Add 48 per cent of beam weight at center.

Beam with built-in ends, loaded at center:

$$k = 192EI/L^3 \tag{135}$$

Add 37 per cent of beam weight at center.

Simply supported beam of length L, loaded at distance a from one end, b from other end:

$$k = 3LEI/a^2b^2 \tag{136}$$

Torsional Spring Constants. Formulas for torsional spring constants (inch-pounds per radian) and equivalent moments of inertia are given below.

Any torsional spring:

$$k_t = \text{moment, in.-lb/angle of rotation, radians} \tag{137}$$

Coil spring in torsion. Wire diameter d, mean coil diameter D_c, n active turns:

$$k_t = Ed^4/64nD_c \tag{138}$$

Uniform circular solid shaft. Length L, diameter d:

$$k_t = \pi Gd^4/32L \tag{139}$$

Add one-third moment of inertia of shaft at each end.

Hollow circular shaft. Length L, diameters d_1 and d_2:

$$k_t = \pi G(d_2^4 - d_1^4)/32L \tag{140}$$

Add one-third moment of inertia of shaft at each end.

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SECTION 31

VIBRATION CONTROL AND NOISE SILENCING

BY ROBERT L. BOOTH, P.E. AND DONALD H. VANCE

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VIBRATION CONTROL AND NOISE SILENCING

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NOMENCLATURE

C = coefficient of viscous damping = damaging force/velocity, lb-sec/in.
C_c = coefficient of critical damping = $2\sqrt{kW/g}$
$\dfrac{C}{C_c}$ = damping factor = $(c/2)\sqrt{g/kW}$
e = crank radius
f_N = natural frequency of spring support = $188\sqrt{1/\delta_{st}}$ cycles per min
f_D = disturbing frequency
F = magnitude of shaking force ($F \sin \omega t$)
F_T = force transmitted
F_N = force transmitted at resonance
g = acceleration of gravity = 32.2 ft/sec^2
k = spring constant W/δ_{st} lb per in.
r = frequency ratio f_D/f_N
t = time
T = period = $1/f$
T_D = period of damped free vibration with periodic damping
T_N = natural period
W = weight, lb
w = weight of reciprocating part
x = displacement from neutral position during vibration
x_{st} = F/k spring deflection due to static action of the magnitude F of the shaking force $F \sin \omega t$
x_{nst} = spring deflection due to static action of the shaking force at resonance
X = amplitude of forced vibration undamped
X_D = amplitude of forced vibration damped
X_N = resonant amplitude of forced vibration $F/c\omega_N$
δ_{st} = static deflection W/k, in.
ω = angular frequency
ω_N = natural angular frequency = $2\pi f_N$
ω_D = natural angular frequency damped

VIBRATION CONTROL

General Theory. Vibration problems occur in every field of engineering and in practically every type of plant or industry. While the presence of vibration is usually perceptible to the human senses, its source is not always easily determined. General vibration disturbances may be created by the operation of machinery within a plant, in which case the treatment of the problem might be classed as positive isolation, or the vibration may be transmitted to a plant from outside sources, in which case the treatment of the problem within the building might be defined as negative isolation.

Since it is not always possible or profitable to design machinery which will be in complete balance through its entire range of operation, it can be expected that practically every machine or piece of equipment containing moving parts will create vibration. Whether or not the vibration will be of sufficient intensity to become a hazard to the building structure, to personnel and adjacent equipment, or to neighboring localities can be determined only through thorough consideration of all aspects of the proposed installation.

Some of the benefits which may be derived from the control of vibration are listed and briefly described below.

1. *Reduction of Vibration Transmission.* By correct application of vibration-engineering principles and vibration control, a machine, which might ordinarily cause considerable transmission of unbalanced forces if mounted solidly, may in many cases be operated with such a reduction in transmitted force that its vibration cannot be detected by sensitive vibration instruments.

2. *Increased Efficiency and Lengthened Life of Machine.* Practical experiments have indicated that a valuable portion of the energy lost in transmitting vibration to a rigid supporting structure may be saved through the use of resilient mountings which periodically store rather than dissipate energy. The application of resilient supports also reduces the internal stresses within the structure of the machine by allowing freedom of motion resulting in increased machine life.

3. *Reduced Construction and Installation Cost.* If a vibrating machine is rigidly bolted to a foundation, the unbalanced forces causing the vibration will be transmitted to the foundation undiminished, and excessive stresses may be set up in the machine. Through the correct application of vibration control, the transmission of forces to the foundation and the creation of stresses within the machine can be greatly reduced. A considerable saving in construction and installation cost may, therefore, be realized through the employment of smaller foundations and lighter supporting structures. An additional saving is also realized through reduced maintenance of the equipment resulting from the relief of stresses within the vibrating machine.

4. *Improved Working Conditions.* Excessive vibration and the noise which generally accompanies it have a very decided influence on the nervous system and contribute materially to mental and physical fatigue. Such a condition may become detrimental to the health and efficiency of the personnel. The cost of vibration control will generally be recovered in a short time with an actual increase in personnel efficiency thereafter.

The correction of a vibration problem after an installation has been completed is not always simple or economical, and therefore, adequate consideration prior to the installation of machinery or other equipment in motion can often result in considerable saving in time and expenses. A basic knowledge of the theory of vibration and its control permits the plant engineer to predetermine the vibration-control requirements, if any, of a proposed installation and is helpful also in dealing with vibration problems already in existence.

Control of mechanical vibration is a specialized branch of engineering, and many texts and papers have been written on the subject. A bibliography of such material will be found at the end of this section. A brief explanation of the theory and application of the principles of vibration control is given in the following pages.

Definition of Vibration Terminology in Common Use. *Vibration* is a periodic motion in alternately opposite directions from the position of equilibrium of a machine, machine element, or other mass.

Periodic motion is a motion which repeats itself in all its characteristics at definite intervals of time or periods of vibration.

Free vibration is the harmonic motion of a body elastically supported after being displaced from its position of rest and released.

Forced vibration is the periodic vibration of an elastically supported system being acted upon by an external periodic force.

Steady state is a continuous repetition of a vibration cycle after an elastic system has been acted upon by a system of forces for sufficient length of time.

Transient state is the change in the vibratory motion of the system due to the application of changing external forces.

Natural frequency is the number of vibrations or cycles per unit of time that a freely vibrating body will undergo.

Forced frequency or disturbing frequency is the number of vibrations or cycles per unit of time which a body undergoes owing to the external forces acting upon it.

Frequency ratio is the ratio of forced frequency to natural frequency.

Resonance is the state reached when an elastically supported system is acted upon by an external force having the same frequency as the natural frequency of the system, resulting in increasing amplitudes of the system and increasing forces transmitted by the system.

Beats. When two vibrations of slightly different frequencies act upon a common system, the vibrations will at times add, increasing the amplitude of vibrations, and at other times cancel each other, reducing the amplitude of vibrations, and these increases and decreases in amplitude result in periodic beats.

Critical speed is a speed at which the disturbing frequency equals or is close to the natural frequency of the part or system, resulting in a resonant condition.

Amplitude is the maximum distance or displacement of a vibrating body from a neutral or mean position.

Damping is the introduction of friction into a vibrating system.

Isolation efficiency is the percentage of reduction of a disturbing force transmitted to a supporting structure.

Static conditions are conditions dealing with bodies at rest.

Dynamic conditions are conditions dealing with bodies in motion.

Vibration Instruments. The need for vibration instruments in solving a vibration problem depends a great deal on the nature of the disturbance and the factors to be determined in order to make an intelligent analysis of the problem. In the case of a disturbance caused by what might be considered as practically a simple harmonic motion, such as a single-cylinder vertical engine or compressor so balanced that the horizontal forces are very small in comparison to the vertical, conducting a vibration test would not be necessary since all the principal characteristics of the machine are known before the installation is made. The disturbing frequency will be equal to the constant speed of operation. The primary unbalanced force along the cylinder center line can easily be determined from the equation $F = (W/g)\omega^2e$

where W = weight of reciprocating parts, lb
g = acceleration of gravity (32.2 ft/sec^2)
ω = angular frequency, radians/sec
e = one-half stroke, in.

All the unbalanced forces and moments, both primary and secondary, for machinery such as reciprocating compressors, pumps, and engines can usually be obtained directly from the equipment manufacturer. With this information, plus the total weight of the equipment, the center of gravity, and over-all dimensions, the necessary calculations can be made to arrive at the solution to the problem of isolating the machine.

In many pieces of equipment, however, the disturbance may be caused by a combination of two or more simple harmonic vibrations or a combination of various types of vibrations, making it practically impossible to determine by calculation the characteristics of the disturbance resulting from the combination of all the various forces acting within the machine. In such cases the resulting characteristics of the disturbing vibration can be obtained through the use of vibration instruments.

The instruments most generally used for this type of work can be broken down into three general categories: (1) frequency measuring, (2) vibration measuring, and (3) acceleration measuring.

Frequency-measuring Instruments. Instruments used for determining disturbing frequencies in a system are nearly all based on the principle of resonance. The simpler types of instruments are known as tachometers and are useful for determining frequencies below about 6,000 cycles per min. They are in two general types. One contains a single adjustment reed which can be easily lengthened or shortened to correspond to the frequencies within its range by simply turning a knob. The other type, known as the Frahms tachometer, consists of a box containing several reeds, each having a slightly higher natural frequency than the next. With either type of instrument, the maximum amplitude of vibrating reed determines the disturbing frequency. At this point, a condition of resonance exists between the disturbing frequency and the natural frequency of the instrument. Where the frequency varies rapidly over a wide range, it becomes difficult to follow the changing frequency with this type of instrument. In most practical problems, however, it is sufficient to determine the lowest disturbing frequency in the system, and this can readily be found by use of the above instrument. Where it is necessary to have a complete knowledge of a complicated vibration wave shape, a recording instrument is used which produces a graphic record of the wave on a paper or film, thus permitting a detailed analysis. Another type of frequency-measuring instrument known as the stroboscope works on the principle by which the frequency of a flashing light thrown on a moving body is varied until it is in synchronization with the frequency of the body at which point the body appears to stand still.

Vibration-measuring Instruments. Vibration-measuring instruments are used for determining the amplitude of a vibration. A "seismic" instrument known as a vibrometer is ordinarily used for this purpose. It consists of a mass mounted on relatively soft springs inside a box.

The instrument is fastened or firmly held to the vibrating machine, and the ratio between the relative amplitude of the machine and the suspended weight within the instrument is observed. When the disturbing frequency is close to the natural frequency of the instrument, the relative amplitude observed will be large, and in order to determine the actual amplitude of motion of the vibrating machine, it will be necessary first to determine the disturbing frequency. Where the ratio between the disturbing frequency and the natural frequency of the instrument is large, the relative amplitude will be small, and the mass of the instrument will tend to stand still in space. The relative amplitude observed for high-frequency ratios, at least 2:1 or greater, can, for the most practical purposes, be considered equal to the actual amplitude of motion of the machine. In such cases, the instruments can be used for determining amplitude directly without taking into consideration the frequency. There are many types of amplitude meters or vibrometers intended for very rugged use such as might be expected in power plants, machine shops, and industry in general. They are easily read and are quite accurate, especially when used with high-speed machinery.

A more elaborate form of vibrometer containing a recording mechanism is known as a vibrograph. This instrument is often equipped with a timing device, and in operation, the relative amplitude is recorded on a moving strip of paper or film. This strip travels through the instrument at a constant known speed. The strip is marked with a standard timing wave or timing notches for the instrument. By comparison between the standard timing curve and the inscribed vibration curve, it is possible to determine both the frequency and amplitude of the vibrating machine.

Acceleration-measuring Instruments. It was pointed out that, in the case of vibrometers and vibrographs, accurate direct reading of amplitude can be made only where the disturbing frequency is high compared with the natural frequency of the instrument and that such a machine will increase in accuracy as the speed of the machine or disturbing frequency increases. Damping is therefore not necessary since the possibility of resonance does not exist. Instruments for measuring acceleration are designed on the opposite side of the resonant curve. Accelerometers are built somewhat similar to a vibrometer with the exception that stiff rather than soft springs are used for suspending the mass within the instrument. The object here is to have the natural frequency of the instrument higher than the disturbing frequency. For very small frequency ratios, the acceleration can be found directly by measuring the indicated amplitude and multiplying it by the square of the natural frequency of the instrument. Hence, for an impure vibration containing higher harmonics, the instrument will be subject to higher frequency than the one for which the acceleration is desired. It is therefore very possible to obtain a resonant condition, which, of course, is undesirable. Accelerometers are, therefore, generally constructed with a high degree of damping so as to prevent resonance and reduce the effect of the higher harmonics.

Theory of Vibration Control. By careful study and analysis, it is often possible to determine the cause of vibration and reduce it at its source by means of balancing. It is, of course, necessary to balance the machine both statically and dynamically in order to effect any appreciable amount of vibration reduction. Since it is impossible to balance completely any machine in operation, a certain amount of vibration is to be expected. The relative cost of balancing a machine to the point where the disturbing vibration is reduced to a tolerable amount, compared with the cost of isolation, is a determining factor in deciding the steps to be taken for correcting the problem. It usually develops that the cost of providing isolation is much less than the cost of eliminating the cause.

Efficient isolation is in general provided for by mounting the equipment in question on a resilient material having a natural frequency lower than the disturbing frequency.

The disturbing frequency is generally known and often equal to the speed of the machine, or it can be determined through the use of frequency-measuring instruments. The natural frequency of the isolating material is largely dependent upon the static deflection of the material due to the weight which it carries. The curve of Fig. 31-1 shows the relationship between static deflection and natural frequency and applies to all materials where the damping factor is small.

The equation of the curve of Fig. 31-1 based on an undamped single degree of freedom is

$$\omega_N = \sqrt{g/\delta_{st}} \qquad \text{radians/sec}$$

or

$$f_N = 3.14\sqrt{k/W} = 3.13\sqrt{1/\delta_{st}} \qquad \text{cps}$$

$$f_N = 188\sqrt{1/\delta_{st}} \qquad \text{cycles/min}$$

If viscous damping is introduced into the system by means of a dashpot or the equivalent, the equation becomes

$$\omega_{N(\text{damped})} = \sqrt{1 - (C/C_c)^2}\,\omega_N \qquad \text{radians/sec}$$

$$f_{N(\text{damped})} = \sqrt{1 - (C/C_c)^2}\,f_N \qquad \text{cps}$$

It can be seen from this curve that, for cases where it is required to provide a natural frequency in the isolation material of 100 cycles per min, a static deflection of approximately 3½ in. is required. While it is possible to obtain this large deflection with various types of material such as cork or rubber, it would be necessary to provide a considerable thickness of material for this purpose. Steel coil springs, on the other hand, can be designed for this deflection or even greater deflection without involving the use of an excessive amount of material. In general, steel-spring mountings are used where the required static deflection range from about ³⁄₁₀ to about 5 in., and for special cases, even a larger deflection can be provided. Fabricated rubber mountings are generally intended for small static deflections, usually not exceeding approximately ¼ in., although some rubber mountings are capable of providing as much as ½-in.

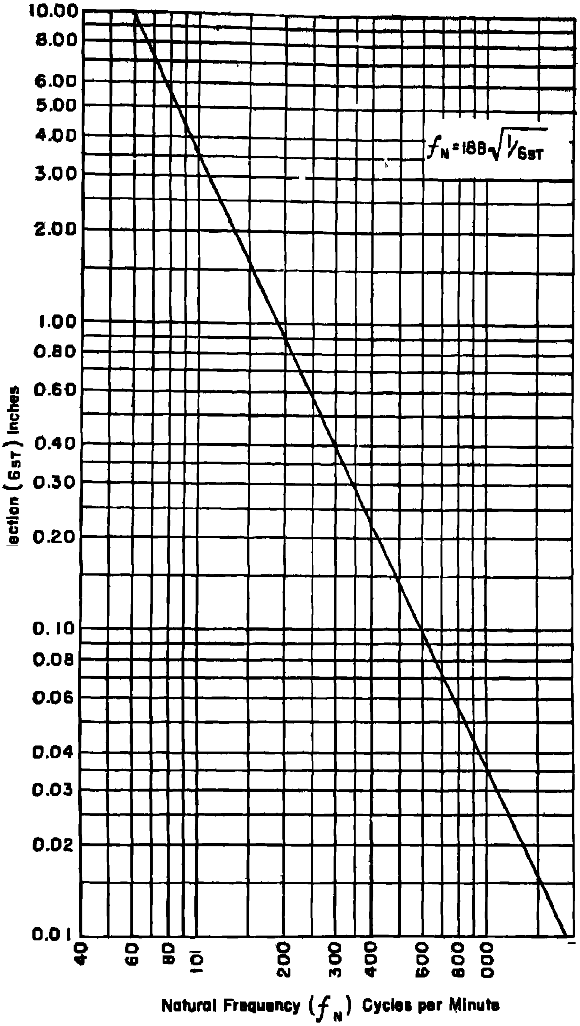


Fig. 31-1. Deflection-frequency relationship for undamped systems.

deflection. The static deflection provided through the use of cork is for the most part dependent upon the thickness of material that can be practically and economically employed. This would also hold true for the case of solid rubber applied in sheets under a foundation or machine and for other sheet material similarly employed.

The usual application is for cases where the static deflection required does not exceed about $\frac{3}{10}$ in.

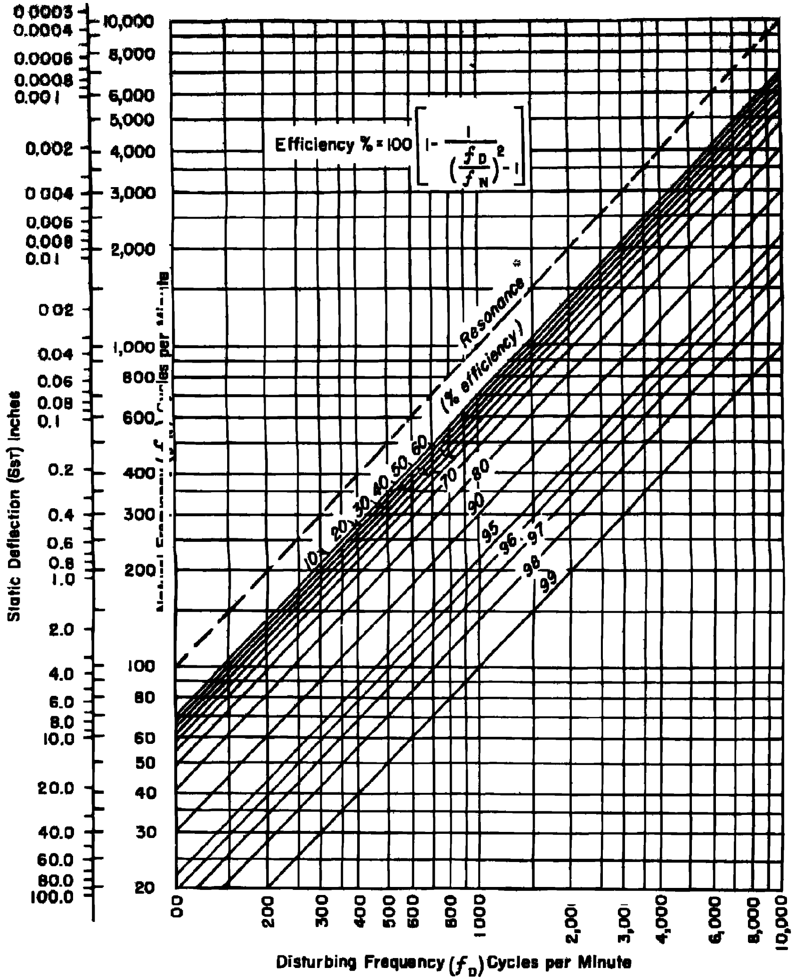


FIG. 31-2. Vibration isolation efficiencies for various natural and disturbing frequencies in undamped systems.

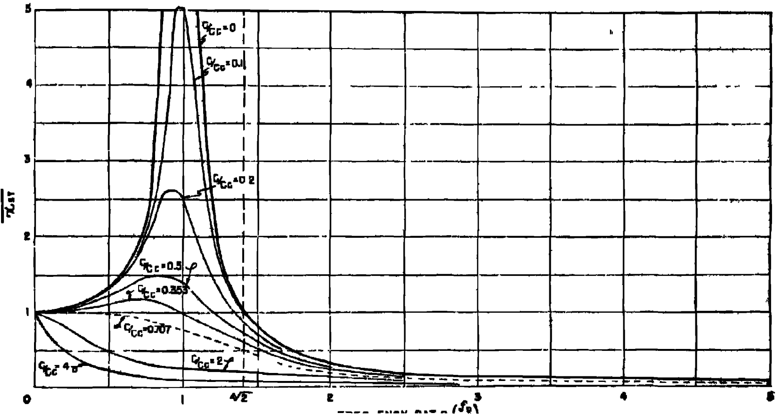
With the disturbing frequency of vibration and the natural frequency of the isolation material known, the expected isolation efficiency can be obtained directly from the curve of Fig. 31-2 or from the equation

$$\text{Efficiency, per cent} = 100\left[1 - \frac{1}{(f_D/f_N)^2 - 1}\right]$$

EXAMPLE: A machine operates with a disturbing frequency of 564 cycles per min, the vibrations being vertical and harmonic. A spring support is placed under the machine and undergoes a 1-in deflection because of the dead weight of the machine. The natural frequency of the support is therefore 188 cycles per min (from Fig. 31-1). Assuming zero damping in the system when operating continuously at a constant speed, find the efficiency of isolation.

$$\text{Efficiency, per cent} = 100\left[1 - \frac{1}{(564/188)^2 - 1}\right] = 87.5$$

The intersection of the vertical lines of disturbing frequency with the horizontal lines of static deflection or natural frequency of the isolation material gives the per cent of isolation efficiency. The curve is actually a representation of the ratio of the force transmitted to a supporting structure to that of the force created by the machine. The curve holds true for cases of vibration systems with little or no damping and for frequency ratios above $1\frac{1}{2}:1$.



SPEED CONSTANT—SPRING VARIES

FIG. 31-3. Damping effect on relative magnification factor for various frequency ratios.

In all cases where vibration isolation is considered necessary, the primary function of the isolation system is to reduce the forces transmitted to a minimum. There are many cases, however, where, in addition to reducing the forces, it is also of great importance for a machine to operate with a minimum amount of movement or relatively small amplitude. This is especially true for cases such as compressors, pumps, engines, and similar installations where piping-system connections leading from the isolated equipment are involved. Continuous vibrations of large amplitude would, of course, be detrimental to the piping system and would result in repeated piping failures. Accordingly, flexible connections should be employed in the pipe lines whenever possible. Through the use of proper isolation and sufficient inertia mass the amplitude of the system can be made small and the possibility of piping failures greatly reduced.

Constant-speed Machines. The curves of Fig. 31-3 will illustrate the effect of damping on the relative amplitude of vibration for an isolated system where the operating speed of the machine remains constant while the natural frequency of the spring is varied. The heavy line of the curve represents a system in which there is no damping ($C/C_c = 0$) and is derived from the equation for relative amplitude:

$$\frac{X}{x_{st}} \quad \frac{1}{(f_D/f_N)^2 - 1}$$

The amplitude of the system is

$$X = \frac{F}{k[(f_D/f_N)^2 - 1]} \text{ for } \frac{f_D}{f_N} > 1$$

$$X = x_{st} = F/k \text{ for } \frac{f_D}{f_N} = 1$$

$$X = \frac{F}{k[1 - (f_D/f_N)^2]} \text{ for } \frac{f_D}{f_N} < 1$$

This undamped curve is seen to start out at a point where the frequency ratio equals zero and the amplitude of the system is equal to the spring deflection due to the static action of the shaking force. As the frequency ratio approaches one, the relative amplitude climbs rapidly until a point is reached where the disturbing frequency is equal to the natural frequency of the system, and resonance occurs. If the system were allowed to operate at this point, the resilient support would compress solid or extend beyond its capacity, and the system would soon break down. As the frequency ratio increases above one, the curve is seen to drop rapidly until, at a point where the ratio is equal to $\sqrt{2}$, the relative amplitude again becomes equal to unity. Further increase of the frequency ratio causes a gradual decline and reduction in the relative amplitude, approaching zero as the frequency approaches infinity. The relative amplitude obtained at a frequency ratio of 2:1 and higher is generally considered satisfactory for most types of industrial and power-plant equipment.

The effect of viscous damping on the system is illustrated by the curves falling under the undamped curve. It can be seen that increasing the damping factor results in a decrease in the relative amplitude for small frequency ratios and is especially effective at or near the point of resonance. As the frequency ratio increases above resonance, however, the effect of damping in the reduction of relative amplitude is greatly diminished, and at frequency ratios of 2:1 and above, the damping curves are very close to the undamped curve, indicating that damping beyond this point is practically ineffective in reducing relative amplitude.

The amplitude of the damped system is given by the equation

$$X = \frac{F}{k\sqrt{[(f_D/f_N)^2 - 1]^2 + 4(C/C_c)^2(f_D/f_N)^2}}$$

The curves of Fig. 31-4 are based on the same conditions as those of Fig. 31-3, except that here the relationship is between frequency ratio and the ratio of the force transmitted to that of the shaking force. This force ratio is generally referred to as transmissibility.

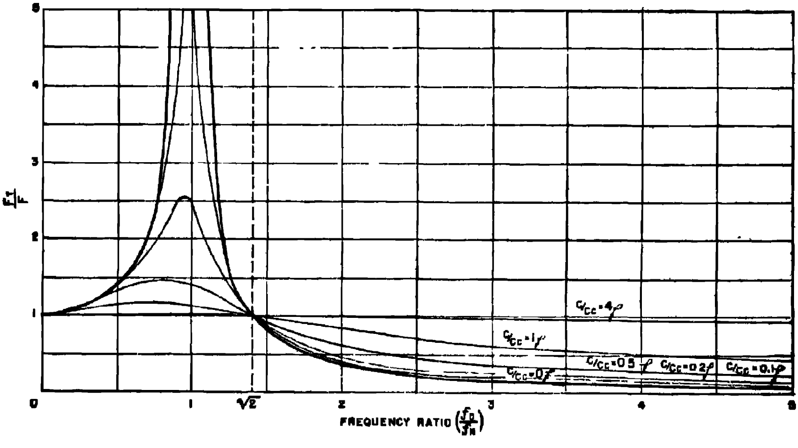
Here again the heavy curve indicates a system in which there is no damping. The transmissibility is seen to start out at unity where the frequency ratio is zero, corresponding to a machine mounted on a solid foundation. A transmissibility of one at this point indicates that all the unbalanced force created by the equipment is transmitted undiminished to the supporting structure. As the frequency ratio increases, the curve rises rapidly until, at the point of resonance, frequency ratio = 1, the force transmitted would be infinitely great, and allowing a machine to operate at this point would lead to its destruction. A further increase in the frequency ratio causes a rapid decrease in the transmissibility until again a point is reached where the force transmitted is equal to the shaking force at a frequency ratio equal to $\sqrt{2}$. Beyond this point, the curve gradually drops, approaching zero at a point where the frequency ratio approaches infinity. Frequency ratios of 2:1 and higher are generally considered satisfactory from the standpoint of reduction in the forces transmitted to the supporting structure. This ratio would correspond to approximately 66⅔ per cent reduction in force. The actual amount of reduction required will, of course, depend upon the construction of the supporting structure and the amount of vibrating force that it is able to sustain. Slightly higher frequency ratios will give an even greater reduction in force such as a ratio of 3:1, corresponding to a force reduction of 87½ per cent.

This undamped force ratio can be calculated from the equation

$$\frac{F_T}{F} = \frac{1}{\left(\frac{f_D}{f_N}\right)^2 - 1} \text{ for } \frac{f_D}{f_N} > 1$$

$$\frac{F_T}{F} = \frac{1}{1 - \left(\frac{f_D}{f_N}\right)^2} \text{ for } \frac{f_D}{f_N} < 1$$

The light lines falling under the undamped curve for frequency ratios below the $\sqrt{2}$ and above the undamped curve for ratios above the $\sqrt{2}$ indicate that viscous damping is effective only in reducing transmitted forces for the lower ratios, while its presence actually increases the forces transmitted above those in an undamped system for higher frequency ratios.



SPEED CONSTANT — SPRING VARIES

FIG. 31-4. Damping effect on relative transmissibility for various frequency ratios.

The transmissibility for the damped system is

$$\frac{F_T}{F} = \sqrt{\frac{1 + 4\left(\frac{C}{C_c}\right)^2\left(\frac{f_D}{f_N}\right)^2}{\left[1 - \left(\frac{f_D}{f_N}\right)^2\right]^2 + 4\left(\frac{C}{C_c}\right)^2\left(\frac{f_D}{f_N}\right)^2}}$$

Since the major portion of industrial and power-plant machinery is designed for constant-speed operation or sometimes two-speed operation, an elastic system of mounting will, when properly designed and selected for the particular installation, provide a highly efficient method of mounting both from the standpoint of reduction in amplitude and reduction in transmitted forces. A small amount of damping is beneficial, however, especially for machinery which is slow in reaching its operating speed. While approaching this speed, the system will have to pass through resonance, and if the increase in speed is slow, there is the possibility of creating excessive amplitudes and transmitted forces in the region of resonance. A small amount of damping will limit these undesirable conditions and yet will have no noticeable effect on the system under normal operation.

Example: A machine having a total weight of 4,000 lb is mounted directly on a set of four spring units located at the corners of the base. The spring constant of the combined springs is 29,400 lb per in. A viscous damping device is placed between the machine and the support and has a damping factor of 0.2. The system is vibrated by a vertical reciprocating part in the machine which weighs 100 lb and oscillates harmonically at a frequency of 1,000 cycles/min through a 4-in. stroke. Find the amplitude of the system and the force transmitted to the supporting structure.

The natural frequency of the system is

$$f_{N(\text{damped})} = \sqrt{1 - (C/C_c)^2}\, f_N = \sqrt{1 - (0.2)^2}\, f_N = 0.982\, f_N$$

and the natural frequency of the springs without damping is

$$f_N = 188\sqrt{k/W} = 188\sqrt{29{,}400/4{,}000} = 510 \quad \text{cycles/min}$$

then

$$f_{N(\text{damped})} = 0.982 \times 510 = 500 \quad \text{cycles/min}$$

The force created by the reciprocating weight is

$$F = \frac{w}{g}\omega^2 e = \frac{w}{g}(2\pi f)^2 e = \frac{100}{386}\left(\frac{2\pi 1{,}000}{60}\right)^2 \times 2 = 5{,}680 \text{ lb}$$

and the amplitude of the system is

$$X = \frac{F}{k\sqrt{\left[\left(\frac{f_D}{f_N}\right)^2 - 1\right]^2 + 4\left(\frac{C}{C_c}\right)^2\left(\frac{f_D}{f_N}\right)^2}} = \frac{5{,}680}{29{,}400\sqrt{\left[\left(\frac{1{,}000}{500}\right)^2 - 1\right]^2 + 4(0.2)^2\left(\frac{1{,}000}{500}\right)^2}} = 0.0645''$$

The force transmitted is

$$F_T = F\sqrt{\frac{1 + 4\left(\frac{C}{C_c}\right)^2\left(\frac{f_D}{f_N}\right)^2}{\left[\left(\frac{f_D}{f_N}\right)^2 - 1\right]^2 + 4\left(\frac{C}{C_c}\right)^2\left(\frac{f_D}{f_N}\right)^2}}$$

$$F_T = 5{,}680\sqrt{\frac{1 + 4(0.2)^2\left(\frac{1{,}000}{500}\right)^2}{\left[\left(\frac{1{,}000}{500}\right)^2 - 1\right]^2 + 4(0.2)^2\left(\frac{1{,}000}{500}\right)^2}} = 2{,}350 \text{ lb}$$

Variable-speed Machinery. There are other types of machinery less frequently encountered in industrial and power-plant installations which normally operate over a wide range of variable speed. Figures 31-5 and 31-6 show the relationship of absolute amplitude and transmissibility to frequency ratio for such installations where the speed or disturbing frequency varies while the frequency of the supporting system remains constant.

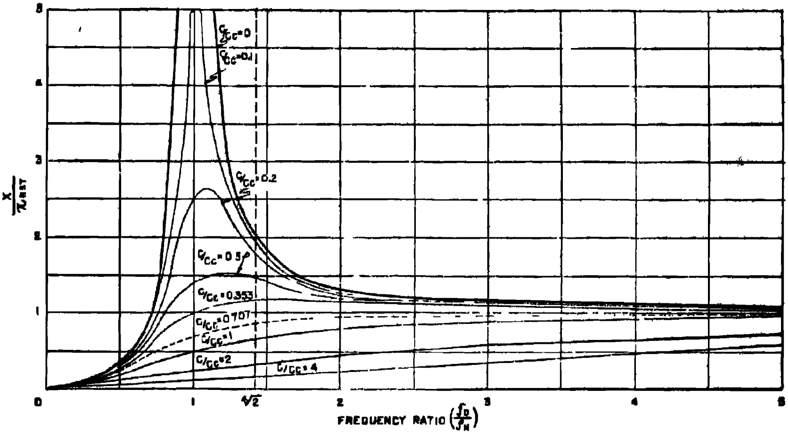
In Fig. 31-5 it is seen that all curves, regardless of the damping factor, start out at zero absolute amplitude and zero frequency ratio, indicating that, where there is no speed, there will, of course, be no amplitude. As the speed or forced frequency increases, the undamped curve rises rapidly, and when the frequency ratio reaches unity, resonance occurs. Increasing the speed of the machine above this point results in a reduction in amplitude in a manner similar to that shown in Fig. 31-3, except that here the curve is seen to approach unity as the disturbing frequency approaches infinity. Instead of indicating relative amplitudes, as in Fig. 31-3, the curve shows the relationship of absolute amplitudes. For any given frequency ratio occurring below the point of resonance, the absolute amplitude of the undamped curve of Fig. 31-5 will be the relative value given for Fig. 31-3 minus 1. For frequency ratios above 1, the values given in Fig. 31-3 plus 1 will correspond to the absolute value of Fig. 31-5.

This curve for zero damping follows the relationship

$$\frac{X}{x_{st}} = \frac{1}{[1/(f_D/f_N)^2] - 1}$$

and the amplitude becomes $X = \dfrac{F_N}{k\left[\dfrac{1}{(f_D/f_N)^2 - 1}\right]}$

where $F_N = (w/g)\omega_N{}^2e$



SPRING CONSTANT— SPEED VARIES

Fig. 31-5 Damping effect on absolute magnification factor for various frequency ratios.

By introducing damping into the system the equation becomes

$$\frac{X}{x_{st}} = \frac{1}{\sqrt{\left[\dfrac{1}{\left(\dfrac{f_D}{f_N}\right)^2} - 1\right]^2 + 4\dfrac{\left(\dfrac{C}{C_c}\right)^2}{\left(\dfrac{f_D}{f_N}\right)^2}}}$$

$$X = \frac{F_N}{k\sqrt{\left[\dfrac{1}{\left(\dfrac{f_D}{f_N}\right)^2} - 1\right] + 4\dfrac{\left(\dfrac{C}{C_c}\right)^2}{\left(\dfrac{f_D}{f_N}\right)^2}}}$$

Damping is seen to be effective in reducing the absolute magnification of the amplitude at all frequency ratios and might, therefore, be considered beneficial from this standpoint alone. It does, however, have a very undesirable effect on the transmitted forces as shown in Fig 31-6.

In Fig 31-6, the undamped curve starts at zero, absolute transmissibility when the machine is at rest. As the speed of the machine increases, the curve rises to infinity at a point of resonance, drops rapidly to a frequency ratio equal to the $\sqrt{2}$, and then gradually reduces to unity as the speed approaches infinity. A high degree of damping is seen to be beneficial for all values of frequency ratios below $\sqrt{2}$ since it greatly reduces the transmitted forces in this region as compared with small amounts of damping or no damping. For speeds resulting in frequency ratios greater than the $\sqrt{2}$, it is seen that damping in any amount is detrimental to the system since it results in a considerable increase in the transmitted force.

The equation for the undamped system is

$$\frac{F_T}{F_N} = \frac{1}{[1/(f_D/f_N)^2] - 1}$$

and with viscous damping

$$\frac{F_T}{F_N} = \sqrt{\frac{1 + 4\left(\frac{C}{C_c}\right)^2\left(\frac{f_D}{f_N}\right)^2}{\left[\frac{1}{\left(\frac{f_D}{f_N}\right)^2} - 1\right]^2 + 4\frac{\left(\frac{C}{C_c}\right)^2}{\left(\frac{f_D}{f_N}\right)^2}}}$$

Practical Application. *Reciprocating Engines and Machines.* Reciprocating engines, pumps, and compressors in the single- or multicylinder types are used extensively throughout the power and industrial field. The extent of vibration-control precautions to be used in the mounting of such equipment depends to a large extent upon the unbalanced forces which the machine will create. It is possible in many cases to balance the machine by the addition of counterweights to the rotating parts and thus arrive at an acceptable solution to the vibration problems. Nevertheless, since a machine can never be perfectly balanced, forces causing vibrations will always exist.

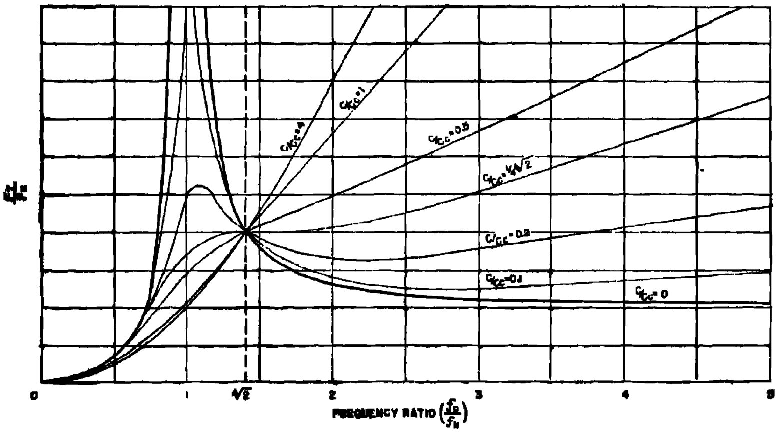


Fig. 31-6. Damping effect on absolute transmissibility for various frequency ratios.

Table 31-1 will be found useful, at least for the purpose of comparison in judging the type of unbalance to be expected from reciprocating machines of various number of cylinders and crank arrangements.

The forces and couples usually considered are primary, occurring once per revolution, and secondary, occurring twice per revolution, and are caused by the inertia of the rotating and reciprocating weights. The forces and moments are calculated from the weight of the reciprocating parts w_1, the weight of the rotating parts w_2, the speed of the machine ω, the crank radius e, the length of connecting rod l and the center-line distance between cylinders d. The primary forces acting along the axis of the cylinder

Table 31-1. Type of Unbalance from Reciprocating Machines

No. of cylinders	Crank arrangement	Primary unbalanced forces		Secondary unbalanced forces along ℄ of cylinder	Primary moments of unbalanced forces		Secondary moments of forces along ℄ of cylinder
		Along ℄ of cylinder	Across ℄ of cylinder		Along ℄ of cylinder	Across ℄ of cylinder	
1	. . . . . .	F_{p1}	F_{p2}	F_{s1}			
2	0–180	O	O	$2F_{s1}$	$F_{p1} \times d$	$F_{p2} \times d$	
3	0–120–240	O	O	O	$\sqrt{3}\,F_{p1} \times d$	$\sqrt{3}\,F_{p2} \times d$	$\sqrt{3}\,F_{s1} \times d$
4	0–180–180–0	O	O	$4F_{s1}$	O	O	
4	0–90–270–180	O	O	O	$\sqrt{10}\,F_{p1} \times d$	$\sqrt{10}\,F_{p2} \times d$	O
5	0–72–144 216–288	O	O	O	U	U	U
6	0–120–240 240–120–0	O	O	O	O	O	O
7	Equal intervals of 360°	O	O	O	U	U	U
8	0–180–90–270 270–90–180–0	O	O	O	O	O	O

NOTE: O = forces cancel out, U = moment unbalanced.

(F_{p1}) for a single-cylinder engine are given by the equation

$$F_{p1} = [(w_1 + w_2)\omega^2 e]/g$$

and the secondary forces (F_{s1})

$$F_{s1} = w_1\omega^2 e/g$$

The primary forces (F_{p2}) acting perpendicular to the axis of the cylinder are

$$F_{p2} = w_2\omega^2 e/gd$$

and the secondary forces

$$F_{s2} = 0$$

The cases given in Table 31-1 are representative of some of the more frequent cylinder arrangements, and it can be seen that, for all engines shown, except the six- and eight-cylinder units, there will exist unbalanced forces or couples. The couples are the result of unbalanced forces acting simultaneously in opposite directions and not along the same line.

Effective vibration control has been provided for all sizes of reciprocating engines from small fractional-horsepower single-cylinder units to the very large multicylinder diesel generator plants. The six- and eight-cylinder units can generally be mounted directly on isolators, although a concrete foundation is usually inserted between the engine and the isolation. The operating speed must be taken into consideration in all cases, and particularly for engines having unbalanced characteristics. Large diesel electric plants operating at speeds as low as 250 rpm have been isolated by means of coil springs and a concrete foundation to such a high degree of efficiency that their operation could not be felt or picked up by sensitive testing equipment anywhere in the vicinity of the installation.

The use of the concrete foundation as an inertia mass is not a necessity from the standpoint of reducing the transmission of unbalanced forces to the substructure, but it is very advantageous in reducing the amplitude of vibration. An effective installation must reduce the transmission of vibration to a satisfactory minimum and at the same time provide stability.

Figure 31-7 illustrates a foundation design widely used for the isolation of large engines, pumps, compressors, and other types of reciprocating machines.

Figure 31-8 is an illustration of isolation applied directly under the base of a small reciprocating compressor. The unit is supported on coil springs within the isolator housing, and lateral motion as well as vertical oscillation is limited through the use of adjustable chocks also located within the isolator housing.

The combination of coil-spring mountings and concrete foundation provides high isolation efficiencies and extremely small amplitudes. A mechanical dry-friction type of damping device is often used in conjunction with the spring mountings to limit excessive motion when the equipment is started up and passes through resonance or when loads are suddenly changed. The usual capacity range for this type of isolation mounting is from 5,000 to 35,000 lb.

Cork is also widely used for this same type of equipment and provides very effective isolation if properly applied. At a loading of about 2,000 psf, which is generally considered a safe and efficient stress for the more widely used materials, cork will deflect approximately 5 per cent of its thickness. It is evident, therefore, that for slow-speed unbalanced machines where large deflections are required for high efficiencies, excessive thicknesses of cork may be required. In such cases a compact spring unit will be found more practical and less expensive.

Rotating Equipment. Machinery designed for rotational operation is generally very well balanced statically and dynamically. This is especially true of high-speed rotational machines. Included under this type of equipment are centrifugal compressors, pumps, turbines, blowers, exhausters, and electric motors. Other types of rotational equipment which are well balanced initially but become unbalanced in use through uneven distribution of the material being handled or an uneven deposit of scale or other solids are rotating mills, pulverizers, crushers, hoggers, extractors, centrifuges, washers, and similar equipment.

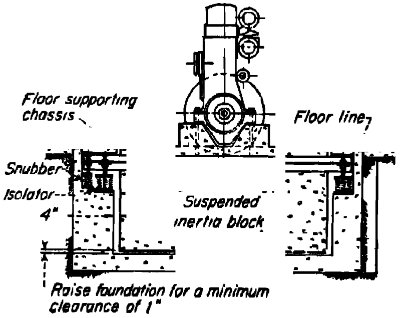


FIG. 31-7. Spring-isolated concrete foundation used for mounting large engines, compressors, and similar equipment. (*Courtesy Korfund Co., Inc.*)

Vibration will be created by all such equipment regardless of how well it is balanced. The high-speed machines, which are generally extremely well balanced for the protection of the machine parts themselves, do not usually cause a severe mechanical disturbance, and the vibrations transmitted are usually in the sound range. These high-frequency vibrations can nevertheless be very troublesome since they may be transmitted over a considerable distance to neighboring plants and dwellings. A remote structure or any of its parts, if in resonance with the vibration, may be forced to vibrate excessively while there may be no apparent vibration in the immediate vicinity of the machine.

Isolation applied at the source can prevent vibration from traveling any further. There have been many occurrences where a so-called perfectly balanced machine has vibrated a building so violently that its operation was prohibited simply because the supporting structure, even though very heavily constructed, happened to have a natural frequency equal to or close to the disturbing frequency of the machine. Isolation in one form or another will usually provide an economical assurance of vibration control. The type of isolation material and method of application are dependent upon the characteristics of the machine and the conditions under which it is to be installed.

Impact Machinery. This type of equipment has not been covered in the theoretical application of this article since it does not involve vibration in the same sense as

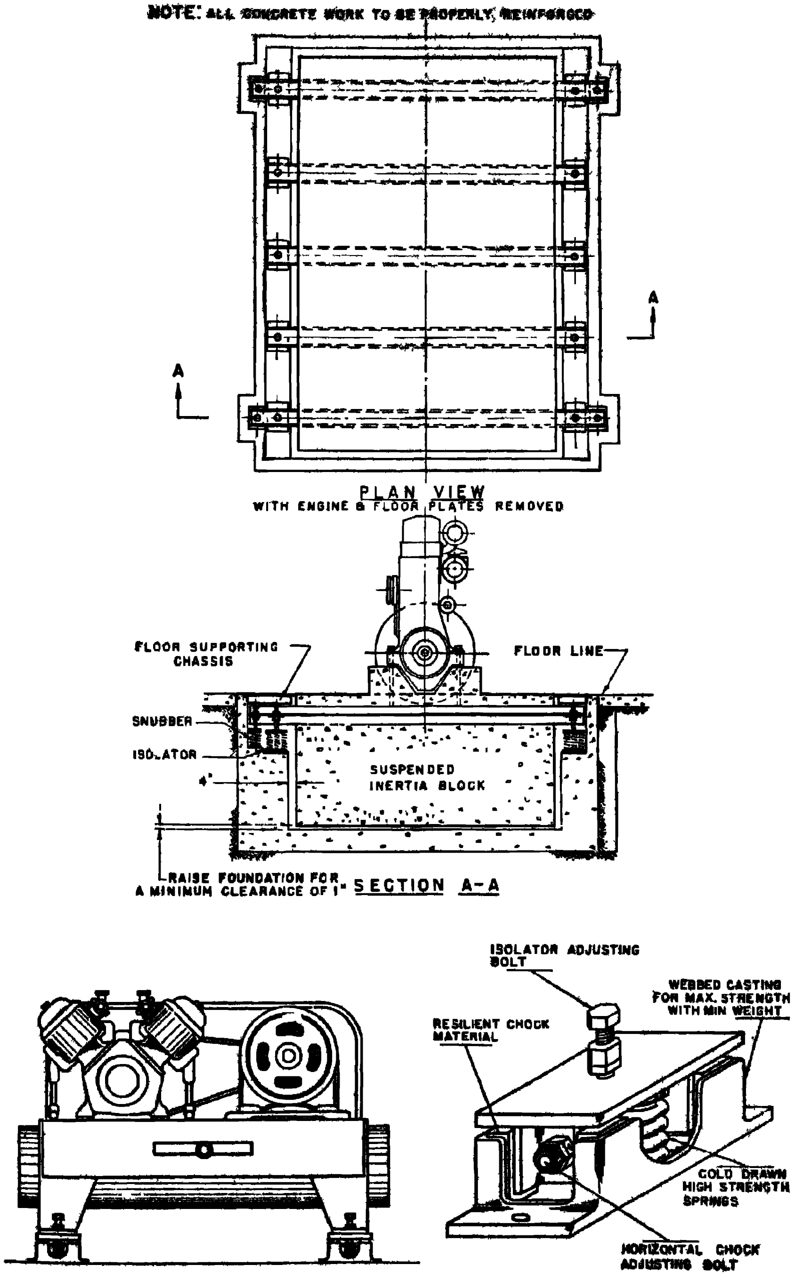

NOTE: ALL CONCRETE WORK TO BE PROPERLY REINFORCED
A
A
PLAN VIEW
WITH ENGINE & FLOOR PLATES REMOVED
FLOOR SUPPORTING CHASSIS
FLOOR LINE
SNUBBER
ISOLATOR
4"
SUSPENDED INERTIA BLOCK
RAISE FOUNDATION FOR A MINIMUM CLEARANCE OF 1"
SECTION A-A
ISOLATOR ADJUSTING BOLT
WEBBED CASTING FOR MAX. STRENGTH WITH MIN WEIGHT
RESILIENT CHOCK MATERIAL
COLD DRAWN HIGH STRENGTH SPRINGS
HORIZONTAL CHOCK ADJUSTING BOLT

FIG. 31-8. Compressor unit mounted on spring isolators. (*Courtesy Korfund Co., Inc.*)

does rotating and reciprocating machinery. Machines operating primarily on the principle of impact, however, are to be found in many types of plant and industry and are often a cause of serious disturbance.

In the case of heavy impact machinery, the determination of the type of resilient mounting, size of foundation, and other factors requires individual study and thorough analysis because of the many variable factors involved. Complex equations have been derived and tested whereby the action of such an installation can be accurately predetermined. In general, a most efficient impact-machine installation is made by suspending the machine and its concrete foundation on spring isolators as illustrated in Fig. 31-9. In this type of suspension a sufficient amount of concrete is used to act as an inertia mass which will limit vertical motion of the block under maximum impact to a predetermined amount considered satisfactory for the operation of the particular machine involved. The spring mountings then absorb shock, changing its form to that of a more slowly applied force to the outside structure. In many cases where actual tests have been conducted with large forging hammers in operation, it is impossible to obtain any indication of shock or vibration transmission from the hammer to the adjacent pit wall or surrounding structure.

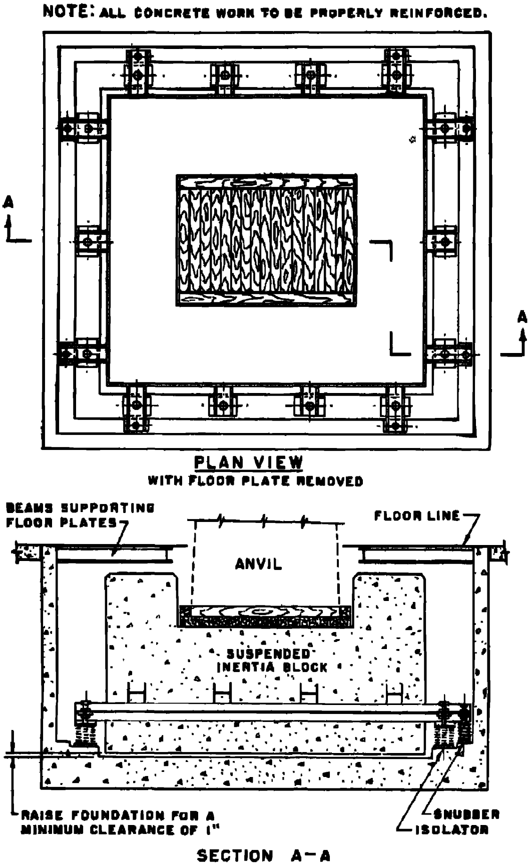
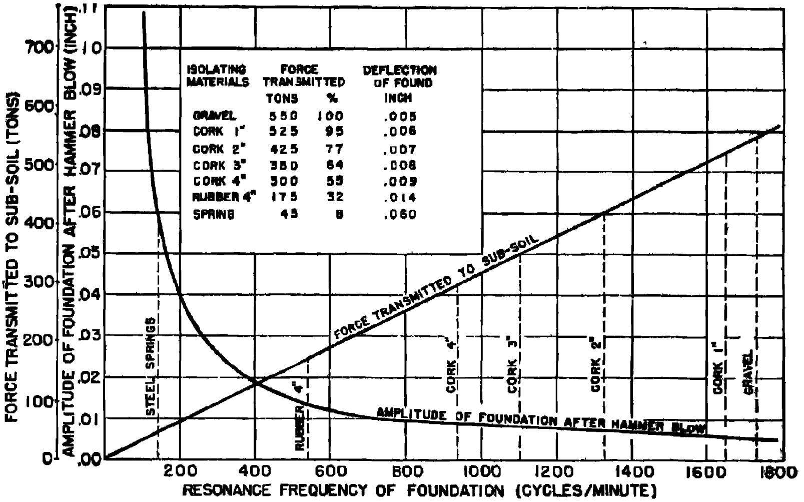

NOTE: ALL CONCRETE WORK TO BE PROPERLY REINFORCED.
A
A
PLAN VIEW
WITH FLOOR PLATE REMOVED
BEAMS SUPPORTING FLOOR PLATES
FLOOR LINE
ANVIL
SUSPENDED INERTIA BLOCK
RAISE FOUNDATION FOR A MINIMUM CLEARANCE OF 1"
SNUBBER
ISOLATOR
SECTION A-A

Fig. 31-9. Typical spring-isolated hammer foundation. (*Courtesy Korfund Co., Inc.*)

An advantage of the adjustable-spring-type method of mounting is the ability to level the hammer at any time in case of possible uneven settlement of the outer concrete pit, which is impossible if an unisolated hammer foundation settles. This adjustment can be made at any time without taking the hammer out of operation simply by making a few extra turns on the isolator adjustment bolts.

Figure 31-10 illustrates the effect of various isolating materials on the amplitude of the foundation and the force transmitted to the substructure for a particular hammer installation. If the hammer were mounted on a very rigid support, practically all the hammer's energy except that portion used in deforming the work would be transmitted, causing severe shock disturbances. Hammer anvils are generally mounted with several layers of timber separating the concrete foundation from the anvil. The timbers will deflect slightly under impact, resulting in a slight reduction of shock transmission, but usually the reduction is not sufficient to prevent the shock transmission to other parts of the building. Soil conditions have a very important influence on shock transmission from an unisolated hammer.



ISOLATING MATERIALS	FORCE TRANSMITTED TONS	FORCE TRANSMITTED %	DEFLECTION OF FOUND INCH
GRAVEL	550	100	.005
CORK 1"	525	95	.006
CORK 2"	425	77	.007
CORK 3"	350	64	.008
CORK 4"	300	55	.009
RUBBER 4"	175	32	.014
SPRING	45	8	.060

FIG. 31-10. Efficiency of various isolating materials based on hammer with 65,000 ft-lb blow energy. Total weight of hammer and foundation, 715 tons. (*Courtesy Korfund Co., Inc.*)

It can be seen from Fig. 31-10 that, for increasing thickness of sheet-isolation materials placed under the foundation, a small reduction of force transmitted to the subsoil results, and where steel springs are used, a very great reduction is obtained. The motion of the foundation after impact is largest for the spring installation because of its greater resiliency, but this can be reduced to any desired amount by the addition of weight to the system without necessarily increasing transmission. It might appear as though an isolated hammer installation would reduce the efficiency of the machine because of the movement of the foundation after impact. This, however, is not the case since the timbers usually incorporated with an unisolated hammer are practically eliminated except for a thin protective layer between the anvil and the concrete. Thus the foundation becomes a part of the anvil, offering even greater reaction to hammer ram.

Certain types of presses require isolation in the same manner as described above for hammers. In general, however, a press usually has sufficient inertia mass in its own frame so that a concrete foundation is not required to obtain operating stability and efficient shock control.

Impact machinery used in the metal-forming industry, such as punch presses, shears, and brakes, can be efficiently isolated even on an upper story of a building by use of properly designed spring-type of mountings. In some cases where the height of the machine is large in comparison to the area of its base, an extended structural-steel or other type of base covering a greater floor area than occupied by the machine itself may be required in order to provide the necessary machine stability as well as to spread the weight over a sufficient floor area to fall within allowable static-load limitations.

Machine Tools. Machine tools can either be the cause of vibration disturbance, or they may have to be protected against external vibration. The effect of vibration on the latter type of machine tools, such as grinders, jig borers, etc., can result in costly work spoilage and damage to the machine. In most cases there are many sources of vibration disturbance and shock with a resulting wide variation in disturbing frequency. It is therefore usually necessary to provide an isolation material with a large static deflection to isolate against low steady-state vibration as well as intermittent shock. Owing to the large deflection requirements of the isolation material, it is usually necessary to incorporate a spring-type mounting to obtain a practical design.

Machine tools which are often a source of vibration, such as broaching machines, milling machines, shapers, and planers, can be located close to precision machines and sensitive laboratory equipment without fear of vibration disturbance if properly isolated. Cork and rubber mountings have been used under this type of equipment with considerable success, but here also, as in practically all cases, the type of structure upon which the equipment is mounted is an important factor. A vibration-creating machine tool mounted on a 2- or 3-in. thickness of cork located in a sturdy reinforced-concrete building can usually be expected to operate without the slightest vibration disturbance, while the same machine mounted in the same manner in a wood-frame-constructed building might be the cause of a great deal of vibration disturbance. In the case of the latter installation a comparatively soft spring mounting or a combination spring-and-rubber mounting may be required for high isolating efficiency.

Laboratory Equipment. It is often found advantageous to locate sensitive testing equipment and precision measuring devices in a locality where vibration disturbance from an outside source is to be expected. The setup may be of a temporary nature or a portable unit, and it would therefore be impractical to attempt to eliminate the source of vibration. On the other hand, the installation may be of a permanent nature, located in a plant having many sources of vibration, or even a single source of vibration, where the expense of isolating the disturbance at its source may be unwarranted.

In these cases it is desirable to isolate the sensitive equipment against the possibility of incoming vibration. Owing to the many variables involved in an installation of this type, it is almost always necessary to design a special type of mounting to suit the equipment under consideration. Generally speaking, it is usually necessary to provide considerable spring deflection in order to isolate against many vibration frequencies as well as shocks.

NOISE SILENCING

There is little need to emphasize the growing importance of sound insulation and acoustical treatment to reduce the noise level in buildings. The need for a quiet atmosphere in hospitals, schools, libraries, and other public buildings has been long felt, and special precautions are generally taken to eliminate noise from these structures. More recently noise elimination has been carried into offices because tests have shown that subjection to noise decreases workers' efficiency and promotes fatigue. For instance, a metabolism experiment made in an eastern university showed that typists used 19 per cent less energy and gained slightly more than 4 per cent in typing speed when the noise intensity was reduced 50 per cent. Similar tests in industry indicate that noisy factory conditions promote fatigue, decrease worker efficiency, and lead to accidents as well as physical and mental sickness.

General Theory. Sound results from stimulation of the auditory nerves by wave energy set in motion by a vibrating body and transmitted through an elastic medium (usually air) to the ear.

The two most important characteristics of a sound are its frequency and intensity.

Frequency, the simpler of the two, is the number of vibrations per unit of time experienced by the sound source. This controls the "pitch" of the sound. For example, a "middle-C" musical tuning fork vibrates at a frequency of 264 cps, and one octave higher corresponds to a vibration of 528 cps. The audible frequency range is from about 20 to 20,000 cps. The approximate frequency range for the male voice is from 80 to 465 cps, while the female range is from 195 to 1,045.

Frequency has a great effect upon sound intensity and is also an important factor in locating the source of disturbing noises.

Sound intensity is defined as the energy transmitted per unit of time through a unit area taken perpendicular to the path of the sound wave. The amplitude of vibration of any membrane (including the eardrum) upon which a sound wave impinges is proportional to the sound intensity, which, in turn, is proportional to the square of the product of the amplitude and frequency of the vibration.

Sound intensity also varies as the square of the pressure created by the sound wave, and this principle is widely used in sound-intensity meters. The ear is sensitive to a minimum root-mean-square pressure variation of 0.001 dynes per sq cm, assigned a relative value of 1. The maximum pressure variation the ear can endure without pain is 1,000 dynes per sq cm or a relative value of $1{,}000/0.0001 = 10^7$. The unit of proportional intensity in the first case would then be $1^2 = 1$, and the proportional intensity in the second would be $(10^7)^2 = 10^{14}$.

Since an individual's sensitivity to sound is roughly proportional to the logarithm of the intensity, a system was devised where intensity is measured in units equal to ten times the logarithm of the proportional intensity as shown in Table 31-2. The unit is called the decibel.

Table 31-2

Pressure variation, dynes/sq cm rms pressure	Relative pressure	Proportional intensity	Intensity, db
0.0001	1	$(1)^2 = 1$	$10 \log_{10} 1 = 0$
0.001	10	$(10)^2 = 10^2$	$10 \log_{10} 10^2 = 20$
0.01	10^2	$(10^2)^2 = 10^4$	$10 \log_{10} 10^4 = 40$
0.1	10^3	10^6	60
1.0	10^4	10^8	80
10.0	10^5	10^{10}	100
100.0	10^6	10^{12}	120
1000.0	10^7	10^{14}	140

However, as indicated in Table 31-2, the physical reaction to sound is only roughly proportional to the intensity in decibels. As the intensity increases, the ear becomes less sensitive, and sounds of lower frequency do not produce so great an audible effect on the eardrum as higher frequency sounds. The "loudness scale" shown in Fig. 31-11 resulted from tests on thousands of people to determine the actual average reaction to sound intensity.

Since sound-measuring instruments give an intensity reading in decibels, for practical purposes the curve shown in Fig. 31-11 is very helpful in converting these readings to loudness units.

Example: Suppose vibration isolation and acoustical treatment in a factory had resulted in a decrease in the loudness intensity (as read by the instrument) from 80 to 70 db. What was the actual physiological per cent reduction in loudness?

Solution: 80 db corresponds to 17,000 units on the loudness scale (Fig. 31-11) and 70 db to 8,000. This then represents a 53 per cent reduction in actual loudness.

Table 31-3 shows the results of tests in buildings to determine the amount of noise present. The "observed" effect of an external noise entering a room is dependent upon the amount of noise normally encountered in a room. For example, the noise effect produced by dropping a book in a library is much greater than when the same

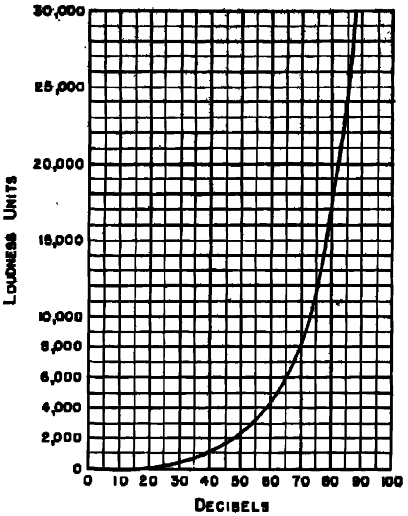


FIG. 31-11. This loudness-unit-decibel curve gives the average physiological reaction of the human ear to different sound intensities.

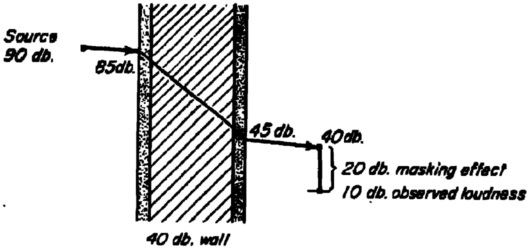


FIG. 31-12. Wall and sound source showing effect of typical sound-intensity losses and masking upon final observed loudness.

Table 31-3. Noise in Buildings*

Location and source	Level above threshold of hearing, db
(Painful noise level)	130
Boiler factory	97
Subway, local station with express train passing	95
Noisy factories	85
Very loud radio in home	80
Stenographic room, large office	70
Average of six factories	68
Information booth, large railway station	57
Noisy office or department store	57
Moderate restaurant clatter	50
Average office	47
Noises measured in residence	45
Very quiet radio in home	40
Quiet office	37
Quietest nonresidential location	33
Average residence	32
Quietest residence measured	22

* From *City Noise*, Department of Health, New York City.

book is dropped in a busy office because the normal office noise "masks" a portion of the additional sound. This is further illustrated in Fig. 31-12.

Sound transmission into and through buildings results from three phenomena:

1. Air-borne sounds transmitted from room to room through doors, windows, cracks, through air-conditioning duct-work, etc.
2. Structural-borne sounds transmitted through building materials in contact with each other
3. "Diaphragm-transmitted" sounds resulting from sound waves impinging upon a wall, door, etc., and inducing a vibration in the material which then sets up sound waves on the other side of the diaphragm

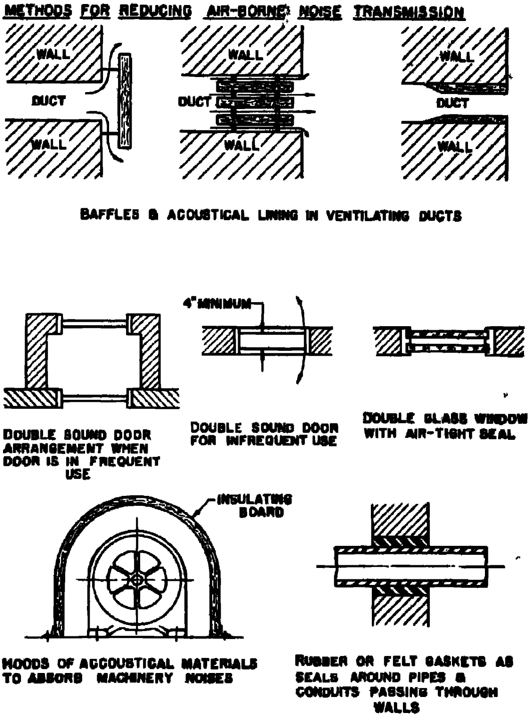

METHODS FOR REDUCING AIR-BORNE NOISE TRANSMISSION
WALL
DUCT
WALL
WALL
DUCT
WALL
WALL
DUCT
WALL
BAFFLES & ACOUSTICAL LINING IN VENTILATING DUCTS
4" MINIMUM
DOUBLE SOUND DOOR ARRANGEMENT WHEN DOOR IS IN FREQUENT USE
DOUBLE SOUND DOOR FOR INFREQUENT USE
DOUBLE GLASS WINDOW WITH AIR-TIGHT SEAL
INSULATING BOARD
HOODS OF ACCOUSTICAL MATERIALS TO ABSORB MACHINERY NOISES
RUBBER OR FELT GASKETS AS SEALS AROUND PIPES & CONDUITS PASSING THROUGH WALLS

FIG. 31-13. Methods for reducing air-borne noise transmission.

Air-borne Noise. Control of air-borne noise is extremely important and is often the limiting factor in a noise-control program. For example, a door or window may transmit more noise than the entire wall, thus making it impractical to improve the wall insulation if nothing can be done about the openings.

The smallest cracks around pipes passing through a wall, poorly fitting doors, and other almost unnoticeable openings may destroy the sound insulation value of a wall. Figure 31-13 shows a few of the many practical solutions which have been devised for preventing air-borne noise transmission. It is sometimes possible to use shields of sound-absorbing material around the noise source. The use of double doors and windows is very effective, and a double sound door can easily be incorporated into an existing structure as shown in Fig. 31-13. Special soundproof windows are now available which permit a flow of air through the window.

Noise transmission through ventilating systems is very common. The noise due to mechanical vibration of the fan and motor is easily isolated from the system by using a short length of flexible duct in the line at the fan outlet. Lining the ducts with absorbent material is very effective in preventing "air-rush" noises, and this can be accomplished without materially increasing the frictional resistance if a thin layer of perforated sheet metal is used on top of the absorber. The most effective single point for lining ventilating ductwork is at the inlet end.

In some factories the use of acoustical materials on the ceiling and walls will be found effective in reducing the noise level within the room. A discussion of the theory of acoustics is beyond the scope of this section, but most manufacturers of acoustical materials are prepared to analyze the noise conditions in a plant and recommend the proper acoustical treatment.

Structural-borne Noise. Structural-borne noise is encountered in all types of buildings, but particularly in modern concrete and steel buildings where contacting members afford an excellent path for sound transmission. It is not uncommon for noise to travel through steel building columns and be extremely objectionable half a dozen floors from the source, without even being noticeable on floors closer to the source. This occurs when a portion of the building structure at some distant point happens to have a natural frequency corresponding to the frequency of the disturbing vibration, which results in great amplification of the induced vibration.

Structural-borne noise and mechanical vibration are practically synonymous since mechanical vibrations transmitted through the structure and activating some portion of the building also have exactly the same effect as described above. Consequently, vibration isolation of offending machines often provides a solution to the problem of structural-borne noise. When noise carried to other parts of the building is known to be a factor, it is usually possible to incorporate organic materials in the vibration isolator to dampen the noise, as discussed elsewhere in this section.

Since structural-borne noise results from sounds passing from one building member to another, it follows logically that breaking up the path of propagation will usually result in improved conditions. The value of vibration isolators is partially due to this factor, and this principle is utilized extensively in "floating" floors and "hanging" ceilings which will be discussed under Floor and Ceiling Structures.

Where one part of the building is to house equipment creating vibration and noise and another part of the structure must be particularly noise free, a complete separation approximately ½ in. wide is sometimes provided between the two parts. Separate building-column footings are used, and where there is a possibility of vibration transmission through the soil between the footings (particularly in wet ground, clay, or rock), it is frequently desirable to provide isolation materials under the columns of the building housing the offending machines.

Diaphragm-transmitted Sounds. Sound transmission of the diaphragm type is very common; every wall, ceiling, floor, door, etc., is a membrane which may be induced to vibrate by sound waves impinging upon one side, thereby transmitting the wave motion to the air on the opposite side of the membrane. The amount of energy transmitted through a membrane depends upon:

1. The energy of the sound wave striking the membrane, since it is this energy which sets the structure vibrating and maintains the vibration.

2. The mass of the membrane, since more energy is expended in vibrating a heavy membrane than a light one.

3. The stiffness of the membrane, because the stiffer the structure, the smaller is its amplitude of vibration and consequently the smaller the amplitude of the sound waves induced on the other side.

4. How the edges of the membrane are held in position, particularly as this affects the damping of the membrane. This consideration is of much more importance on small rather than large membranes.

5. The natural frequency of the membrane, which is of importance only if resonance is encountered—when the frequency of the impinging waves is equal to the natural frequency of the panel, in which case the amplitude of induced vibration may become very great.

Consideration of the above conditions suggests many practical design limitations and practices which can greatly affect the sound insulation value of a structure. These will be discussed further under Wall Structures.

Wall Structures. Sound transmission through walls can be the result of air, structural, or diaphragm noise propagation. The following are a few of the most important fundamental principles (in addition to those previously given) which influence wall design:

1. The sound-insulation value of a homogeneous wall is primarily dependent upon its weight-per-unit area and varies as the logarithm of this factor. For this reason, it is seldom practical to resort only to greater wall thicknesses for improved insulation.

2. For the same weight-per-unit area, a nonhomogeneous membrane consisting of two or more layers is a better sound insulator than a homogeneous membrane. This type of construction offers a great deal of resistance to diaphragm transmission, and the general use of furring strips or other separators between the layers reduces the path of propagation for structural-borne noise (see constructions shown in Fig. 31-14).

3. The empty air space between layers of a nonhomogeneous wall is frequently a better sound insulator than filler materials sometimes used in such spaces, particularly if the wall layers are connected in any way.

4. However, the complexity of a nonhomogeneous structure is not necessarily an accurate index of its efficiency, from either the standpoint of sound transmission or cost, and each design should be checked against existing standard constructions of known insulation value.

5. Wall structures generally absorb greater proportions of high-pitched than low-frequency noise, as shown in Table 31-4. Furthermore, various constructions are better for different frequencies, and it is often possible to choose the type of wall best suited for isolation of particular known noises within a definite frequency range. It should also be noted that the intelligibility of voices depends upon the high-pitched tones. This means that it is frequently possible to eliminate completely the transmission of voice sounds or to reduce them to an unintelligible murmur by absorbing the higher frequency tones.

6. Increasing the insulation value of the walls of a room is effective only against transmission of noises from an external source into the room. Acoustical treatment must be used to reduce the noise level due to sounds originating within the room. On the other hand, the use of both acoustical materials and sound-insulation walls can be very effective in isolating the noise source.

Some of the more common types of wall structures are shown in Fig. 31-14 and described in Table 31-4. Of the very many types tested and reported on in the U.S. Department of Commerce bulletin, "Building Materials and Structures," by V. L. Chrisler, only a few of the best and most practical or most interesting are shown in Fig. 31-14. However, a study of the values for some of these panels is enlightening in view of the principles outlined above.

Comparison of panels 86 and 66 shows the advantage of providing several separated layers in a wall, but it will be noted that in panel 71, where this principle has been combined with the advantages of rigidity and mass provided by a heavier wall, the results are even better.

Two methods of improving the insulation value of basic panel 148 are shown in panels 149 and 150, where the path of propagation is reduced by using large-headed nails driven between the sheets of lath instead of through them and by further breaking up the sound path with spring clips. The same effect can be seen by comparing panel 175 with panels 164 and 165, where an improvement in insulation value is obtained by staggering the studs to prevent solid contact between both sides of the wall.

The value of eliminating rigid connections between the two wall surfaces is shown very well in panel 138, where an increase of approximately 40 per cent in insulation value is obtained over panel 161 by separating the two surfaces with spring clips.

A comparison of panels 63, 64, and 65 seems to indicate that little is sacrificed in sound insulation where it is necessary to use harder burned wall tile in order to support greater loads.

The value of a double-wall construction is clearly shown by comparing panels 159 and 160E, where an increase of approximately 50 per cent results from the use of a double wall. Increasing the distance between the double walls by 122 per cent increases the insulation value approximately 7½ per cent as shown in panel 160A.

The great increase in sound insulation obtainable by "floating" a wall with furring strips or spring clips makes this one of the best methods for improving an existing wall structure. This principle is used extensively where a "room within a room" is constructed to insulate an existing room.

Floor and Ceiling Structures. Sound transmission through floors and ceilings can result from air, diaphragm, or structural-borne noise. The last is the principal source of difficulty, since it is usually possible to correct air-borne noise and because the heavier construction of floors and ceilings reduces considerably the diaphragm transmission.

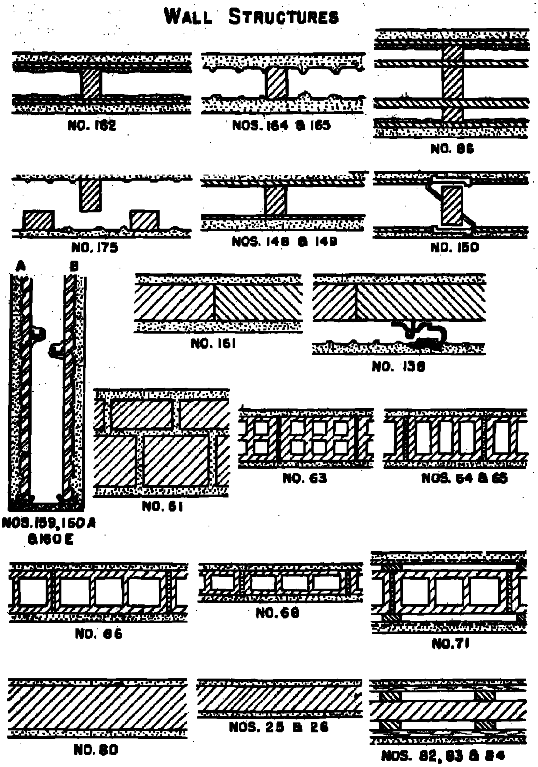


FIG. 31-14. Typical wall structures (see Table 31-4).

In industrial applications, it should be remembered that the noise due to mechanical-equipment vibration is the most readily transmissible of all types of noise and is much worse in the room below the machine than in the room above. Many companies locate their offices on the ground floor of a building and their production equipment on the floors above. The result is far more disturbing noise in the offices than there would be had the offices been located above the machinery. Furthermore, the usually heavier construction of the lower building floors makes them better suited for carrying machinery than the upper floors, which are better suited for offices, laboratories, etc.

The almost universal panacea for noise and vibration transmission through floors and ceilings is the employment of a resilient support under the offending machine.

Table 31-4. Sound-transmission Loss—Wall Structures*

(Values and descriptions refer to Fig. 31-14)

Panel No.	Transmission loss, db, at various frequencies, cps						
	256	512	1,024	2,048	Average		Weight lb/ft^2
					256–1,024	128–4,096	
25	43.1	..	46.7	54.5	44.9		
26	46.4	....	48.8	58.4	47.6		
61	40.4	46.3	48.7	53.3	48.1	..	66.0
63	38.8	42.1	46.6	53.5	42.5	..	39.0
64	41.2	37.4	45.1	52.1	41.2	....	37.0
65	41.1	42.0	43.7	50.1	42.3	....	37.0
66	41.1	40.0	41.5	49.9	40.9	..	29.0
68	40.7	35.9	43.3	51.0	40.0	..	28.0
71	55.6	52.8	57.3	57.6	55.2	...	34.0
80	47.7	49.4	57.0	59.2	51.4	..	97.0
82	52.1	47.4	56.5	53.9	52.0		36.5
83	46.8	44.3	54.4	61.3	48.5	...	38.2
84	48.8	50.5	59.8	55.8	53.0	..	33.3
86	42.4	38.2	44.7	54.1	41.8	....	14.7
138	44.4	54.7	59.1	61.9	52.7		
148	31.0	39.2	45.6	48.7	39.0	41.1	15.2
149	38.9	45.7	50.4	55.1	45.8	47.7	15.7
150	48.1	50.0	56.4	47.6	51.8	51.8	
159	28.8	34.7	32.8	32.2	33.1	33.3	8.1
160A	47.8	53.4	54.6	60.0	53.0	55.2	17.2
160E	43.1	50.6	50.2	61.6	49.4	52.9	17.2
161	35.7	36.2	41.5	46.8	37.8	38.1	21.0
162	36.3	41.0	50.2	54.9	41.8	42.1	15.6
164	40.6	43.7	52.3	56.1	45.0	44.4	19.8
165	34.5	38.2	42.6	44.7	38.1	39.2	20.0
175	47.4	47.4	50.0	52.1	48.5	49.8	19.8

Panel description:

25—4-in. brick panel; plastered both sides with brown coat of lime plaster; smooth, white finish

26—4-in. brick panel, plastered both sides with brown coat of gypsum plaster; smooth finish

61—Hollow clay-tile panel (two units, 3¾ × 5 × 12 in. and 8 × 5 × 12 in.) side construction; plastered both sides with brown coat gypsum plaster; smooth, white finish

63—6 × 6 × 12-in. hollow clay load-bearing partition tile, six cells; plastered on both sides with brown coat gypsum plaster; smooth, white finish

64—Same as No. 63, except medium-burned three-cell tile

65—Same as No. 63 except soft three-cell partition tile

66—4 × 12 × 12 in. hollow clay three-cell partition tile; brown coat gypsum plaster both sides; smooth, white finish

68—Same as No. 66, except 3 × 12 × 12-in tile

71—4 × 12 × 12-in. hollow clay three-cell partition tile; wood furring strips; paper; metal lath: scratch and brown coats gypsum plaster; smooth, white finish

80—8-in. New Hampshire brick panel; brown coat gypsum plaster both sides; smooth, white finish

82—New Hampshire brick laid on edge; furring strips wired; gypsum plaster board; scratch and brown coats gypsum plaster both sides; smooth, white finish

83—Same as No. 82, except furring strips nailed instead of wired

84—New Hampshire brick laid on edge, furring strips nailed; ½-in. Insulite; scratch and brown coats gypsum plaster both sides; smooth, white finish

86—Wood studs; ½-in. Flaxlinum nailed to each side of 1 × 2-in. furring strips; wood lath, scratch and brown coats gypsum plaster both sides; smooth, white finish

138—3 × 12 × 30-in. gypsum tile, U.S. Gypsum resilient clip; metal lath and gypsum plaster on one side; gypsum plaster applied directly to tile on other side; smooth, white finish both sides

148—Wood studs; gypsum lath nailed to studs, nails approximately 6 in. apart; scratch and brown coats sanded gypsum plaster; smooth, white finish; thickness of plaster ½ in.

149—Same as No. 148, except lath held on with special large-head nails driven between sheets of lath

150—Same as No. 148, except lath attached to studs with spring clips instead of nails

159—Panel A only—Metal studs of ¾-in. channel iron expanded metal lath; scratch and brown coats gypsum plaster; smooth, white finish, applied to one side only

160A—Two panels similar to panel No. 159 placed back to back and resting on 1-in.-thick cork; 10-in. distance from face to face of panels

160E—Same as No. 160A, except 4½-in. distance from face to face

161—3 × 12 × 30-in. gypsum tile; brown coat gypsum plaster; smooth, white finish

162—Wood studs; wood lath; scratch and brown coats lime plaster; smooth, white finish

164—Wood studs; metal lath; scratch and brown coats lime plaster; smooth, white finish

165—Same as No. 164, except gypsum plaster used

175—Wood studs staggered; expanded metal lath; scratch and brown coats gypsum plaster; smooth, white finish; thickness of grounds ¾ in.

* From CHRISLER, V. L., *Building Materials and Structures*, National Bureau of Standards.

The most effective way of isolating machines is the use of vibration and sound isolators between the machine and the floor as discussed in another portion of this section. This also has the advantage of isolating each machine against vibration from the other machines. Many companies have all their vibration-creating machines mounted on "floating" bases.

In cases where individual machine isolation is impractical, as for example, when trucks are moving around on the floor and possibly picking up and dropping loads, a "floating" floor is very effective in reducing vibration and noise transmission. In fact, for sound-insulation purposes, a floating floor under the disturbing equipment is much more effective than a hanging ceiling on the room below.

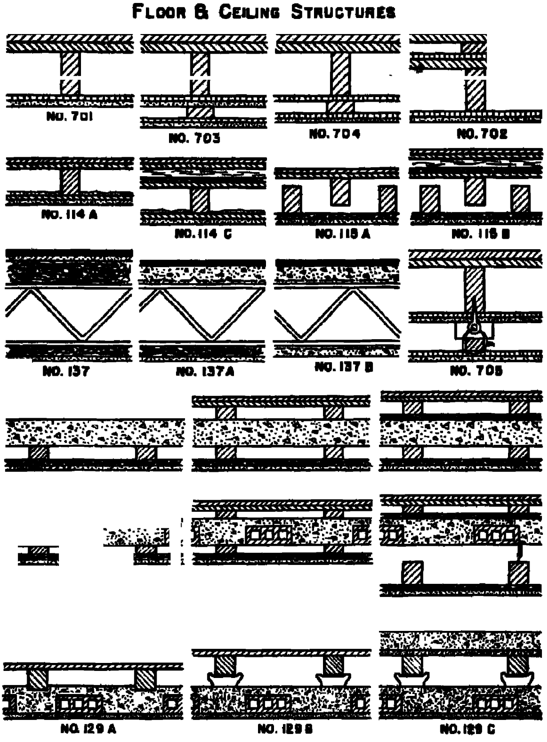


FIG. 31-15. Typical floor and ceiling structures (see Table 31-5).

Figure 31-15 and Table 31-5 illustrate some typical primary floor and ceiling structures and indicate what can be done to improve them by adding secondary floating floors and ceilings.

Considerable caution should be exercised when floating floors are considered for use under impact machinery which may also be supported on vibration mountings. It is very easy to destroy completely the effectiveness of shock mountings by placing them on a floor which is so resilient that it "gives" before the mountings have a chance to deflect and absorb the impact. This will result in vibration and noise transmission into both the floating floor and the main building structure. Generally speaking, if a floating floor is used to absorb the vibration and noise from mobile vibration-creating equipment, the isolation mountings under any stationary impact-creating equipment should be placed on the primary floor, with the secondary floating floor cut out around the machine to assure that the mountings have the most rigid support.

Table 31-5. Sound-transmission Loss—Floor and Ceiling Structures*

(Values and descriptions refer to Fig. 31-15)

Panel No.	Transmission loss, db, at various frequencies, cps							
	256	512	1,024	2,048	Average		Tapping	Weight, lb/sq ft
					256–1,024	128–4,096		
114A	46.8	40.7	50.1	48.5	45.9	....	14.0	
114C	57.5	54.8	62.4	57.6	58.2	....	22.0	
115A	53.6	49.2	54.9	55.3	52.6	....	22.0	12.6
115B	65.3	57.3	68.8	62.3	63.8	....	30.0	16.1
116A	54.8	58.7	56.5	53.2	56.7	....	1.2	54.4
116B	57.0	55.4	67.6	65.2	60.0	....	30.0	58.1
116C	58.2	55.8	66.3	67.3	60.1	....	33.0	58.9
117A	56.6	55.8	57.7	58.8	56.7	....	5.1	69.8
117B	63.1	61.0	65.9	73.7	63.3	....	34.0	73.5
118	67.9	65.8	72.1	< 76.0	68.6	....	51.0	72.8
129A	37.7	38.6	46.8	53.8	41.0	....	22.6	
129B	46.6	58.3	68.5	73.2	57.8	....	33.0	
129C	49.5	60.9	71.3	76.7	60.6	....	38.5	
137	43.7	51.9	58.2	64.5	51.0	52.8	11.7	
137A	46.6	52.0	59.3	65.2	52.3	53.7	13.6	
137B	48.4	53.6	66.0	63.4	55.5	54.9	13.1	
701	33.7	46.7	54.7	54.0	....	45.1	10.8	14.3
702	36.9	49.7	56.8	65.3	....	49.7	12.1	16.2
703	32.4	44.8	47.5	54.3	....	45.4	9.8	19.0
704	37.5	40.1	55.8	58.2	....	47.4	14.2	15.9
705	50.5	54.9	56.4	63.1	....	55.5	25.7	20.3

Panel description:

114A—Floor panel; wood joists; plaster on wood lath on lower side; subfloor and ⅜-in. finish floor upper side

114C—Same as 114A, except flooring; rough flooring; ½-in. Insulite; floating floor, consisting of nailing strips, rough, and finish flooring

115A—Suspended ceiling; wood joists; ½-in. Insulite plaster ceiling; rough floor; finish floor; ends of ceiling and floor joists nailed to common support

115B—Same as 115A, except flooring; rough floor; ½-in. Insulite; floating floor consisting of 1 × 2-in. nailing strips, rough flooring, and ⅜-in. oak finish flooring

116A—Reinforced concrete flat-slab-type floor construction; Insulite furred out and applied as ceiling; brown coat gypsum plaster; smooth, white finish

116B—Same as 116A, except for addition of floor consisting of 1 × 2-in. nailing strips, rough flooring, and ⅜-in. oak finish flooring

116C—Same as 116B, except ½-in. Insulite added between concrete slab and floating floor

117A—Combination floor panel constructed of 4 × 12 × 12-in. three-cell partition tile; ceiling finished with furring strips, ½-in. Insulite, and plaster

117B—Same as 117A, except for addition of floating floor, consisting of 1 × 2-in. nailing strips, rough flooring, and ⅜-in. oak finish flooring

118—Same as 117B, except ½-in. Insulite added between masonary slab and floating floor; ceiling stripped off and suspended ceiling attached by means of wires

129A—Combination floor panel constructed of 4 × 12 × 12-in. three-cell partition tile; ceiling finished with ½-in. brown coat gypsum and smooth white finish coat; floor of 13/16-in. oak flooring, nailed to 2 × 2-in. nailing strips on 16-in. centers grouted into concrete

129B—Same as 129A, except U.S. Gypsum resilient clips used between concrete and nailing strips

129C—Same as 129B, except oak flooring removed, and ½-in. gypsum plasterboard attached to the nailing strips and ½-in. Hydrocal applied on top of the plasterboard

137—Floor panel constructed of 8-in. Mac Mar joist, with 3-in. Thermax clipped on top and 1-in. Thermax clipped on bottom joist; ½-in. concrete poured on top of the 3-in. Thermax; floor finished with ¼-in. battleship linoleum cemented on top of concrete; ceiling finished with brown coat of gypsum plaster and a smooth white finish coat

137A—Same as 137, except 3-in. Thermax removed and standard high-rib metal lath attached to top of joist; 2½-in. concrete poured on top of lath; battleship linoleum cemented to top of concrete

137B—Same as 137A, except ceiling of panel 137A removed and standard high-rib metal lath attached to underside of joist; scratch and brown coats gypsum plaster; smooth, white finish

701—2 × 8-in. joists; ½-in. fiberboard lath and ½-in. gypsum-plaster ceiling; 1-in. pine subflooring and 1-in. pine finish flooring.

702—Same joists and ceiling as 701; 1-in. pine subfloor; ½-in. fiberboard, 1 × 3-in. sleepers, and 1-in. pine finish floor

703—Same as 701, except a second ½-in. fiberboard and ½-in. plaster ceiling added to the existing ceiling of 701 by means of 1 × 3-in. furring strips

704—Same joists and floor as 701, except ceiling consisted of ½-in. fiberboard, 1 × 3-in. furring strips, ½-in. fiberboard lath, and ½-in. plaster face

705—2 × 8-in. joists, 16-in. o.c., 1-in. pine subfloor, 1-in. pine finish floor; 1 ceiling consisting of ½-in. plaster on ½-in. fiberboard lath next to joists; an additional plaster and fiberboard lath ceiling on 2 × 2-in. joists suspended by screw eyes and wire loops 36 in. o.c., 4-in. below upper ceiling, 5 × 5 × 2-in. fiberboard-block pads at fastenings

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* From CHRISLER, V. L., *Building Materials and Structures*, National Bureau of Standards.

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SECTION 32

INDUSTRIAL CONTAINERS AND PACKAGING EQUIPMENT

BY P. P. KENNEDY, M.E.

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INDUSTRIAL CONTAINERS AND PACKAGING EQUIPMENT

BY P. P. KENNEDY, M.E.

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INTRODUCTION

The past 20 years have seen a tremendous growth in the science of economic preparation, packaging, and transportation of commodities produced by American industry. During the last decade with the impetus afforded by the Second World War, industrial packaging and packing have matured to the extent that a new profession has been born—packaging engineer. Today, a packaging engineer must know the chemical and physical properties of construction materials used in containers, the economic and functional basis of container design, the adaptability of containers to plant handling facilities and methods of transportation, and the customers' methods of handling and merchandising. In addition, the packaging engineer must know the optimum equipment to fill, seal, label, and pack the many different types of containers with an almost innumerable number of commodities, each having its own particular peculiarities.

Accordingly, it is impossible to state in a handbook the most efficient container or piece of packaging equipment to use for all commodities. It is hoped that the description and comments following will result in a better understanding of certain basic principles whereby the selection and design of containers and related equipment may be improved.

The containers and equipment covered in this section will pertain to the so-called "industrial" type. Small packages usually for retail trade constitute another complex field of the "package engineer." Readers whose interests lie in that field are referred to the "Modern Packaging Encyclopedia," published by the Packaging Catalogue Corporation, for information on small containers and allied weighing, filling, checkweighing, sealing, etc., equipment.

Inasmuch as the following is at best a generalization, it is suggested that shippers acquaint themselves with the packaging requirements and container specifications in the Consolidated Freight Classification and Agent H. A. Campbell's Freight Tariff No. 4, publishing the Interstate Commerce Commission Regulations for the Transportation of Explosives and Other Dangerous Articles by Freight including Specifications for Shipping Containers. Further, detailed container data may be obtained through procurement of a number of Federal Specifications listed on page 1901.

BAGS

Multiwall Paper Bags. Multiwall paper bags, as their name implies, are bags consisting of three to six plies of paper (two-ply bags are referred to as duplex bags). Each ply of paper may have a basis weight (weight per 500 sheets 24 by 36 in.) of from 40 to 70 lb if made of plain kraft paper. Frequently, one or more plies of a multiwall bag will consist of an asphalt-laminated sheet (two sheets of paper laminated with asphalt; a laminated sheet counts as one ply), asphalt- or wax-impregnated sheet, or other specially treated or laminated sheets to provide additional protection against moisture absorption, to provide greaseproofness, wet strength, heat resistance, etc.

The types of multiwall paper bags are described by the construction of the top and bottom of the bags. They are

1. Sewn open mouth (SOM) (see Fig. 32-1*a*)
2. Sewn valve (SV) (see Fig. 32-1*b*)

3. Pasted open mouth (POM) (see Fig. 32-1c)
4. Pasted valve (PV) (see Fig. 32-1d)

Several kinds of valves may be procured, although two are in common use. They are the plain valve and the valve with tuck-in sleeve. The latter is used where siftproofness is required or moisture absorption by the contents is to be avoided. The tuck-in sleeve is extra paper and requires an extra manufacturing operation and so costs more than the plain valve.

Sewn-bottom bags are nominally constructed with gussets and pasted bags without gussets (see Fig. 32-1). However, several bag manufacturers have installed equipment permitting the manufacture of gusseted pasted bags. The presence of gussets lends filled bags to more stable stacking and better appearance, and the gussets afford an excellent display surface for advertising copy when bags are stacked flat. It should be understood that gussets are not mandatory in a sewn-bottom bag. Nongusseted or flat bags are frequently specified when bags are of very heavy construction and no sifting of contents through the closure can be permitted. The use of a flat bag in such instances halves the amount of paper which the needle must penetrate, and the bag is of uniform thickness throughout its width.

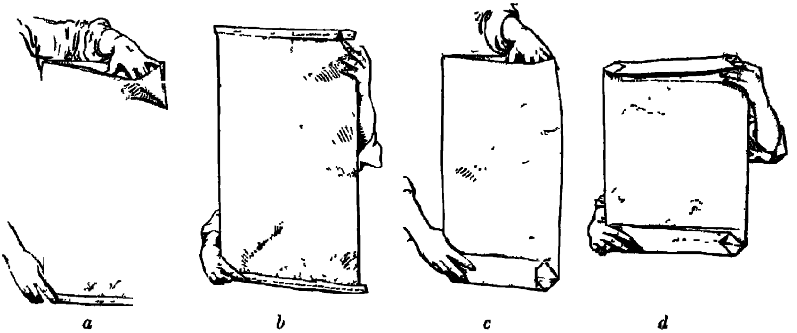


Fig 32-1. Types of multiwall paper bags.

Multiwall paper bags are of sizes and constructions to meet the trade demands. Maximum and minimum sizes will vary with various bag producers, but it is generally understood that bags ranging in capacity from 0 3 to 7.5 cu ft may be obtained. The proper size of bag to hold a specific amount of a given material must be determined by trial. However, Table 32-1 gives paper width and length dimensions for various capacities of bags from which bag sizes may be computed. Bag sizes arrived at by use of Table 32-1 are for sewn open-mouth bags. Methods of converting such bags to pasted or valve bags are given in explanation notes with Table 32-1. In choosing a bag size it should be borne in mind that an ideal bag for handling and stacking is one *which when filled* and lying flat has a length equal to twice the width and depth 30 to 40 per cent of the width. Generally not over 100 lb of product should be shipped in a paper bag.

The construction, *i.e.*, number and basis weight of plies, of a multiwall paper bag will depend upon (1) density of product, (2) net weight, (3) product characteristics, (4) content protection required, (5) value of contents, and (6) method of transportation. No set rule can be established for determining bag construction. Experience is the only teacher. With a range in bag strength from 120 lb of paper (3 plies of 40-lb paper) to 360 lb (6 lb of 60-lb paper) and the capacity range mentioned above, a wide variation is possible. Several important factors in determining bag construction should be remembered:

1. The larger the bag, the more strength required.
2. The heavier the net weight, the more strength required.
3. Products of relatively high value ($0.50 to $1.00 per lb) deserve more protection than cheaper products.

Table 32-1. Method of Determining Sizes of Multiwall Paper Bags

Volume, 100-lb bag, cu ft		Product density, lb/cu ft		Bag lengths, in., for paper-roll widths shown																					
Min	Max	Max	Min	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	43	44	45	47	48	49	52
3 9540	4 0336	25 3	24 8																			58	57	57	53
3 8563	3 9200	26 0	25 7																			57	56	55	52
3 7526	3 8074	26 7	26 3																			56	55	54	51
3.6526	3 6960	27 4	27 0																			55	54	53	50
3 5535	3 5857	28 2	27 8																			54	53	52	49
3 4552	3 4765	29 0	28 8																			53	52	51	48
3 3663	3 4170	29 8	29 3																57	56	53	52	51	50	47
3 2615	3 3242	30 6	30 2																56	55	54	51	50	49	46
3 1557	3 2323	31 6	30 9																55	54	53	50	49	48	45
3 1002	3 1413	32 3	31 8																54	53	52				
3 0145	3 0756	33 2	32 5																53	52	51	49	48	47	44
2 9296	2 9800	34 1	33 5																52	51	50	48	47	46	43
2 8455	2 8854	35 1	34 7																51	50	49	47	46	45	42
2 7444	2 7938	36 4	35 8																50	49	48	46	45	44	41
2 6446	2 7032	37 8	37 0																49	48	47	45	44	43	40
2 5981	2 6136	38 5	38 2																48	47	46	44	43	42	
2 5059	2 5461	39 7	39 3																47	46	45	43	42	41	39
2 4148	2 4488	41 4	40 9															48	46	45	44	42	41	40	38
2 3248	2 3584	43 0	42 4														48	47	45	44	43	41	40	39	37
2 3052	2 3052	43 3	43 3																						36½
2 2359	2 2802	44 8	43 8													48	47	46	44	43	42	40	39	38	36
2 1483	2 2029	46 5	45 4												48	47	46	45	43	42	41	39	38	37	35
2 1039	2 1265	47 5	47 0														45	44	42	41	40			36½	34½
2 0618	2 0977	48 5	47 7												47	48						38	37	36	34
2.0154	2 0510	49 6	48 8												46	45	44	43	41	40	39	37	36½	35½	33½
1 9652	1 9977	51 0	50 1										48	47		44	43	42	40	39	38	36½	36	35	
1 9150	1 9500	52 2	51 3										47	46	45							36	35	34½	
1.8715	1 9063	53 4	52 5											45	44	43	42	41	39	38	37	35½	34½	34	
1 8301	1 8542	54 6	53 9										46		43	42	41	40	38	37	36½	35	34	33½	
1 7932	1 8146	55 7	55 1										45	44					37½	36½	36	34½	33½	33	
1 7481	1.7753	57 0	56 3											43	42	41	40	39	37	36	35½	34	33	32½	
1 7135	1 7363	58 3	57 6									45	44						36½	35½	35	33½	32½	32	
1 6742	1 7026	59 7	58 7										43	42	41	40	39	38	36	34	34½	33	32	30½	
1 6342	1 6608	61 2	60 2									44			40	39	38	37	35½	34½	34	32½	31½	31	
1 5963	1 6276	62 6	61 4								45	43	42	41				36½	35	34	33½	32	31	30½	
1 5684	1 5832	63 7	63 1											40	39	38	37	36	34½		33	31½		30	

1.5359	1.5603	65.2	64.0								44	42	41				36½	35½	34	33½	32½	31	30½
1.4932	1.5197	66.9	65.8						45	44	43	41	40	39	38	37	36	35	33½	33	32	30½	30
1.4650	1.4826	68.3	67.5						44	43	42					36½	35½	34½	33	32½	31½	30	29½
1.4348	1.4539	69.7	68.7									40	39	38	37	36	35	34	32½	32	31	29½	29
1.4111	1.4245	71.1	70.3					45	43	42	41				36½	35½	34½	33½	32	31½	30½		28½
1.3765	1.3985	72.6	71.5				46	44	42			39	38	37	36	35	34	33	31½	31	30	29	
1.3555	1.3703	73.8	73.0				45			41	40			36½	35½	34½	33½					28½	28
1.3252	1.3485	75.4	74.1					43	41			38	37	36	35	34	33	32½	31	30½	29½	28	27½
1.3037	1.3160	76.7	76.0				44			40	39		36½	35½	34½	33½		32	30½	30	29		
1.2745	1.2975	78.4	77.0			45		42	40			37	36	35	34	33	32½	31½	30	29½	28½	27½	27
1.2518	1.2654	79.8	79.1				43			39	38	36½	35½	34½	33½		32	31	29½	29		27	26½
1.2362	1.2457	80.9	80.3			44		41	39							32½	31½				28		
1.2171	1.2306	82.2	81.8				42			38	37	36	35	34	33	32		30½	29	28½		26½	26
1.1872	1.2081	84.3	82.8			43		40	38		36½	35½	34½	33½	32½	31½	31	30	28½	28	27½	26	
1.1630	1.1783	86.0	84.9		44		41			37	36	35	34	33	32		30½	29½		27½	27		
1.1421	1.1605	87.6	86.2			42		39	37	36½	35½	34½	33½	32½	31½	31	30		28		26½	25½	
1.1150	1.1355	89.6	88.1		43	41	40		36½	36	35	34	33	32	31	30½	29½	29	27½	27		25	
1.0859	1.1048	92.0	90.5		42		39	38	36	35½	34½	33½	32½	31½	30½	30	29	28½	27	26½	26	24½	
1.0658	1.0787	93.8	92.6	43		40				35	34	33	32	31		29½		28	26½	26	25½		
1.0443	1.0636	95.8	94.0		41		38	37	35½	34½	33½	32½	31½	30½	30	29	28½	27½		25½	25	24	
1.0258	1.0400	97.5	96.1	42		39		36½	35	34	33	32	31		29½		28		26				
1.0060	1.0230	99.5	97.8		40		37	36	34½	33½	32½	31½		30	29	28½	27½	27	25½	25	24½		
0.9808	1.0011	102.0	99.9	41	39	38	36½	35½	34	33	32	31	30½	29½	28½	28	27	26½			24		
0.9627	0.9766	103.9	102.4	40			36	35	33½	32½	31½		30	29		27½		26	25	24½			
0.9433	0.9686	106.1	104.3		38	37	35½	34½				30½	29½	28½	28	27	26½		24½	24			
0.9268	0.9405	107.8	106.3	39		36½	35	34	33	32	31	30	29		27½		26	25½					
0.9403	0.9186	110.5	108.8		37	36	34½	33½	32½	31½	30½	29½	28½	28	27	26½		25	24				
0.8880	0.9002	112.7	111.1	38	36½	35½	34	33	32	31	30	29		27½		26	25½						
0.8658	0.8812	115.3	113.5		36	35	33½	32½	31½	30½	29½	28½	28	27	26½	25½	25	24½					
0.8498	0.8618	117.6	116.0	37	35½	34½	33	32	31	30	29		27½		26		24						
0.8272	0.8431	120.9	118.5	36½	35	34	32½	31½	30½	29½	28½	28	27	26½	25½	25	24½	23½					
0.8140	0.8246	122.9	121.5	36	34½	33½		31				27½		26		24½	24						
0.7983	0.8114	125.4	123.3	35½	34	33	32		30	29	28	27	26½	25½	25								
0.7808	0.7944	128.1	126.0	35	33½	32½	31½	30½	29½	28½			26		24½	24							
0.7653	0.7763	130.7	129.0	34½	33	32	31	30	29	28	27½	26½	25½	25									
0.7429	0.7595	134.6	131.7	34	32½	31½	30½	29½	28½	27½	27	26	25	24½	24	23½							
0.7289	0.7378	137.5	136.0	33½	32	31	30	29	28	27	26½	25½		24	23½	23							
0.7088	0.7214	141.0	138.8	33	31½	30½	29½	28½	28½	26½	26	25	24½		23								
0.6988	0.7063	143.1	141.6	32½		30	29	28	27		25½		24	23½									
0.6839	0.6950	146.4	143.9	32	31					26	25	24½		23									
0.6691	0.6820	149.7	146.8	31½	30½	29½	28½	27½	26½	25½	24½	24	23½										
0.6556	0.6641	152.7	150.6	31	30	29	28	27	26				23										
0.6410	0.6495	155.8	154.1	30½	29½	28½	27½	26½		25	24	23½											
0.6253	0.6301	159.9	158.4	30	29	28	27	26	25½	24½	23½	23											
0.6073	0.6133	164.7	163.1	29½	28½	27½	26½	25½	25	24	23	22½											
0.5872	0.5966	170.4	167.8	29	28	27	26	25	24½	23½	22½	22											

Table 32-1. Method of Determining Sizes of Multiwall Paper Bags (*Cont*)

Volume, 100-lb bag cu ft		Product density lb/cu ft		Bag lengths in for paper roll widths shown																					
Min	Max	Max	Min	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	43	44	45	47	48	49	52
0 5731	0 5807	174 5	172 2	28½	27½	26½	25½	24½	24	23	22														
0 5545	0 5638	180 3	177 4	28	27	26	25	24	23½	22½															
0 5436	0 5493	184 1	182 1	27½	26½	25½	24½	23																	
0 5316	0 5373	188 1	186 1	27	26	25		23½	22½	22															
0 5185	0 5270	192 9	189 9	26½	25½	24½	24	23	22																
0 5001	0 5106	200 0	195 9	26	25	24	23½	22½																	
0 4845	0 4944	206 4	202 3	25½	24½	23½	23	22																	
0 4691	0 4821	213 2	207 4	25	24	23	22½	21½																	
0 4539	0 4626	220 3	216 3	24½	23½	22½	22	21																	
0 4388	0 4474	227 9	223 5	24	23	22	21½																		
0 4298	0 4335	232 7	230 7	23½	22½		21																		
0 4155	0 4240	240 7	235 9	23	22	21½	20½																		
0 4009	0 4093	249 4	244 4	22½	21½	21																			
0 3875	0 3947	258 0	253 3	22	21	20½																			
0 3734	0 3804	267 8	262 9	21½	20½	20																			
0 3602	0 3662	277 6	273 1	21	20																				
0 3532	0 3532	283 1	283 1	20½																					
0 3404	0 3404	293 8	293 8	20																					

Example Determine bag size for bag to hold 100 lb of product weighing 35 lb per cu ft From table go down column of product density to 35 lb per cu ft and go horizontally to various bag lengths listed Arbitrarily choose a bag length say 45 in for which a paper roll width of 49 in is indicated Subtract 1 in from roll width for longitudinal bag seam (49 − 1 = 48), divide remainder by 2 (48 = 24) Flat bag will be 24 by 45 in For gusseted bag arbitrarily choose gusset size (2½ to 5 in) For this example say gusset of 4 in Bag size then becomes 20 by 4 by 45 in

To determine bag sizes for other than 100 lb of material use cubic foot capacity column for base point

To convert sewn-open-mouth sewn closure bag to pasted bottom bag sewn closure, assume sewn-open-mouth bag 18 by 3 by 36 in

Deduct from length ½ in gusset width—1½ in.
Deduct from length ½ in for sewing— ½ in
2 in

$$36 - 2 = 34 \text{ in}$$

Add to tube length one-half width of bottom plus 1¼ in. Assume 5 in wide bottom—$\frac{1}{2} \times 5 + 1\frac{1}{4} = 3\frac{3}{4}$ in

$$34 + 3\tfrac{3}{4} = 37\tfrac{3}{4}$$

Pasted bag size—21 by 37¾ in.

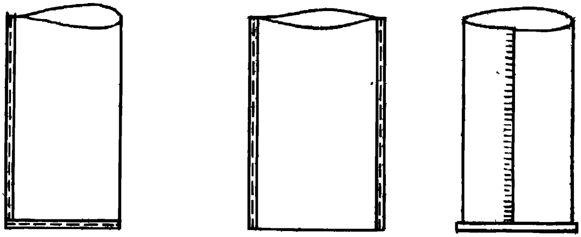
4. Less-than-carload and less-than-truckload shipments demand containers of greater strength than carload or truckload.

For new users of paper bags it is recommended that complete facts be given the bag supplier and his recommendations be followed by shipping tests in order to assure proper bag construction.

A new type of multiwall paper bag consisting of three or more plies of finely creped paper with all plies loosely adhered with asphalt, latex, or some similar adhesive is now manufactured. While the creping requires additional paper compared to a plain multiwall bag and extra manufacturing operations result, it is said that less paper strength is required, owing to the creping which permits the bag to stretch when subjected to rough handling. In addition, the creped paper gives excellent stack stability.

Textile Bags. Textile bags are commonly limited to burlap and cotton and usually require a paper liner either attached to the fabric or entirely separate and inserted in the bag prior to filling.

Textile bags are made in a wide variety of sizes and grades or strengths of fabric. Burlap is specified on the basis of weight per yard of 40-in.-wide goods; *i.e.*, 8-oz burlap means 36 in. of 40-in.-wide burlap weighs 8 oz. Burlap shipping bags are normally made of burlap weighing from 7½ to 12 oz. Cotton or osnaburg shipping bags are specified on the basis of yards of a given width of material weighing 1 lb, or the number of ounces 1 yd of a specified width will weigh; *i.e.*, 2.25–36-in. osnaburg means 2.25 yd of 36-in.-wide material weighs 1 lb, or 40-in.–7-oz means 1 yd of 40-in.-wide material weighs 7 oz.



Sewn one side and bottom

Sewn two sides seamless bottom

Cemented center seam, taped bottom

Fig. 32-2.

Textile bags are always of a flat (nongusseted) construction. However, the seams will vary depending upon the protection desired for the bag contents and/or the size of the bag relative to the width of goods from which the bags are to be made. Various types and locations of seams for plain textile and waterproof-paper-lined (WPPL) textile bags are illustrated in Fig. 32-2.

Both burlap and osnaburg bags have high tensile strength in either direction of the fabric and high tear resistance. Accordingly, weights up to 225 lb are safely shipped in textile bags, although the handling of such heavy bags is cumbersome. Textile bags normally have a relatively high resale or reuse value in comparison to paper bags and for that reason are favored by some industries.

DRUMS

Drums are straight-sided or bilge-type containers of circular cross section and may be fabricated from metals, rubber, wood, fiberboard, or combinations of these materials.

Metal Drums. Metal drums range in size from 5- to 110-gal capacity and in gauge of metal from 28 to 12 gauge (steel). The two broad classifications into which drums are segregated are (1) *tight head*, in which the top and bottom heads are attached to the drum body by welding, double seaming, or a combination of the two, and (2) *open head*, in which construction when the cover is removed the drum opening is the same diameter as the side wall of the drum at the section where the cover is applied.

Tight-head Drums—Liquid Contents (*see Fig.* 32-3). These drums are of either two- or three-piece construction. Two-piece drums are made only in 16 and heavier gauges and are fabricated from two pressed sheets circumferentially welded near the middle of the drum body. Three-piece drums consist of a circular body longitudinally welded with the two heads double-seamed, welded, or double-seamed and welded to the body. When heads are attached by double seaming only, it is necessary that a seaming compound be employed which serves as a gasket in the seam. This compound must be one which is unaffected by the drum contents. For this reason it is important that a drum supplier be advised of the nature of the contents being packed. Drums meant for returnable service or those containing dangerous liquids frequently have chime reinforcements. A chime reinforcement consists of a heavy-gauge steel ring (usually two gauges heavier than the drum body) seamed around the chimes of a drum (see Fig. 32-4). Openings for liquid drums are usually of the screw or cap type and are welded or pressed in place. Welded forged fittings are employed for some drums containing dangerous liquids and for heavy-duty drums. Pressed-in fittings are used for single-trip drums and those used in the lighter returnable-drum service.

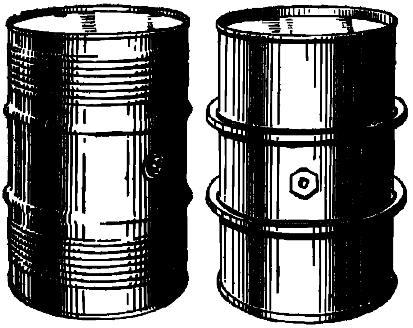


Fig. 32-3. (*Left*) Drum with expanded rolling hoops and corrugations in body. (*Right*) Drum with I-bar rolling hoops.

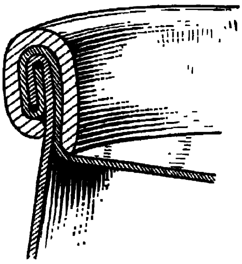


Fig. 32-4. Section of top chime showing double seam and chime reinforcement.

Fittings are available in many types and sizes, but ¾-, 1½-, and 2-in.-diameter fittings are considered standard. Of prime importance in the closure of a drum is the gasket material used between the plug and the flange seat and, in the case of pressed-in fittings, the gasket material between the flange and the body or head of the drum. In effecting a closure, it is necessary to see that the gasket is of proper size and resilience, that it is not twisted, and that the proper amount of torque is applied in tightening the plug. Some types of gaskets and/or closures will not be liquid-tight if too much torque is applied. Drum bodies may be reinforced by rolling hoops and in the case of lighter gauge drums (18 gauge and less) by corrugations in the drum body. I-bar hoops are either 1 by 1½ in. or ¾ by 1¼ in. and are usually fastened to the drum by welding or by swedging. I-bar hoops are employed only on drums of 16 guage and heavier. Swedged hoops are expanded from the drum body and are normally used with drums made of 16-gauge steel or lighter. The so-called light-gauge drums (18 gauge and less) frequently have the sections of the body between the rolling hoops and the head stiffened by corrugations, usually three to five corrugations per drum end with the depth of the corrugations less than the depth of the rolling hoops. Drums for dangerous liquids usually have the heads convex which adds strength to the head and helps to prevent distortion.

Tight-head Drums—Dry Contents. Drums for dry products may be constructed similar to drums for liquids or may have a lock-type of longitudinal seam known as "box" or "Gordon" seams (see Fig. 32-5). These seams are frequently used on drums containing nonhygroscopic products and for products which are molten at time of packing but solidify when cooled. Closures for tight-head drums containing dry products will vary in size from 5 to 16 in. in diameter, but in no instance should they be larger than the head diameter minus 4 in.

Open-head Drums—Dry or Liquid Contents. Open-head drums may be either straight side or bilge type and of two- or three-piece construction. The bodies of the bilge type may be seamless with the bottom head brazed or welded in place, or the body and bottom head may be entirely seamless. The three-piece drums are made in the conventional manner previously described. With open-head drums the top of the body is rolled or curled outward to provide a seat for the cover and to stiffen the body. Closures for open-head drums may be lug type (for small light-gauge drums only), bolted ring seal, or lever-lock ring seal (see Fig. 32-6). Of interest in the open-head-drum picture is a new screw-type closure. This is obtained by a deep drawing of the portion of the cover which is inside the periphery of the body and expanding an "interrupted thread" in several portions of the vertical section of the cover. Similar female threads are expanded in the top side wall of the drum body. Cover closure is made by an approximate 30 deg turn of the cover.

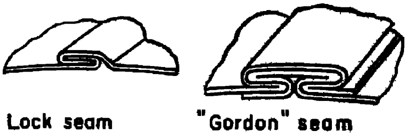


Fig. 32-5.

Although carbon-steel gauges have been used in the foregoing, construction details other than gauge of metal would apply to drums constructed of stainless steel, aluminum, nickel, monel, and other metals.

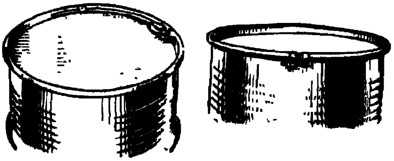


Fig. 32-6. (*Left*) Lever-lock ring. (*Right*) Bolted ring seal.

Of significance at the present time is the widespread interest shown in interior linings for metal drums. Galvanized and tin-dipped drums have long been available, and lacquer-lined drums have been used in the food industry for some years. At present a great deal of work on phenolic- and vinyl-resin-base linings is being done in an effort to produce a satisfactory interior lining which will permit the shipping of acidic and alkaline liquids and solids in carbon-steel drums rather than in more expensive containers.

Rubber-lined Steel Drums. Rubber-lined steel drums with open- and tight-head constructions have been used for many years. The lining is applied to the separate parts of the drum and the parts are then assembled. The tight-head drum is equipped with a special reversible pour-out spout and cap (see Fig. 32-7).

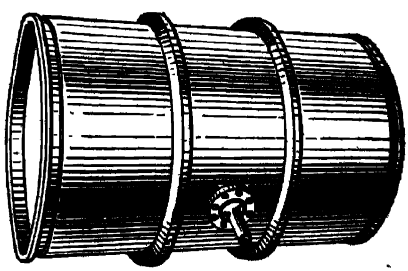


Fig. 32-7. Rubber-lined steel drum with pour-out spout in position.

Fiber Drums. Fiber drums are convolutely wound cylindrical paper shells equipped with fiber, wood, or metal heads. A section of one type of chime construction is shown in Fig. 32-8. These drums have found wide acceptance for shipments of dry and solid products, viscous liquids, and pastes.

Fiber drums are made in sizes ranging from 1- to 75-gal capacity. The majority of capacities are available in two different sizes which permits the nesting of empty drums, thus decreasing freight costs and warehouse space. Because of the wide range of capacities and the variety of products shipped in fiber drums, several strength constructions are available for most of the capacities manufactured. Increase in strength is obtained by adding to the number of plies making up the bodies and heads or by employing higher strength fiberboard. In addition, on some sizes of drums, the bottom chimes are reinforced with a metal hoop. To prevent damage to drums setting on wet platforms, etc., those drums having fiber bottoms are treated. High resistance to moisture-vapor penetration through the fiber is obtained by incorporating in the side wall moisture barriers such as aphalt-impregnated paper, plastic wax, regenerated cellulose, Pliofilm, metal foils, and other moisture-vapor barriers. In the same manner, resistance to chemical attack, fiber contamination, etc., may be obtained by having such a treatment or barrier on the inside facing of the drum. Disks of the same materials are used between the product and the cover to assure complete protection to the contents.

Outside treatments or coatings for fiber drums are also available. Such treatments permit limited exposure to water without deleterious effect to the drum plus enhancing the appearance.

Economically, the fiber drum is competing with light-gauge steel drums, slack barrels, and plywood drums. It merits careful investigation.

Plywood Drums. Plywood drums are straight-sided containers constructed of two or four plies of veneer bonded to each other. The grains of adjacent plies are always at right angles to one another. Although several types of plywood drums have been in general use, a single type is the only one currently being used. This drum is constructed by butting the ends of a plywood sheet and securing the seam by wire stitching a metal strip to the butt seam. Heads are held in place by means of head liners on both sides of the top and bottom heads. These head liners are thin strips of wood, usually elm, which are wire-stitched or stapled to the drum body. The exterior end sections of the drums are reinforced with wood or metal bands which also are stapled. This construction results in a reinforced end or chime, permitting rolling of the drums on their heads, and also keeps the heads of the drums elevated from the floor. Because of the metal stitching of the longitudinal seam, heavy gummed kraft-paper tape is frequently specified to go over the seam on the interior of the drum to afford a smooth interior. It is customary to use a creped-paper drum or barrel liner with plywood drums.

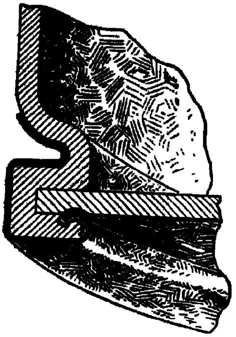


FIG. 32-8. Section view of bottom chime construction of one type of fiber drum. Note the metal chime reinforcement.

Plywood drums are available in sizes of approximately the same capacities as fiber drums. However, it is not customary to make two different sizes of plywood for a given capacity, and nesting of empty drums is not standard practice.

Four-ply veneer drums are used for export shipments and for very heavy loads in domestic service.

BARRELS

Barrels are uniformly bulging side cylindrical-cross-section containers constructed of wood staves bound together with hoops and have flat wood heads of equal diameter. Figure 32-9 illustrates and names the various parts of a barrel. Barrels are classified in two major categories—slack and tight.

Slack barrels are those barrels having parts not sufficiently tight to hold liquids and are therefore used for shipping dry products. Slack barrels are available in capacities from 12 to 64 gal (see Table 32-2 for barrel dimensions and capacities) and in four classifications—commodity, No. 1, mill run, and No. 2.

Commodity barrels are used for heavy loads and are essentially the same as No. 1 with the exception that staves are ½ in. thick, hoops are usually of heavier gauge or wider than those used on No. 1 barrels, and more hoops are employed.

No. 1 barrels are siftproof and are made of No. 1 grade staves and heading. The staves are ⅜ in. thick and tongued and grooved. The heading is ½ or ⅝ in. thick depending on the barrel capacity and the load it is to carry, and heads are single piece, *i.e.*, tongued and grooved and glued or butt-jointed and glued. Content weight up to 600 lb may be shipped in No. 1 barrels provided head liners or cleats are used.

Mill-run barrels are not siftproof and are made of mill-run staves (containing at least 40 per cent of No. 1 staves) and mill-run heading (containing at least 50 per cent of No. 1 pieces). Staves are ¾ in. thick and may be tongued and grooved or butt-jointed as specified. Heading is ½ in. thick and may be made of three to five pieces.

No. 2 barrels are not siftproof and are made of No. 2 staves and heading. Thickness of staves and heading is the same as for mill-run barrels. Generally weights not exceeding 450 lb are shipped in these latter two grades of barrels.

Slack barrels normally have bilge and head hoops of steel, although the commodity barrels are equipped with quarter hoops owing to the heavier weights contained. Wood and wire hoops are also employed, and small economies may be effected by substituting wood or wire hoops for steel hoops.

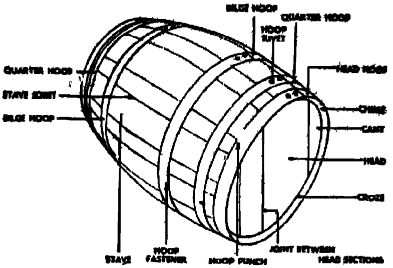


Fig. 32-9. Tight-barrel nomenclature.

As all slack barrels require the replacement of one head after being filled, it is important to note that bilge and quarter hoops should be driven tight after the head is in place and the hoops fastened by hoop-fastener nails or other means of equal efficiency. The head hoops and head are secured by nails or staples driven through the head hoop and into the head. The barrel supplier should be contacted for full details on the proper method of heading. When wood or metal cleats are used to reinforce the heads, they should be positioned perpendicular to the grain of the heads.

Tight barrels are those barrels having parts sufficiently tight to hold liquids without

Table 32-2. Slack-barrel Dimensions and Capacities (Approximate)

Capacity, gal	Head diam, in.	Stave length, in.	External bilge circumference, in.
12	12½	21	47 3
18	14⅛	24	54 1
21	15½	24	59 5
25	17⅛	24	65.4
30	16½	30	62.6
35	17⅛	30	68 0
38	17⅝	30	70 6
43	19⅛	30	74 8
46	20	30	77.0
48	19⅛	32½	74.9
51	19⅛	34	76 0
54	20	34	78 6
60	22	32	84 0
64	22	34	85 0

Table 32-3. Tight-barrel Dimensions and Capacities (Approximate)

Capacity, gal	Head diam, in.	Stave length, in.	External bilge diam, in.
5	10⅜	17½	38 5
10	12½	21⅜	47 5
15	14⅝	23⅜	55 0
20	16	23⅝	60.0
25	16½	29½	62 5
30	17¾	29½	67 5
50	21⅜	33¼	80.0
57	21⅝	35¼	83 0

They are manufactured in nominal capacities from 5 to 57 gal (see Table 32-3 for barrel dimensions and capacities). Tight barrels are of heavier construction than slack barrels and thus are higher priced. Because of this, tight barrels are frequently used as returnable containers.

Staves and heading of tight barrels are usually made of white oak, red oak, fir, gum, or ash. Various grades of white oak may be specified, *i.e.*, sap free, sap clear, tight sap, etc., depending on the quality of barrel required.

Staves and heading will range in thickness from ⅝ to 1 in. Staves are always butt-jointed, while the heading may be doweled and flagged or glued.

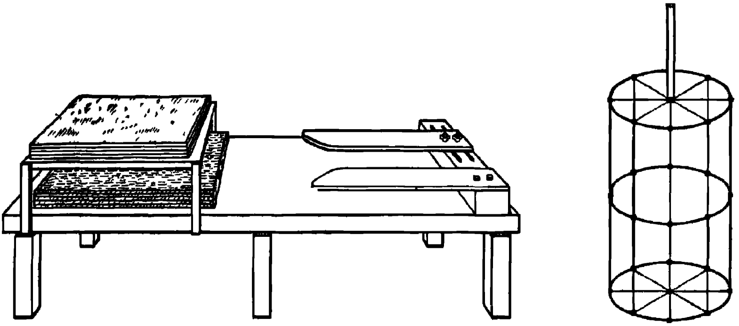
Hoops for tight barrels are always steel and may vary in number from six to eight hoops and in size from 1 in. wide by 19 gauge to 1¾ in. wide by 16 gauge. The locations of hoops are bilge, second quarter, first quarter oil head for eight-hoop barrels. When six hoops are specified, the barrels do not have the second-quarter hoops.

As tight barrels are used for liquids, they generally have an interior treatment of some kind, the most well known probably being the charred interior of whisky barrels. However, other linings such as animal glue, paraffin, asphalt, silicate of soda, and rubber lattices have been successfully employed.

Generally, tight barrels are filled through a bunghole located in the bung stave. Bungs should be treated with the same material as the drum interior, or a piece of suitably treated or coated cloth should be placed over the bunghole prior to driving the bung in place to assure maintaining product quality. Wood bungs should be driven with the grain of the bung parallel to the grain in the bung stave. Bungs should be secured by means of a bung strap placed across the bung and driven into the barrel.

LINERS

It is frequently necessary to utilize liners, which are not an integral part of a shipping container, with various types of containers in order to (1) prevent contents from sifting, (2) afford water and/or moisture-vapor protection, and (3) provide a chemically resistant material between contents and container. Liners may be fabricated from paper, flexible films, or combinations of both. They must be of a size to fit the interior dimensions of the shipping container without strain being placed on the liner or liner closure when contents shift during the course of handling and shipping. Liners must have sufficient flexibility and stretch to withstand the distortion which the shipping container may undergo. Liner closures may be made by tying, sewing, adhesives, or heat sealing. The type of closure will depend upon the type of shipping container, the liner material, and the degree of protection to be afforded by the liner.



Form for inserting liners in bags

Form for inserting liners in barrels or drums

FIG. 32-10. Devices for inserting liners in bags, barrels, and drums.

Paper liners, other than box or case liners, are usually creped or crinkled so as to stretch 15 to 25 per cent when tension is applied in the creped direction. Liners are creped or crinkled in one direction (single stretch) or in two directions (all-way stretch). Paper liners are made in various basis-weight papers and are designated by the basis weight of the paper before creping. They may be untreated paper, wax-impregnated (5 to 60 per cent by weight) paper, asphalt-laminated paper, or plastic-coated paper. Manufactured seams are usually made by sewing or with adhesives.

Film liners may be manufactured from polyethylene, polyvinylidene chloride, rubber hydrochloride, and other materials depending upon the chemical or moisture-vapor resistance required. These films do not require creping as they have inherent stretch characteristics to obviate the necessity for creping. Film liners are usually

1 to 5 mils in thickness depending upon weight of load and physical characteristics of the film. Manufactured seams are normally heat-sealed. Closure is made by tying or heat sealing. Increasing use is being made of film liners owing to their physical and chemical properties which permit the shipping of pastes and liquids in metal or fiber drums and corrosive solids in film-lined bags.

It is important that all liners be inserted in containers properly or their function may be lost. Devices similar to those shown in Fig. 32-10 are recommended for inserting liners in bags and barrels or drums. The correct procedure is to slip the liner over the form and, in the case of barrels or drums, to fold the liner "ears" over the bottom; insert the form in the container and fold the top of the liner over the top of the container; remove the form, fill the container, and then bring the top of the liner together and tie or wire the liner closed about 2 in. from the end of the liner. *Never tie the liner close to the product.* When film liners are used it is essential that the interior of the shipping container be free from sharp projections or burrs which might snag the liner. In addition, hermetically sealed film liners should have as much air as possible removed from the interior prior to sealing so as to avoid film breakage due to air pockets.

BOXES

Fiber Boxes. Two types of fiberboard are utilized for making fiber shipping boxes—corrugated fiberboard and solid fiberboard.

Corrugated fiberboard consists of two sheets called liners or facings between which is adhered a corrugated medium. Linerboard is normally made of pure kraft or jute (reclaimed papers) and is designated by caliper (points) and weight (pounds per 1,000 sq ft). Typical linerboard specifications are shown in Table 32-4.

Table 32-4. Typical Corrugating Medium and Linerboard Specifications

Type	Caliper, in.	Bursting strength, psi	Weight, lb per 1,000 sq ft
Corrugating medium:			
Straw	0.009	......	32–33
Kraft	0.009	....	26–28
Linerboard:			
Jute	0.012	....	50
	0.016	100–110	64
	0.023	140–150	85–90
	0.030	190–200	105–110
Cylinder kraft	0.016 dry	105–115	50–52
	0.016 semiwater	115–125	52–54
	0.016 water	125–150	54–56
	0.030 semiwater	185–200	90–95
	0.030 water	220–250	95–100
Fourdrinier	0.012	60–70	33
	0.014	75–85	38
	0.016 dry	85–100	42
	0.016 semiwater	100–105	47
	0.016 water	105–110	52
	0.023	135–150	69
	0.030 semiwater	145–160	88
	0.030 water	150–165	95

Note: To determine the weight of combined board, use the above figures plus factors of 1.55 and 1.33 for A and B flutes, respectively, for weight of corrugating medium, plus the following weights of adhesives:

	Silicate of soda	Starch
A flute	12 lb/1,000 sq ft	7–9 lb/1,000 sq ft
B flute	14 lb/1,000 sq ft	8–10 lb/1,000 sq ft

Linerboard is either of fourdrinier or cylinder construction indicating the method employed in the manufacture of the liners. Most kraft liners are fourdrinier type which results in a better balance of liner strength between "machine direction" and "across-machine direction" than is obtained with cylinder kraft or jute board. Also jute board usually weighs more per given area than kraft board of similar strength because of the extra material required owing to the use of reclaimed paper stock.

Corrugating material is usually strawboard, chestnut board, or kraft and is 0.009 in. thick and weighs 24 to 28 lb per 1,000 sq ft. Three types of corrugations are employed—(1) A-flute corrugations being 3⁄16 in. high and having 34 to 38 corrugations per linear foot; (2) B-flute corrugations, 1⁄8 in. high and having 50 to 52 corrugations per ft; and (3) C-flute corrugations, 3⁄16 in. high and 42 to 44 corrugations per ft. A-flute corrugations are employed for maximum cushioning, whereas B-flute corrugations are less easily crushed and are used for heavier loads. The C flute is essentially a compromise of the A and B flutes. When double-wall corrugated board is specified, it is usually composed of A- and B-flute corrugations, although some suppliers may be able to furnish AA, BB, or CC if necessary.

When corrugating material is combined with the liners, the grain of the liners is perpendicular to the corrugations. Hence the normal corrugated box is cut from the board so that the corrugations are vertical in the sides and ends of the box.

Solid fiber boxes are constructed of a solid board made up of three or more plies of paperboard glued together. Solid fiberboard may be constructed entirely of kraft or, as is usual, made with chipboard filler and kraft facings or linings. These facings may be treated with resins to provide high scuff resistance and resistance to water absorption. Increased resistance to water absorption is obtained if waterproof adhesives or barriers are employed in the fabrication of the board.

When specifying the strength of corrugated or solid fiberboard desired, it is customary to specify the minimum bursting strength per square inch (Mullen test) and/or the minimum caliper and weight of the liners or solid board.

Both corrugated and solid fiber boxes are made in a variety of styles and are sized to meet each product requirement. Styles of boxes in common usage are illustrated in Fig. 32-11. Close examination of the several styles illustrated will show

1. *Regular Slotted Box.* Manufacturers' joint or seam is at vertical edge. Inner and outer flaps are same width which, unless box is square, results in a gap between the inner flaps when box is sealed.
2. *Center Special Slotted Box.* Inner flaps are cut so that they will butt the same as outer flaps when box is sealed. This type of box has two thicknesses of material over entire top and bottom area. When box is longer than it is wide, the center special slotting requires the use of more material and an extra labor operation than regular slotted-style boxes of similar inside dimensions.
3. *Overlap Slotted-style Box.* Inner flaps are cut to same width as outer flaps, but the outer flaps will overlap at least 2½ in. or will overlap the entire width of box (full overlap).
4. *Telescope-style Box.* This box consists of two pieces—top and bottom. Each piece must be assembled by the user by wire stitching or staples. The top or cover telescopes the bottom resulting in two thicknesses of board along the sides and ends and single thickness of board of top and bottom. (Extra thickness results at ends or partway along sides depending on how the top and bottom are slotted.)

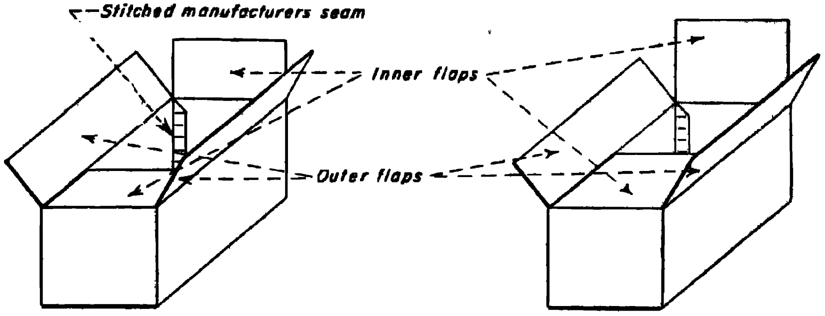
The size of a fiber box is always designated by the inside dimensions in the order of length, width, and height. It is well to remember that the flaps result in double or almost double coverage and that the area which the flaps will cover should be kept at a minimum to keep box cost down.

The slotted styles of boxes are made with a manufacturers' seam which is where the body of the box is joined. This seam may be made with a gummed tape, either cambric or reinforced paper, or by cutting the body of the box so that there is a flange which will fold against the side or end and be stapled to the side or end. The stapled manufacturers' joint is more rigid than the taped joint but also costs more owing to the extra board and a more laborious seaming operation.

Wood Boxes. Solid wood boxes are those constructed entirely of wood and utilizing nails for fasteners. The styles of wood boxes are shown in Fig. 32-12 and are as follows:

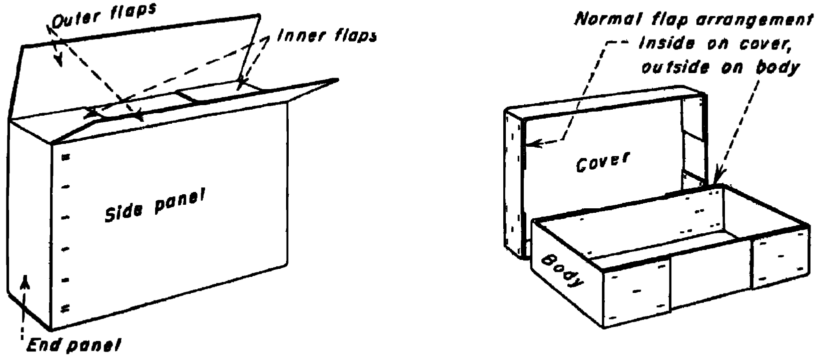
Style 1 is the cheapest and weakest construction of wood box. Top and bottom nailed into ends and sides, sides nailed into end grain of ends result in poor nail-holding power and frequent splitting of ends.

Styles 4 *and* 5 are a single-cleated-end box in which cleats may be placed on outside of box as shown or inside box (style 5). Cleats are of length equal to outside height of box minus ¼ in. Cleats are nailed to ends with nails clinched. Side-board nailing is staggered, nails being driven alternately into end grain of end boards and side grain of cleat. Cleats not only strengthen end by decreasing the unsupported length of the end board but also result in greatly increased nail-holding power of nails securing the sides. Style 5 boxes are used where box contents are of a shape which permits the placing of the cleats inside thereby decreasing the cubic displacement of the box. Style 2 boxes are sometimes made with the cleats horizontal and the grain of the end pieces running vertical. This is done to provide hand holds for lifting the box.



Regular slotted style

Center special slotted style



Full overlap style

Full telescope style

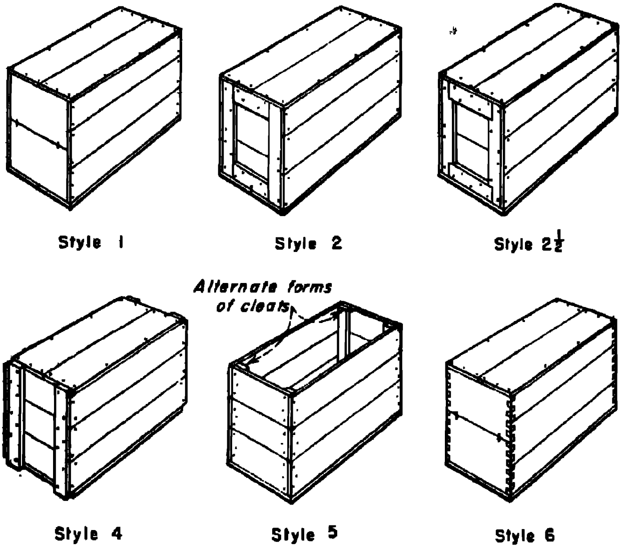
FIG. 32-11. Four styles of fiber boxes and the nomenclature of parts.

Styles 2 *and* 2½ are double-cleated-end boxes. Both vertical and horizontal cleats are provided on each end. These styles are used for heavy loads and for loads which will give a high end thrust (welding rods, bolts, etc.). For boxes having small end areas, it is frequently more economical to use double ends (two thicknesses of end boards) instead of cleating.

Style 6 is the lock-corner box. The sides and ends are joined by gluing the mortise and tenon joints. The top and bottom are secured by nails. This style box is normally classed as comparable to the load-carrying ability of style 1. However, through the use of good lumber and the proper cutting of the tenons and adequate gluing, this style is believed superior to style 1.

Assuming the choice of the proper style of box, the dependability of wood boxes is governed by (1) lumber used and (2) nailing. Wood should be good-grade box lumber free from loose knots, shakes, cross grain, and tight knots which will interfere with nailing. Lumber should not contain more than 18 per cent moisture. Wood is classified into four groups. The principal woods in each group are as follows:

Group I. White pine, ponderosa pine, spruce, cottonwood, yellow poplar, basswood, willow, cedar, white fir, butternut



Style 1 Style 2 Style 2½

Style 4 Style 5 Style 6

Fig. 32-12. Styles of nailed wood boxes.

Group II. Southern yellow pine, hemlock, Douglas fir

Group III. White elm, sweet gum, sycamore, black ash, black gum, tupel ft maple

Group IV. Hard maple, beech, oak, rock elm, birch, white ash, hickory

It will be noticed that these groups are arranged from soft to hard woods. The thicknesses of pieces and the size of nails used in box assembly depends upon what group of wood is used in the box construction. Table 32-5 shows the commonly accepted thicknesses to be used for the various groups of wood for boxes to carry loads up to 1,000 lb. Also shown in the table are the nail sizes for the thickness of parts specified.

In addition to the information above, the following should be borne in mind so that good box construction will result:

1. Keep inside dimensions to a minimum so as to reduce possible shifting of load.
2. Width of parts making up sides and ends should not be the same.
3. Nails should be cement-coated box, sinker, or cooler nails except those nails which are clinched. These need not be cement coated.
4. When unsupported span of box part exceeds thirty times the thickness of the part, additional cleats or battens of the same size as end cleats should be nailed across all parts to reduce the unsupported span length.

Table 32-5. Thickness of Wood-box Parts for Various Loads and Size of Nails to Be Used

	Group I and II woods			Group III and IV woods		
Gross weight, lb	Thickness of sides, top, and bottom, in.	Thickness of ends and thickness and width of cleats, in.	Nail size	Thickness of sides, top, and bottom, in.	Thickness of ends and thickness and width of cleats, in.	Nail size
100	3/8	1/2 ends with 1/2 × 1 5/8 cleats	5d	5/16	7/16 ends with 7/16 × 1 5/8 cleats	5d
250	9/16	11/16 ends with 11/16 × 2 1/16 cleats	6d	7/16	9/16 ends with 9/16 × 1 3/4 cleats	5d
500	11/16	25/32 ends with 25/32 × 2 5/16 cleats	7d	9/16	11/16 ends with 11/16 × 2 1/4 cleats	6d
750	25/32	7/8 ends with 7/8 × 2 5/8 cleats	8d	11/16	13/16 ends with 13/16 × 2 1/2 cleats	7d
1,000	13/16	1 1/16 ends with 1/16 × 3 1/4 cleats	9d	25/32	3/4 ends with 3/4 × 2 3/4 cleats	7d

Table 32-6. Number and Sizes of Steel Strapping for Wood-box Reinforcement

Gross weight, lb	Size of flat band, in.			Size of round wire, gauge		
	One strap	Two straps	Three straps	One wire	Two wires	Three wires
100	1/2 × 0.020	3/8 × 0.020	3/8 × 0.015	12	12 1/2	14
250	3/4 × 0.020	5/8 × 0.020	3/8 × 0.020	10	10 1/2	12
500	....	3/4 × 0.023	1/2 × 0.023	....	9	11
750	...	3/4 × 0.028	5/8 × 0.023	......	8	10
1,000	....	......	3/4 × 0.023	....	....	9

Note: Boxes over 15 in. long should have two or more straps applied.

5. Large heavy boxes should have skids on bottom to facilitate handling.

6. Considerable increase in box strength may be obtained by correct application of steel strapping. Steel strap reinforces the nailing and should be positioned with that thought in mind. Table 32-6 shows the commonly used sizes of flat and wire strap and the number of straps to be used for various box weights.

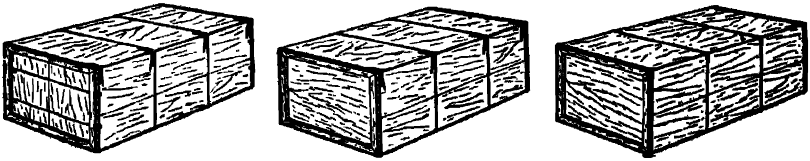


Fig. 32-13. (*Left*) All-Bound box. (*Center*) Rock-Fastener box. (*Right*) Regular-Twist box.

Wire-bound boxes are constructed of veneer, plywood, or resawn lumber having cleated ends and with girthwise wires stapled to the box blank for reinforcement. The use of wires permits decreased lumber thickness resulting in low tare weight and low initial cost. Three styles of wire-bound boxes are in common usage: (1) All-Bound, (2) Rock Fastener, and (3) Regular Twist (see Fig. 32-13).

The *All-Bound box* has the ends attached to the body by means of wires. The box is shipped knocked down and is set up by passing the ends of the wires attached to the box ends through the solts in the cleats and bending the wires back. No nails are required for assembly or closing this style of box.

The *Rock-Fastener box* consists of a body blank and two ends, the ends being separate. The parts are shipped knocked down, and the box is assembled by nailing through the body cleats into the end cleats. Closure is made by use of the rock fasteners.

The *Regular-Twist box* is essentially the same as the rock fastener with the exception that the closure is made by twisting the wires together which extend beyond the closing edge of the box.

Each of these styles of wire-bound boxes can be obtained in thickness from 1/10 to 3/8 in. for veneer and plywood and 3/16 to 3/8 in. for resawn lumber.

The cleats furnished on wire-bound boxes may be either mitered or tongued and grooved. The former is considered more desirable from a strength standpoint.

Binding wires are available in a number of gauges and hardnesses and may be galvanized or black. It will be seen that the rock-fastener type of closure is designed for multiple openings, and accordingly the wire used must not be too hard or soft. The regular-twist closure is destroyed when opened, and soft wire is desirable for effecting a good twisted closure. Of interest is the fact that, if a rock-fastener closure breaks, a new closure piece may be spliced onto the box and satisfactory service continued.

Wire-bound boxes are manufactured in many different shapes besides the conventional rectangular container, depending upon the shape of the contents. In addition, wire-bound crates are also marketed.

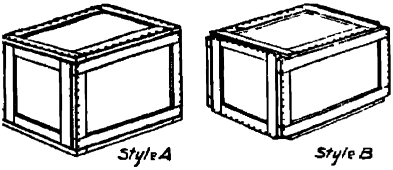


FIG. 32-14. Two styles of cleated plywood or fiberboard boxes with cleats at each edge of each panel.

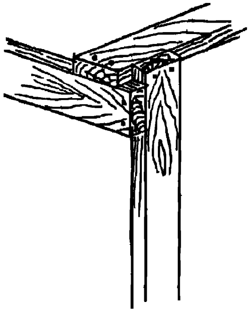


FIG. 32-15. Three-way cover construction for crates.

Inasmuch as wire-bound boxes are manufactured on automatic equipment which requires considerable setup time, it is not economical to consider using wire-bound boxes unless several hundred or more of one size of box can be purchased at one time.

Cleated Plywood and Fiberboard Boxes. Cleated plywood (fiberboard) boxes consist of panels of plywood having cleats attached along the edges. These panels are assembled by nailing. There are many styles of cleated boxes depending upon the number and position of the cleats. The strongest constructions are those which have two cleats at each box edge permitting side-grain nailing throughout. These styles are known as style A and style B and are shown in Fig. 32-14.

Cleated boxes are used for carrying weights up to 1,000 lb. The strength of the box depends upon the thickness of the plywood and the size and number of cleats and battens. The battens are used to reduce the unsupported span of the plywood and add considerable strength. The plywood is stapled to the cleats by wire staples or by flathead nails. Both staples and nails should penetrate the cleats and be clinched.

The same type of box utilizing fiberboard, either corrugated or solid fiber, is in use. The same method of construction is employed, although in attaching the cleats to the fiberboard, glue is sometimes employed. Boxes with fiberboard panels are not used to carry such heavy loads as plywood paneled boxes.

Crates. It is impossible in the space reserved to establish any concrete facts concerning proper crate construction. In a very general manner the following is offered as a guide:

1. Determine if open or closed (sheathed crates may be employed).
2. If size and weight of object to be crated are not excessive and quantities are of a good magnitude (1,000 and over), consider the use of wire-bound crates.

3. Mount the object on skids whenever possible.
4. Employ three-way corner construction which gives best nailing at corners (see Fig. 32-15).
5. Use diagonal members to prevent distortion of crate by weaving. The angles which the diagonal member makes with the horizontal should not be less than 30 or greater than 60 deg.
6. Use good-sized lumber for frame members, and cut down unsupported spans by intermediate vertical and horizontal members.

GLASS DEMIJOHNS AND CARBOY BOTTLES

A large number of 5-, 6½-, and 13-gal-capacity carboy bottles are used by the chemical industry as returnable containers, generally carrying acids. The 5- and 6½-gal bottles are machine made. The former are obtainable with either a cork or screw finish, whereas the latter are obtainable with screw finish only. Figure 32-16 shows data pertaining to the 6½-gal bottle.

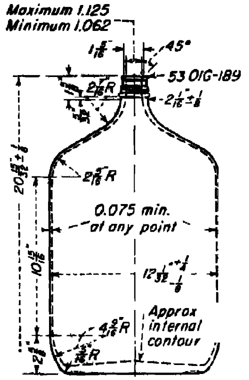


FIG. 32-16. A 6½-gal carboy bottle.

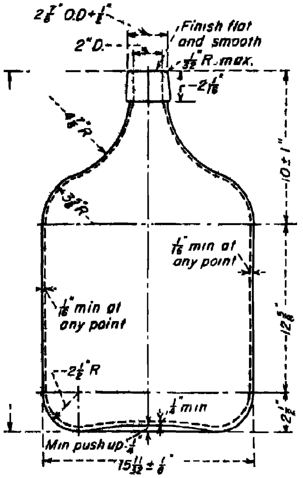


FIG. 32-17. A 13-gal carboy bottle.

The 13-gal carboy bottles are manually blown and may be obtained with either a plain cork finish or a ground-glass finish for a corresponding ground-glass stopper. Figure 32-17 shows the standard 13-gal bottle.

Carboy bottles must be packed in a shipping container for transportation. If dangerous articles are to be shipped, there are stringent requirements on the construction of the boxes and cushioning which must be used (see Agent H. A. Campbell's Freight Tariff No. 4). For nondangerous commodities the shipping container may be a fiber carton or drum, plywood drum, slack barrel, etc., with sufficient interior cushioning to prevent bottle breakage. Normally, it has been found to be economical to pack bottles in a substantial wood box with interior cushioning of relatively long life and use the carboys as returnable containers.

PACKAGING EQUIPMENT

It is again emphasized that this section deals with the "industrial" type of containers previously described. Other than equipment for container assembly, the users of "industrial" or bulk containers are essentially interested in equipment for weighing, filling, and closing. These operations will be discussed in the following material.

General. The proper choice of weighing and filling equipment will vary with the characteristics of the products to be handled, the filling rate required, and the capital expenditure which may be allocated. In many instances, there are several choices of equipment for a specific job. It is the engineers' duty to study the equipment available and utilize that which will return the greatest savings commensurate with investment. It is hoped that the following review will aid engineers in their job.

It is essential for accurate weighing of materials, dry or liquid, that uniform flow of the materials to a scale or to filling equipment be maintained. Primarily, in the case of dry materials, this will depend upon proper bin design. For free-flowing granular or crystalline products, this is not difficult, and conical bins may be used without fear of difficulty. However, with fine powders which will deaerate in storage and pack or "bridge" in a bin, designing to prevent this is difficult. With such products, it is suggested that the following be considered:

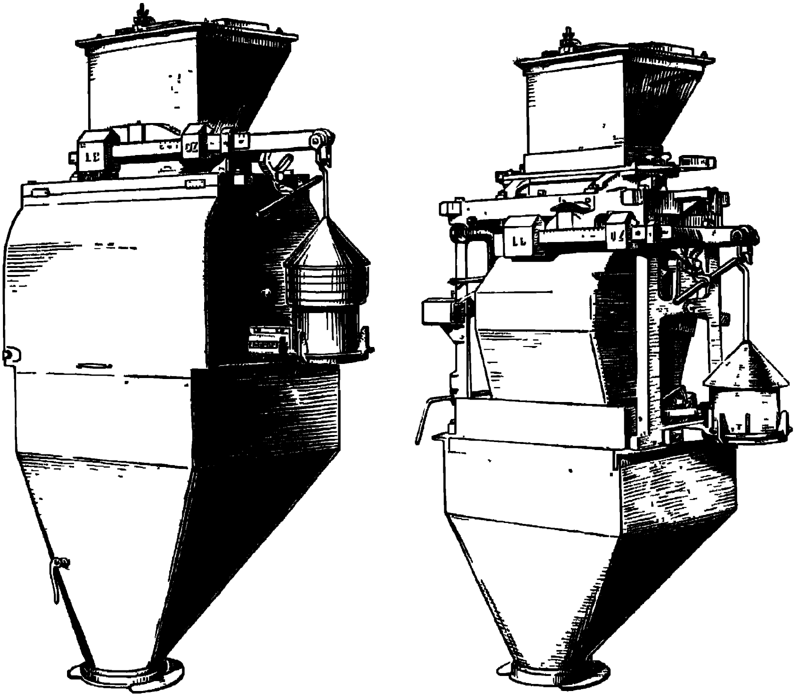


Fig. 32-18. Net-weigh scales. (*Left*) Hoepner Model 02 fully enclosed automatic discharge with counter. (*Right*) Hoepner 02.

1. Design bin with two vertical adjacent sides and two sides at an angle well in excess of the angle of repose of the material going in the bins.

2. Keep material moving by utilizing either internal or external vertical conveyor which will constantly remove material from the bottom of the bin and deposit it at the top.

3. Insulate bins from normal building vibration in order to keep packing tendency to a minimum.

If uniform flow of material is maintained, weighing equipment will give excellent results. With some types of filling equipment, it is necessary not only to obtain a uniform flow but also to maintain a uniform amount of material in the hopper of the filling equipment. This is easily accomplished by (1) having a hopper whose minimum size is at least twice the capacity of the container to be filled and (2) installing bin-level controls at a minimum and maximum hopper capacity which experience has indicated will give the best results.

Another factor which must be kept in mind when filling containers with dry materials is that, with all containers other than bags, it is desirable to fill them to their maximum capacity. With products that deaerate or settle, this means that filling equipment or auxiliary equipment such as vibrators which will settle the material as it is being packaged should be employed. However, a dense tight-filled bag is not so safe a carrier as one which has some slack. Accordingly, care should be exercised on the amount of deaerating obtained with bag-filling equipment.

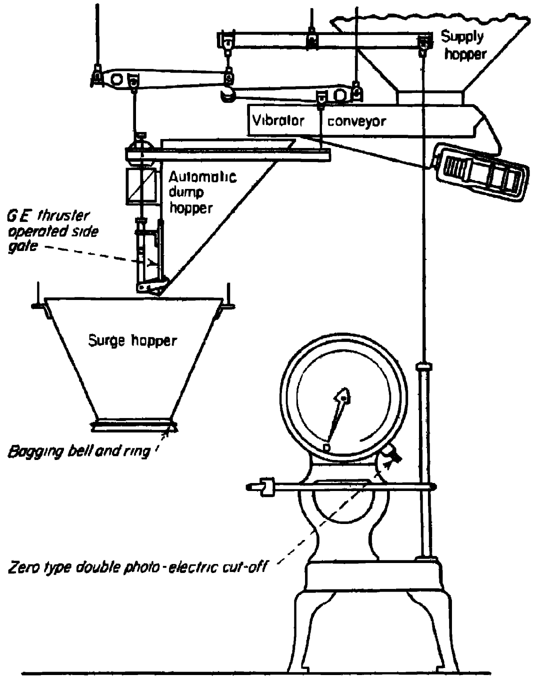


FIG. 32-19. Toledo automatic weighing and discharging equipment for filling bags. One or two vibrating conveyors may be used to feed material to the weigh hopper. These feeders may operate at one or two speeds depending upon filling rate and accuracy required.

Weighing Equipment. Two general methods of weighing are employed: (1) net weigher and (2) gross weigher.

In net weighing, the amount of material to be placed in a container is weighed independently of the container and then discharged into filling equipment or direct to a package. This type of weighing is considered more accurate than gross weighing, as variations in the tare weights of containers are not factors, and the total weight on which the percentage accuracy of the scale operates is less than with a gross weigher (see Fig. 32-18). Illustrated are several types of net-weigh scales. Bucket- or hopper-type scales, whether they be of the beam or dial indicator, are uniformly net-weigh scales in which material flows through a gate into the top of the scale bucket. As the weight of the material approaches the desired net weight, the movement of the weigh hopper causes the inlet gate to close, partially thus slowing the amount of incoming material to a dribble. When the final weight is reached, the gate is completely shut by mechanical trip. The charge is then released from the weigh hopper by push button or other manual control. Completely automatic weighing is accomplished by the same principle but utilizing mercoid switches and solenoids, etc., to release the charge and start another weigh cycle. Weighings at the rate of five to ten 100-lb units per min may be obtained.

Similar automatic weighing may be achieved with a dial scale equipped with magnetic or photoelectric cutoffs. With this setup weigh hopper and feed mechanism are actuated by the dial indicator making a magnetic circuit or intercepting a light beam at a predetermined setting. When the final weight is in the weigh hopper, a thruster is actuated which releases the charge from the hopper (see Fig. 32-19).

Other methods for obtaining accurate final weights by controlling the feed to the weigh hopper are (1) belt feed, (2) screw conveyors, and (3) vibrating feeders. Very close tolerances are realized by employing a bulk and a dribble feed simultaneously, with the bulk feed being stopped a few pounds short of required weight. The choice of method used will depend upon the physical characteristics of the material being weighed, weight accuracy desired, and production rate. It should be borne in mind that every product costs money to manufacture and that over-all plant yield is determined by the pounds of material sold. The closest tolerance consistent with product and packaging labor costs should be the goal of every weighing operation.

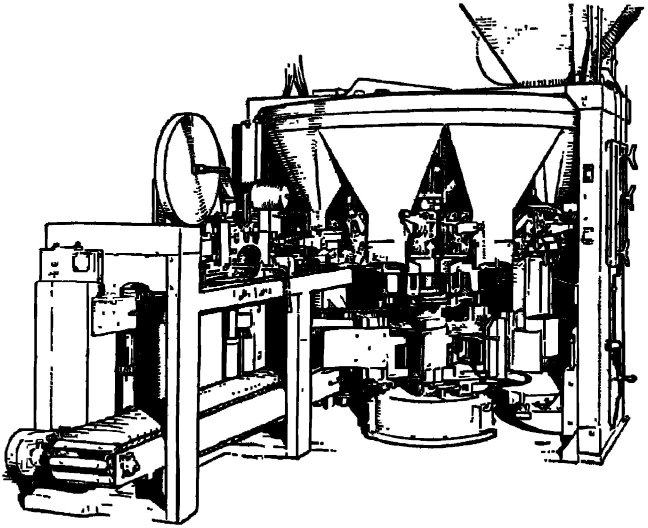


FIG. 32-20. Model A Bagpak. High-speed automatic weighing, filling, and sewing of multiwall paper bags.

Platform dial and beam scales are commonly used for gross weighing. The container is placed on the platform and tare weight adjustment made or the tare weight noted and added to the desired net weight. Dial scales for manual gross weighing are frequently used, as the approach of the indicator to the final weight acts as a signal to the operator to throttle the discharge valve and reduce the flow of material until the final weight is reached.

Bagging Equipment. *Open-mouth Bags.* The scales heretofore described and equipped or supplemented with bag holders may be used for accurate filling of open-mouth multiwall paper and texile bags at rates up to six 100-lb bags per min. For high bagging rates of fifteen to twenty 100-lb bags per min, the Model A Bagpak machine shown by Fig. 32-20 may be used. This unit weighs, fills, and closes bags at the described rate using but one operator. The unit consists of a battery of three or four scales mounted over a rotary turntable on which are suspended a number of bag holders with surge hoppers. The operator positions bags on the holders as they rotate. When the empty bags are indexed to the proper position under the scales, the charge is released into the bag. As the filled bags continue around the turntable on a joggling roller conveyor, the material is settled leaving sufficient outage for the bags to be sewn. When the bags reach a position approximately 270 deg from the operator, they are automatically transferred to a conveyor running tangential to the turntable and with the help of chain belts the bags are guided through a sewing machine and then to conveyor to warehouse or freight car. A similar smaller unit for ten to twelve bags per minute is also manufactured.

A simple and relatively inexpensive bag holder and scale for manual filling and weighing is shown in Fig. 32-21. This unit is a bag holder mounted on a small hopper

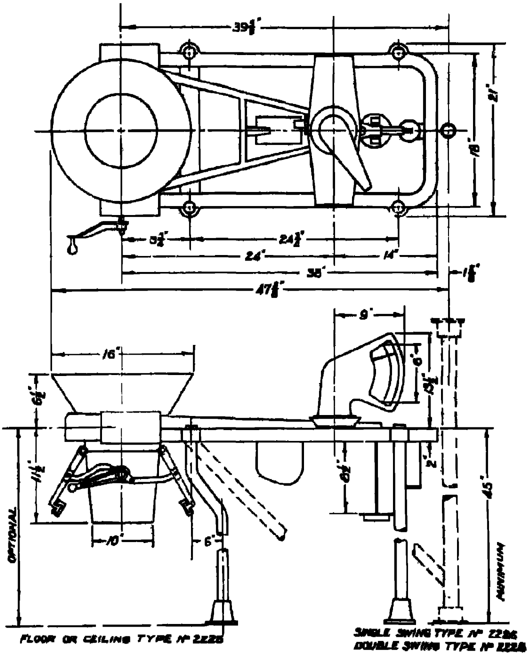

39¼"
21"
18"
5½"
24¼"
24"
14"
38"
1½"
47½"
9"
16"
6½"
10"
6"
45"
OPTIONAL
MINIMUM
FLOOR OR CEILING TYPE Nº 2225
SINGLE SWING TYPE Nº 2226
DOUBLE SWING TYPE Nº 222A

FIG. 32-21. Exact-weight bag holder and sacking scale.

pivoted on a frame. The frame may be stationary or pivoted. The bag holder and hopper are at one end of a lever arm to which weights equaling the gross weight are attached at the other end. An "under-over" dial indicator guides the operator in adjusting the flow of material to the bag. This scale permits filling to final weight in one operation, the weight accuracy depending upon the skill of the operator.

Many dry materials are of a nature which prohibits their being discharged from a scale hopper into a bag because of serious dusting. In such instances where open-mouth bags are used, it is desirable to fill the bags by means of an auger packer (see Fig. 32-22). This equipment consists of a small hopper feeding a vertically mounted auger. A counterweighted platform serves as a base for the container during the fill time. The auger may be belt- or direct-motor-driven through a clutch. In addition to the counterweight on the platform, the packers may be obtained with an adjustable brake shoe which gives added compression against which the filling auger must act. In operation, an empty bag is slipped over the end of the tube surrounding the auger and up the tube until the bottom of the bag is at or near the bottom of the tube. A foot pedal or push button is then actuated which causes the platform to rise until it is directly under the bag. At this location the platform trips a switch which throws in the clutch and starts the auger. As material is forced into the bag, the bag and platform descend until a second switch is contacted which disengages the clutch, thereby stopping the flow of material. The platform continues to descend until it is at floor level where the bag is removed. Inasmuch as this is a volumetric filling method, it is necessary to checkweigh the filled bag prior to closing. While considerable deaeration of product may be realized by adjusting the brake pressure and using augers of varying pitch, it is frequently necessary to compromise the deaeration with dustless operation and higher filling rate in order to achieve the optimum operating efficiency.

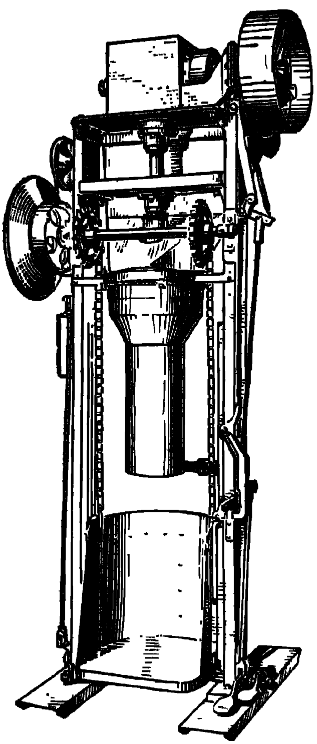


FIG. 32-22. Vertical auger packer for filling bags, barrels, and drums.

Valve Bags. Numerous types of valve-bag filling equipment are available depending on the bagging rate desired and the physical characteristics of the material to be bagged. Basic types of valve-bag packers are (1) impeller, (2) belt, (3) auger, and (4) gravity. The principles of each type are explained below.

1. IMPELLER. The impeller-type packer consists of a housed rotating shaft on which are mounted blades (similar to a turbine). Material is fed from a hopper into the housing where it is picked up by the blades and given sufficient velocity to force it through the spout outlet of the housing on which the bag is held. The impellers may be mounted vertically or horizontally depending upon material feed. This type of packer is normally used with nonabrasive materials which will not break down by impact of the impellers.

2. BELT. Belt packers (see Fig. 32-23) are used for fast bagging of free-flowing granular and crystalline products. The belt packers may be either the preweigh or gross-weigh type. If preweigh, the material is discharged from a scale into the hopper of the packer. The hopper outlet is constricted so that the material is fed into a groove in a rotating wheel. The material is held in the groove through 90 deg change in direction by an endless belt which runs against the flanges on either side of the wheel. As the material leaves the groove, it is forced through the filling spout. The preweigh type of packer permits vibration of the bag during the filling cycle. This is accomplished by automatic joggling of the rest on which the bottom of the bag is placed.

The gross-weigh belt packer operates on the same principle as the preweigh except that there is a shutoff valve at the inlet to the rotating wheel, and the filling spout and bag rest are part of a beam scale. In operation, a control lever is positioned which causes the inlet valve to open and the grooved wheel to rotate. When the predetermined weight of material is in the bag, the lever arm trips which closes the inlet valve and disengages the clutch. It will be seen that uniform feed of material to the rotating wheel is essential if accurate weights are to be obtained. At the instant of cutoff the amount of material between the cutoff valve and the outlet of the filling spout will be added to what is already in the bag. If this amount is not fairly constant for each bag, poor weights will result.

3. AUGER. Auger-valve bag packers (see Fig. 32-24) are designed to be used on materials which are light and tend to deaerate and pack. Material is fed to the packer

hopper which is equipped with a ribbon agitator to prevent bridging. The material drops into an auger running from 1,200 to 2,000 rpm. This auger forces the material out the filling spout on which the bag is suspended. Frequently this type of packer is used for products which cannot be preweighed and discharged. In such instances, two types of auger packers are available. One type depends on a timer which stops the auger after a predetermined period of time. Here again it is of utmost importance that uniform feed to the packer be maintained for accurate weights. The second type recently introduced commercially weighs the material as it is forced into the bag and trips a switch stopping the auger when final weight is reached.

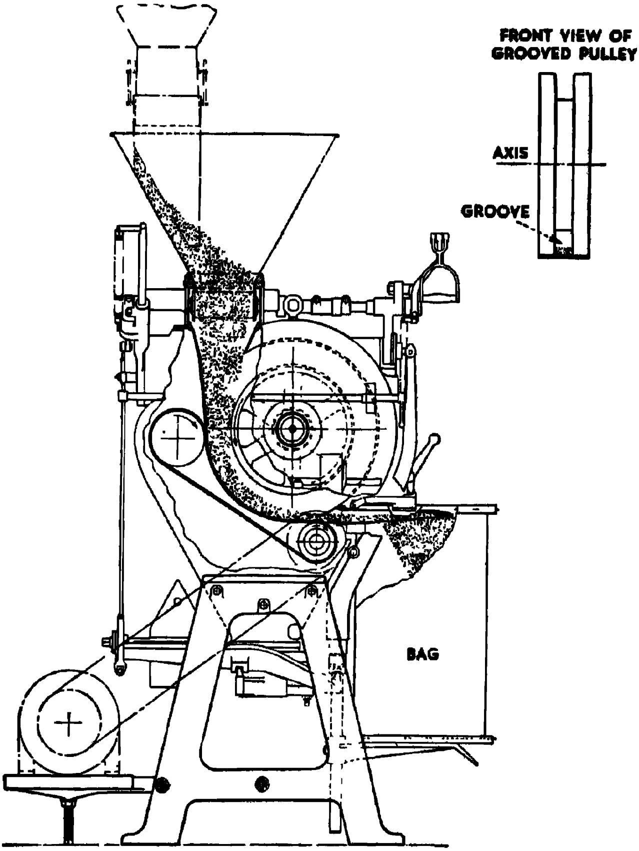


FIG. 32-23. Gross-weigh belt-type valve bag packer.

4. GRAVITY. Gravity-valve bag packers (see Fig. 32-25) are employed for free-flowing crystalline or granular materials where relatively low bagging rates are desired. The unit consists of a small hopper tapering to a filling spout. It depends on the free-flowing characteristics of the material and gravity to fill the bag. Material may be either preweighed and discharged into the hopper of the packer, or the packer may be mounted on a platform scale and the operator control the flow of material to the packer as indicated by the scale.

In considering valve bags and valve-bag packers, it should be kept in mind that they are designed primarily for high bagging rates with small labor requirements. This usually prohibits checkweighing and adjusting of every bag. However, because of the simplicity of closure inherent in valve bags, their use in broader applications is becoming more widespread. Weight adjustment may be made by means of scoops which will easily fit in the valve opening of the bag. Several scoops holding different quantities of the same material will facilitate the adjusting of bag weights.

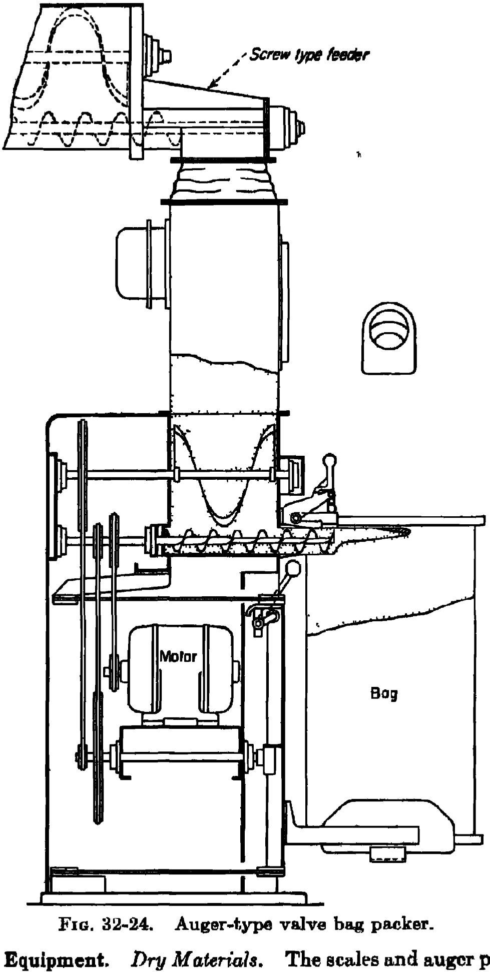


FIG. 32-24. Auger-type valve bag packer.

Drum-filling Equipment. *Dry Materials.* The scales and auger packers described in the foregoing section may also be used for filling drums and barrels. As previously stated, with these containers it is desirable to have them filled to capacity during storage and transportation. This not only reduces container, handling, and freight costs per unit of net weight but also permits better and safer storage and handling. To increase the amount of material which can be packed in a container, consideration should be given to (1) mechanical and electrical vibrators, (2) hydraulic jolters, and (3) vacuum deaeration. Mechanical vibrators are available having a fixed amplitude and frequency of vibration. Electrical vibrators have a fixed frequency, but the amplitude may be varied by means of a rheostat. Generally, the mechanical and electrical vibrators have a relatively high frequency which with some very light finely ground powders will not result in deaeration. Such materials may deaerate if the frequency of vibration is sharply reduced and the amplitude increased. Such an action may be obtained with an hydraulic jolter. The third method of deaeration—vacuum bell—is accomplished by placing the filled container in a vacuum chamber and after the proper amount of vacuum is reached opening the chamber to atmospheric pressure.

In effecting deaeration by the mechanical, electrical, or hydraulic equipment, two methods are optional: (1) vibrate the container while filling, or (2) fill the container, vibrate, and then refill the volume gained by vibration. If preweigh scales are used, it is possible to vibrate while filling. However, if the container rests on a scale platform while being filled, it is impossible to vibrate until after filling. In order to do this and avoid further filling, the use of a collar which will fit on top of the drum and increase the drum capacity temporarily may be used. After the material has settled to where it is all within the confines of the drum, the collar can be removed and the drums closed.

Equipment for filling rigid containers with powders by vacuum has been recently marketed. While similar equipment has been in use for 8 to 10 years for filling small containers, its adaptation to bulk containers is relatively new. Vacuum filling of powders is a quick dustless operation with a packed density of product higher than may be achieved by any other known packaging method. A vacuum filler consists of a vacuum source, vacuum tank, filter, valve assembly, filling head, and hopper and elevating container platform. The filling head is illustrated in Fig. 32-26. It will be noted that the filling head consists of centrally located holes connecting the bottom of the hopper with the container and two separate vacuum inlets which are screened. In operation a container is placed on the platform which ascends and seals the top of the container against a gasket in the filling head. Vacuum is then admitted in one side of the head which evacuates the air from the container. The pressure differential between the evacuated container and the material in the hopper causes the latter to rush into the container. After a fixed time interval, the vacuum is released on the one side of the head and applied at the other. When the vacuum is released, that side is open to atmospheric pressure, and the incoming air cleans the material off the screen on that side of the head. The timing of the application of vacuum to each side of the head and the number of cycles to a filling operation are controlled by a valve mechanism.

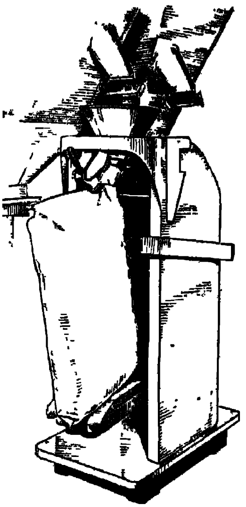


Fig. 32-25. Gravity-type valve bag packer.

Liquid Materials. Liquids may be packaged in drums or other containers by volume or weight. A simple inexpensive volume filler utilizes a float valve to cut off the flow of liquid when the proper level is reached in a container. The depth to which the valve will go in a container is adjustable. With volumetric filling of this type it is important that containers be uniform size.

Another more accurate method of volume filling is done by a metering device. Such a meter measures the amount of liquid flowing through it and automatically cuts off a set volume. The meter is equipped with temperature controls to compensate for temperature variations of the liquid which in turn affect volume. For best results a constant pressure head should be maintained. These meters may be equipped with a control valve for reducing pressures to practical working pressures and with strainers for ensuring a product free of foreign material. The meters are available in stationary and portable models.

Several semiautomatic and automatic methods of filling liquids by weight are utilized. Illustrated in Fig. 32-27 is the Vol-U-Meter method consisting of a valve and a mercoid switch with a trip lever. The latter is attached to the frame of a beam scale in such a manner that when the beam falls, indicating the desired weight is in the container, the beam hits the trip lever which actuates the mercury switch and closes the valve. The valve is opened by moving the manually operated lever.

A second method of weighing liquids is accomplished by utilizing a dial scale equipped with magnetic or photoelectric cell cutoffs connected to a two-stage solenoid-operated valve. In operation, the container is placed on the scale platform and the tare weight adjustment made. The desired net weight is fixed on the beam balance. As the dial indicator approaches the zero mark on the dial, it intercepts a light beam, and the first stage of the solenoid valve is actuated causing the valve partially to close. As the indicator continues to zero, the light beam is again free, and the second stage of the solenoid valve is actuated which shuts the valve completely. The extent to which the second stage of the valve permits the valve to stay open is adjustable so that whatever "dribble" flow is required can be obtained.

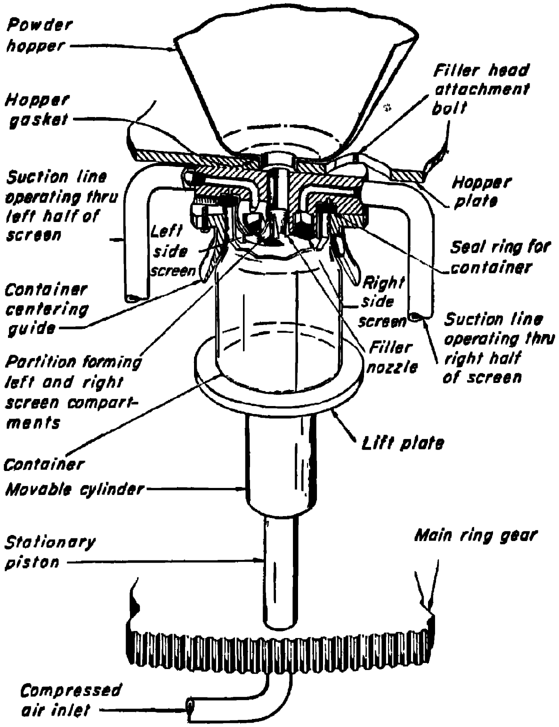

Powder hopper
Hopper gasket
Filler head attachment bolt
Suction line operating thru left half of screen
Hopper plate
Left side screen
Seal ring for container
Container centering guide
Right side screen
Suction line operating thru right half of screen
Filler nozzle
Partition forming left and right screen compartments
Lift plate
Container
Movable cylinder
Stationary piston
Main ring gear
Compressed air inlet

FIG. 32-26. Section of vacuum-filling head for filling rigid containers with powdered material. Flexible containers may also be filled by this principle by placing a rigid shroud around the container and evacuating the air between the shroud and container at the same time that air is evacuated from the interior of the container.

Case Packers. Case packers are utilized for semiautomatically loading canned goods and bottled goods in fiber and wood boxes and small paper bags in paper shipping containers.

Round cans are usually cased in equipment similar to that shown in Fig. 32-28. The cans enter on their sides and are guided into a form holding the desired number.

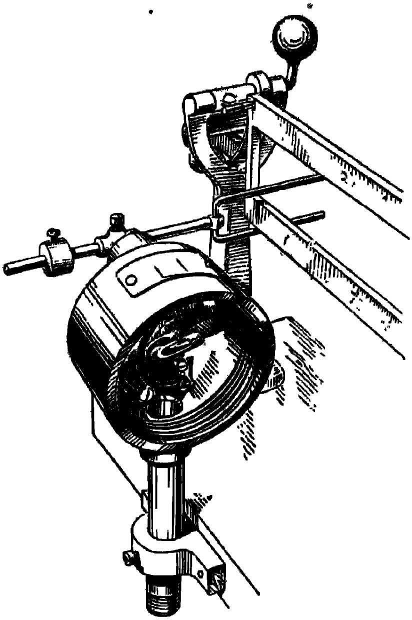


FIG. 32-27. Control switch which is attached to beam scale for actuating Vol-U-Meter filling valve.

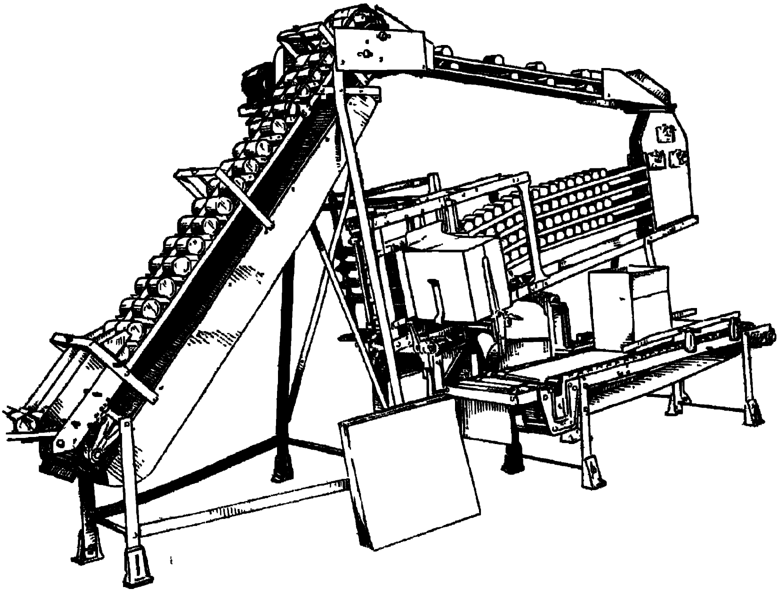


FIG. 32-28. Can case packer.

As one layer fills, the form drops, permitting the next layer to go into position. When the form is filled, an empty carton is placed over the opening and a foot pedal depressed. This actuates a thruster which pushes the cans into the carton. The carton is then lowered to a conveyor leading to a carton-sealing machine.

Bottled goods are kept vertical and separated on a conveyor into the number of rows which is equivalent to the rows of bottles in the case—usually four. The four rows then move forward simultaneously into a jig under which an empty carton is placed. A lever is tripped which lowers the bottles into the case (see Fig. 32-29).

Container Bag Packer. A paper shipping container bag is a multiwall paper bag in which a number of small bags or "pockets" are shipped. Equipment for filling container bags operates on somewhat the same principle as the can packer. Bags to be packed come to the packing station in a single line. The number of bags to go across the width of the container bag are then pushed by a thruster into a form over which the container bag has been placed. One layer of bags is positioned at each stroke of the thruster.

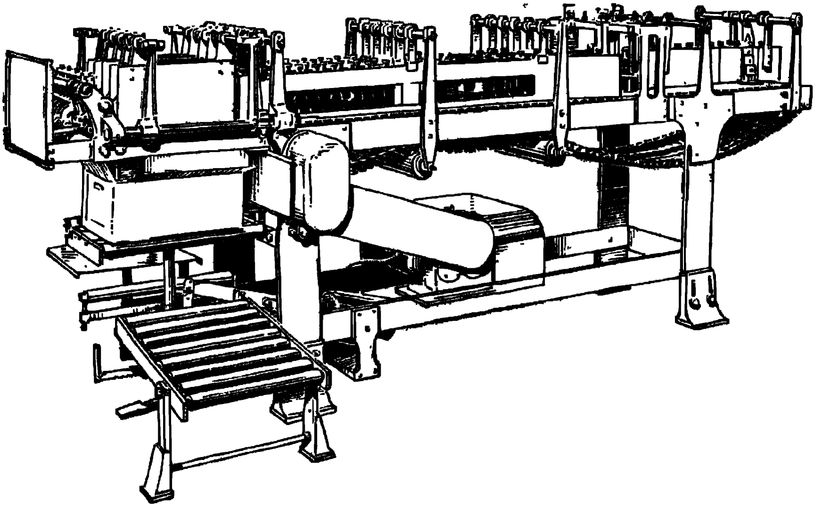
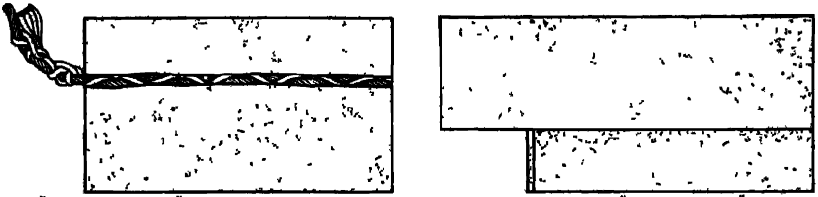


FIG. 32-29. Bottle case packer.

Closing and Sealing Equipment. *Open-mouth Bags.* Multiwall open-mouth paper and textile bags are closed by wire tying, stapling, and sewing. The first two methods are usually employed where bag production is low. This is for the reason that these methods require longer bags than the sewn closure, and thus bag cost is higher. When bags are wire tied, the mouth of the bag is gathered and a wire tie bent around the gather and twisted together by means of a tool. The gathering and application of the wire tie may also be performed automatically on equipment manufactured by the Hamer Machine Company of Minneapolis, Minn. If bags are paper and made of four or more plies, it is difficult manually to gather so much paper in which case the inner ply or plies may be folded down on the product and the outer three plies used for making the closure. When wire ties are used, care should be exercised that the tie is not applied close to the product. There should be 8 to 12 in. of outage, depending upon the bag size, and the tie applied 1 to 2 in. from the end of the bag.

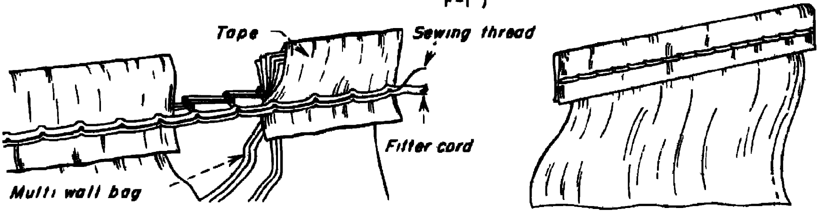
Stapled closures are made by folding down the top of the bag 1 to 2 in. and stapling through the fold. A second fold will make the bag more siftproof. Standard wire-stapling machines are used for this operation. If the bag is placed on a roller conveyor and the stapling head placed at an angle rather than horizontal, an easier operation will result.

The majority of open-mouth bags are closed by sewing Sewing machines utilizing a chain stitch, double-locked stitch, and patented "cushion" stitch are available. The details of these stitches are shown in Fig. 32-30. The two latter types of stitch are those normally used in closing paper bags, while the first two types are used for closing textile bags.



The "Cushion Stitch" closure before taping This type closure effected by models "E-I" and "F-I" Bagpackers

The taped and sealed "Cushion Stitch" closure This type closure effected by models "A" and "D-A" (Tape not applied by models "E-I" and "F-I")



Tape-bound sewed closure for multiwall paper bags

Fig 32-30 Sewn closures for multiwall paper bags

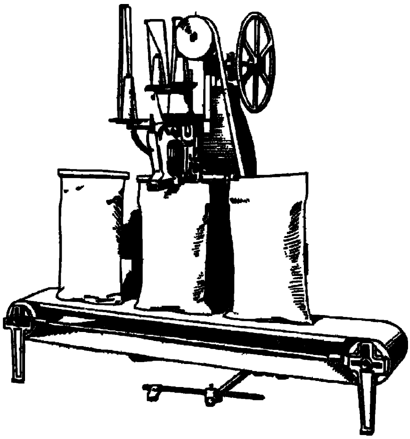


Fig 32-31. Union Special Style 21800 bag sewing machine with Style 80600H sewing head and 21844 tape-sealing attachment.

The Union Special Machine Company bag-sewing machines are available in several styles. A commonly used one is shown in Fig 32-31. Various types of sewing heads are applicable to the different machines. Sewing head styles 80600AC and 80600AD are used for closing heavy-weight textile bags. The AC produces the double-locked stitch and the AD the chain stitch. Styles 80600E and 80600F may be used for both paper and textile bags. A filter cord is normally used with paper bags. This filter cord fills the needle holes and prevents sifting. Sewing head style 80600H makes a tape-bound sewn closure with the stitches penetrating the crepe-paper tape. Simultaneously, a filter cord is run. This head produces the double-locked stitch and has an automatic mechanical tape and thread cutter.

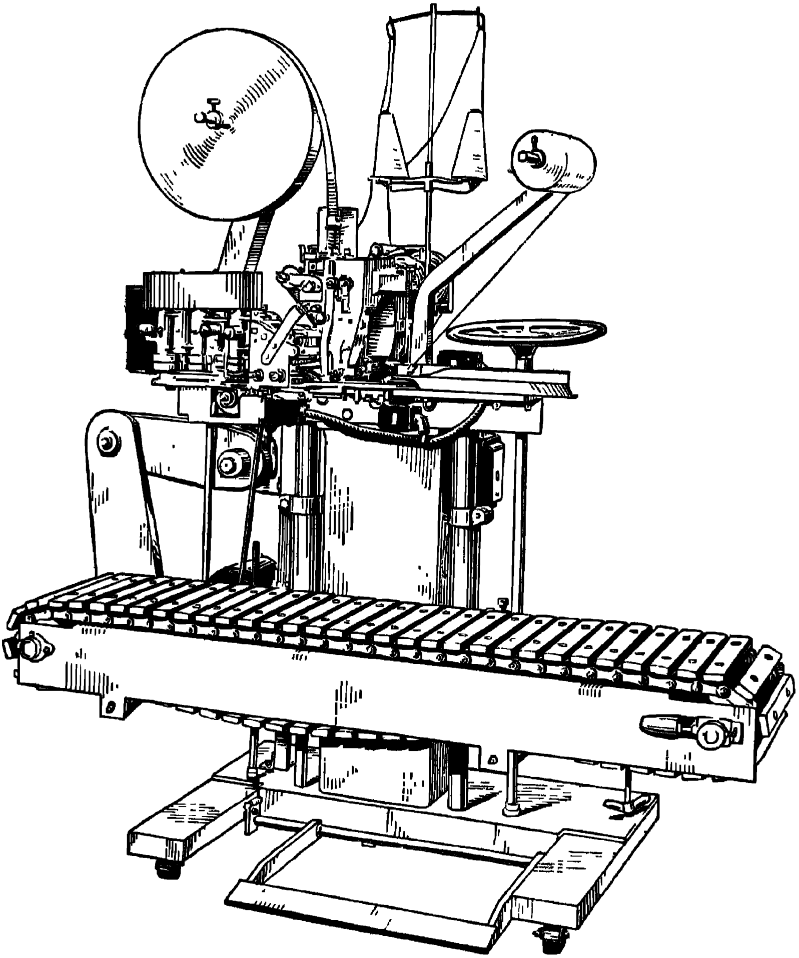


FIG. 32-32. Model DA Bagpak sewing machine for applying tape over "cushion-stitch" closure.

The Bagpak sewing machines are also available in different styles and with heads producing different types of stitching. In wide use is the Model DA Bagpak for closing multiwall paper bags (Fig. 32-32). This unit produces the "cushion stitch" illustrated. Paper tape is applied *over* the stitching if desired.

Fiberboard boxes may be closed by application of gummed paper or cambric tape to all open seams, by application of adhesive to the flap area, or by stapling Adhesive sealing may be entirely manual, semiautomatic, or completely automatic. A carton sealer of the latter type is shown in Fig. 32-33. Cartons are fed automatically to the entrance of the gluing unit where the outside top and bottom flaps are turned outward and pass under glue rolls. The flaps are then turned to their normal closed position, and the carton enters the compression unit consisting of spring-mounted idler rolls pressing down on the top flaps. The carton is carried through the gluer by a push-box conveyor and through the compression unit by a belt conveyor running on a steel bedplate. Speeds as high as 40 cartons per min may be obtained depending on the speed and box spacing of the push-box conveyor and the length of the compression unit.

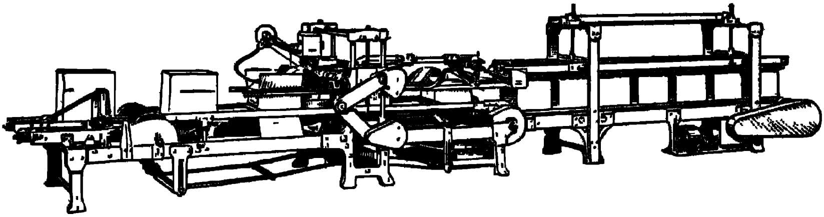


FIG. 32-33. Automatic case sealer (gluer and compression unit).

Overlapping cartons may be sealed by adhesive or by wire-stapling machines in which the anvil is thrust under the two overlapping flaps and the top stapled. Regular slotted boxes may be stapled closed by a machine utilizing a preformed staple. Two anvils having knife edges penetrate the fiberboard. The preformed staples are clinched against the anvil heads, and the anvils retract.

Federal Specifications for Containers

LLL-B-631	Boxes, fiber corrugated
LLL-B-636	Boxes, fiber, solid
NM-B 601	Boxes, wood-cleated plywood
NM-B-621	Boxes wood nailed and lock corner
NM-B-631	Boxes, wood, wirebound
RR-B-116	Barrels steel Type 5
RR-C-96	Cans steel (tinned-plate, terne-plate, and black sheet), friction covers
RR-D 701	Drums, metal Types 5A, 5C, and 5D (for acids and other dangerous articles)
RR-D-726	Drums, steel, Type 5 (for inflammable or poisonous liquids)
RR-D-729	Drums steel, Type 5B (for liquid petroleum products)
RR-D-741	Drums steel, Type 6C (for inflammable solids and oxidizing materials)
UU-S-48	Sacks, paper, shipping

SECTION 33

INDUSTRIAL GLASS

BY HAROLD G. VOGT

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INDUSTRIAL GLASS[*]

BY HAROLD G. VOGT

Physicist, Corning Glass Works

DEFINITION AND TYPES OF GLASSES

Glass is an inorganic product of fusion which has been cooled to a rigid condition without crystallizing.[†] It is typically hard and breaks with a conchoidal fracture. Although basically homogeneous and transparent, masses or bodies of glass may be made colored, transluscent, or opaque by the inclusion of dissolved, amorphous, or crystalline materials.

Glass is made by heating a mixture of dry, powdered or granular materials at about 2800°F in a refractory container until a viscous, homogeneous liquid is formed. The proportions of the raw materials in the batch are seldom in the ratios corresponding to those of chemical compounds. Rather the proportions may be varied continuously over relatively wide ranges to give glasses with some physical and chemical properties varying continuously over wide ranges, while some properties remain practically constant, and others vary over small ranges. Moreover, there is considerable choice as to which and how many ingredients are to be used. Through a study of the properties of some 50,000 glasses, it has been possible to select formulas that are best fitted for specific applications. About 400 different glasses are melted commercially each year. They differ considerably in some of their properties. Of these glasses a relatively small number have such good all-around usefulness that they are melted continuously and in great volume.

The great bulk of commercial production is commonly classified in the following chemical types of silicate glass. Listed in order of decreasing costs, decreasing softening temperatures, decreasing thermal-shock resistances, and of increasing thermal-expansion coefficients they are (1) 99.8 per cent silica glass, *i.e.*, fused quartz or fused silica; (2) 96 per cent silica glass; (3) borosilicate glass; (4) lead glass; and (5) soda-lime glass. The commercial ranges in composition of these types, as determined by analysis and calculation of oxide equivalents of the finished glasses, are given in Table 33-1, and the physical properties of some typical representatives of each type are given in Table 33-3.

Table 33-1

	Soda lime, per cent	Lead, per cent	Borosilicate, per cent	96 per cent silica, per cent	Fused quartz, per cent
SiO_2	70-75	53-67	73-82	96	99.8
Na_2O	12-18	5-10	3-10		
K_2O	0-1	1-10	0.4-1		
CaO	5-14	0-6	0-1		
PbO	....	20-40	0-10		
B_2O_3	....	.....	5-20	3	
Al_2O_3	0.5-1.5	.....	2-3		
MgO	0-4				

In other types of glasses such as colored, opal, and optical, the chemical composition is not the distinguishing characteristic, and they are usually made from the above types or "hybrids" thereof. Colored glasses are made by adding traces generally of metallic oxides to the batch, plus proper heat-treatment of the finished glass where needed. Opal glasses most often are made by adding fluorides or phosphates to the batch, and the finished glass is treated so that small particles appear in the body of otherwise transparent glass and cause the scattering light that makes opal glasses appear "milky." Optical glasses are made from a variety of compositions to get desired combinations of index of refraction and dispersive constant. Their distinguishing characteristic lies in the severe degree of freedom from physical imperfections required in the finished glasses.

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[*] Superior numbers refer to specific references listed at the end of this section.

[†] ASTM C 162-45T.

There are a few nonsilicate glasses, *e.g.*, borate and phosphate glasses, made in small quantities for specialized uses.

PHYSICAL AND CHEMICAL PROPERTIES OF GLASSES

During recent decades industrial users of glass have adopted much more of an engineering approach towards its application, and glassmakers, through intensive research, have acquired scientific understanding and technical control of the material. New superior chemical types, particularly the low-expansion borosilicate glasses and 96 per cent silica glasses, have been developed. Technological improvements have been made in the old soda-lime and lead types, and designing and finishing treatments have been advanced greatly. The resulting improvements in mechanical strength, thermal endurance, chemical durability, and electrical properties have been so marked that glass, particularly of the low-expansion types, has gained wide recognition as a versatile engineering material.

In using the specific values for the various properties of glass listed in Table 33-3, it is necessary to keep in mind the following general comments because the physical nature of glass is so different from that of other engineering materials.

Mechanical Properties. Glass is a supercooled liquid and is not crystalline or endowed with any directional properties the way crystals are. It is nonductile and perfectly elastic; *i.e.*, it shows no cold flow right up to the stresses necessary to break it. It seems to be broken only by tensile stresses, the fracture starting in almost all cases from imperfections in the surface. It does not deform or decompose with aging or exposure to moisture, and it suffers no permanent deformation when subjected to mechanical or thermal stresses even at very low or quite high temperatures. Having no porosity whatever, it does not swell or shrink with changes of humidity or immersion.

1. *Strength.* For safe design practice, tensile stresses should be limited to 1,000 psi maximum for annealed articles and to 4,000 psi for tempered articles of glass. This is true since, although annealed articles have tensile strengths ranging from 4,000 to 10,000 psi and although many products can be tempered to have two to five times the strengths of annealed products, the evaluation of safety factors is rather difficult and depends upon the type of application and condition of the surface.

The nature of the surface has a great effect upon the tensile strength of glass. Not only are loading stresses considerably magnified in regions of surface imperfections, *e.g.*, scratches, but also the stress concentrations cannot be relieved by any yielding or cold flow since glass is nonductile. This results in the strengths of glass articles being markedly less than the 500,000 psi and upwards measured for some fibers of glass. It also results in noticeable variations in the measured strengths. Tempering puts a permanent compressive stress in the surface and thus strengthens a glass article and makes the glass more resistant to abrasion or scratching.

Glass tested in compression or shear seems always to be broken by the tension components of stress, and the terms compressive strength and shear strength are generally considered to have little meaning for glass. However, the apparent or effective compressive strength is about ten times as great as the normally measured tensile strength, and consequently it is a decided advantage, when using glass in conjunction with metals or other materials, to have the designs such that the forces on the glass are compressive rather than tensile. This is particularly true when temperature changes may introduce considerable stress in the glass because of differences in expansion coefficients.

2. *Moduli of Elasticity.* Young's modulus ranges from 6.5 × 10^6 to 12 × 10^6 psi for the various glasses. Poisson's ratio ranges from 0.18 to 0.30, but 0.22 may safely be used in design calculations.

3. *Hardness.* The nonductility of glass makes it difficult to assess the significance of indentation tests, and glass manufacturers use tests different from those used for metals. On the Moh scale of resistance to scratching, where diamond is arbitrarily assigned a figure of 10, glasses range from 5 to 7.

The terms "hard" and "soft" as used in the glass trade seldom refer to the mechanical hardness but rather to the question of whether a glass has a high or a low softening temperature, respectively.

Besides the Moh test, glass manufacturers sometimes use an impact-abrasion test in which the amounts of abrasion required for equal surface destruction are measured and calculated relative to that of regular polished plate glass. Assigning a figure of 1 arbitrarily to plate glass, the values for a few other types of glass are as listed in Table 33-2.

Table 33-2. Impact-abrasion Hardness

Plate glass (soda-lime glass)	1.0
High-lead glass	0.5
Borosilicate glass	3.0
96 per cent silica glass	3.5

Thermal Properties. 1. *Reference Temperatures.* Consonant with the fact that it is a supercooled liquid, glass has no critical temperatures. It has no freezing or melting point, and with change of temperature the viscosity changes continuously. The "melting temperature" of a glass is merely a high temperature at which the batch materials will fuse together and give a viscous, homogeneous liquid in a reasonable length of time. The viscosities of glasses are about 100 poises, even at the melting temperatures.

For the practical control of shaping and annealing operations, either in the glass factory or out in the field where the glass is being used, it has been found useful to define certain reference temperatures. The *softening* point[1] of a glass is the temperature at which a uniform fiber, about 0.75 mm in diameter and 22.9 cm long, elongates under its own weight at the rate of 1 mm per min when the upper 10 cm of its length is heated in a prescribed furnace at the rate of approximately 5°C per min. For glasses of density, about 2.5 grams per cc, *i.e.*, for all the commercial chemical types listed above, except the higher percentage lead glasses, this corresponds to a viscosity of 4.5 × 10^7 poises. The *annealing point* of a glass is the temperature at which internal stresses in the glass are substantially relieved in 15 min. It corresponds to the equilibrium temperature at which the glass has a viscosity of 2.5 × 10^{13} poises and is usually the high-temperature end of the annealing range. It is determined by a loaded-fiber method of measuring viscosity. The *strain point* is the temperature at which internal stresses in the glass are substantially relieved in about 16 hr and represents generally the low-temperature end of the annealing range for articles of ordinary thickness. Under conditions specified for laboratory tests, the viscosity at the strain point is 4.0 × 10^{14} poises. However, in cooling through the strain-point temperature, the properties have already begun to lag appreciably behind the temperature, and thus the viscosity reached in annealing practice may be appreciably greater or less than 4.0 × 10^{14} poises, depending upon the speed of cooling. It is for these reasons that the *maximum operating temperature* for annealed ware, if appreciable thermal and mechanical stresses are avoided, may approach the strain-point temperature, and there will be no danger of introducing appreciable permanent strain. For tempered ware the maximum operating temperature must be lower than the strain point if the mechanical strength is to be maintained at the higher level achieved by tempering.

2. *Linear Thermal-expansion Coefficient.* Coefficients of glasses range from values equal to and down to one-thirtieth those of metals. The values tabulated are averages for the range 0 to 300°C, but they differ little from the values for the range 20 to 80°C.

3. *Thermal Conductivity.* For different glasses the conductivities range from 0.0018 to 0.0033 or (gram-calories)(cm)/(sec)(sq cm)(°C). The conductivity at 250°C of a typical borosilicate glass is 20 to 25 per cent higher than its room-temperature value of 0.0028 gram-calories/(sec)(sq cm)[(°C)/(cm)], [0.0117 watts/(cm)(°C)]. Despite a lower thermal conductivity, glass dishes bake their contents faster than shiny metal dishes since glass reflects away so much less of the incident heat radiation. And because of the surface films that develop in service, heat exchangers made of metal turn out to be not much better than those made of glass.

4. *Thermal Endurance.* The ability to withstand, without cracking or breaking, rapid or uneven changes of temperature is called heat-shock resistance or thermal endurance. No general formula for the thermal endurance of glass articles has been found since it depends on all those complicated factors which control tensile strength as well as other factors. However, there are three useful generalizations: (*a*) the smaller the coefficient of thermal expansion, the greater the thermal endurance; (*b*) the thicker the glass, the more apt it is to break from thermal shock; and (*c*) glass will withstand a considerably greater total-immersion shock when heated than when cooled since in the first instance the surface is put into compression, while in the latter the surface is put into tension. Ordinary bottles are made of high-expansion soda-lime glasses, and they will usually break if plunged from boiling water into ice water, whereas most articles made of low-expansion borosilicate glasses usually will not crack even though some parts may be up to half an inch in thickness. The 96 per cent silica glasses do not crack even though plunged into ice water while they are at a glowing red heat.

Electrical Properties. 1. *Resistivity.* The volume resistivities of commercial glasses are so high, 10^8 to 10^{18} ohm-cm at room temperature, that it is difficult to make reliable measurements, and room-temperature values are available for only a small number of glasses. Generally, they are 10^{12} ohm-cm or higher. Since the resistivity of glass decreases appreciably as the temperature increases, it is more satisfactory to measure the resistivities of annealed samples at 250 and 350°C.

2. *Dielectric Strength.* The dielectric strengths of glasses in air range from 1,000 to 8,000 volts per mil of thickness and are so high that measurements are often made under dry oil to prevent "flashover." But under these conditions the oil generally fails first and sets up point electric stresses of extremely high magnitude with the result that only a fraction of the true dielectric strength of the glass is measured. Consequently, data taken under oil should be used only when the glass insulator is to be used under dry oil.

Corrosion Resistance and Chemical Durability. One of the most outstanding properties of glass is its great stability and chemical inertness. The resistance which it offers to the attack of water, to weathering, which is due primarily to water and carbon dioxide, and to the attack of aqueous solutions of acids (except hydrofluoric), bases, and salts is the main reason why glass is so widely used for the storage of foods, drugs, and chemicals; for chemical-laboratory ware; and for chemical-plant equipment such as containers, pipe lines, and pump parts. The *chemical durability* of glass is the lasting quality, both visible and chemical, of its surface. It is evaluated frequently, after prolonged weathering or storing, in terms of chemical and physical changes in the glass surface or in terms of changes in the contents of a vessel.

The chemical durability of a glass, considered as an intrinsic function of the glass, depends upon such factors as composition, thermal history, and surface treatment.

Today the demands upon the durability are becoming more and more severe because of using higher pressures, higher temperatures, and the like in the chemical industries. To meet these demands it is advantageous to choose a glass best fitted for the specific service, since, although the commercial glasses listed in Table 33-1 all have great chemical resistance in comparison to most all other materials, some have higher durabilities than others.

Some of the main factors which may affect the rate of attack of a reagent upon a glass are temperature, pressure, rate of agitation or flow of the reagent, and the nature of the reagent itself. Different glasses respond in different ways to these factors, and no accelerated (the normal rate of attack being too slow to measure in a reasonable

Table 33-3. Physical Properties of Some Typical Glasses*

Type of glass	Specific gravity	Tensile strength (annealed samples), 10³ psi	Modulus of elasticity, 10⁶ psi	Linear thermal expansion coefficient 10⁻⁵ cm/(cm)(°C)	Softening point, °F	Annealing point, °F	Strain point, °F	Thermal conductivity, 10⁻⁴ (gram-calorie)(cm)/(sec)(sq cm)(°C)	Specific heat, gram-calorie/(gram)(°C)	Volume resistivity (annealed samples), ohm-cm	Dielectric constant at 1 megacycle	Dielectric power factor, per cent at 1 megacycle	Dielectric strength, volts/mil at 60 cps	Refractive index for sodium D line
Soda-lime.........	2.47	4-10†	9.8	0.92	1280	950	890	22‡	0.20‡	$10^{12.5}$ at 20°C $10^{6.4}$ at 250°C $10^{5.2}$ at 350°C	7.2	0.9	>1,000*	1.512
Lead............	2.85	4-10†	9.0	0.91	1160	800	745	22‡	0.20‡	$10^{9.1}$ at 250°C $10^{7.1}$ at 350°C	6.6	0.16	>1,000*	1.542
Borosilicate.......	2.23	4-10†	9.8	0.325	1510	1030	960	28	0.20‡	$10^{14.5}$ at 20°C $10^{8.2}$ at 250°C $10^{6.7}$ at 350°C	4.6	0.46	>1,000*	1.474
96 percent silica...	2.18	4-10†	9.7	0.08	2730	1670	1510	33‡	0.20‡	$10^{10.2}$ at 250°C $10^{8.5}$ at 350°C	3.8	0.05	>1,000*	1.458
Fused quartz, 99.8 per cent silica...	2.20	4-10†	12.0	0.055	3000	2080	1960	33	0.20‡	10^{18} at 20°C $10^{11.7}$ at 250°C $10^{9.5}$ at 350°C	3.8	0.02	>1,000*	1.459

* Use in conjunction with preceding general remarks.
† For safe design practice, tensile stresses should not exceed 1,000 psi for annealed, and 2,000 to 4,000 for tempered glass depending upon piece in question.
‡ Approximate values.

length of time) laboratory test or practical series of tests has been devised to give a universal number for durability or to determine beforehand the durability for all service conditions. Results of accelerated laboratory tests must be used in combination with a background of actual service experience.

For general purposes, where high corrosion resistance is required, certain commercial borosilicate glasses and 96 per cent silica glass have wide application. They have excellent resistance to neutral solutions, to acids (except hydrofluoric), and to alkalies at moderate temperatures. Their low expansions make them valuable for chemical equipment where there are severe heating conditions to contend with. Their compositions are relatively simple, and this results in the contamination, if any, of the contents being limited to a few elements. For this reason they are suited to exacting chemical-analyses usage. This is especially true of 96 per cent silica glass which contains only traces of alkali elements.

WORKING CHARACTERISTICS

Cold glass cannot be turned in a lathe, surfaced in a shaper, or drilled with a machine. In other words, it cannot be shaped with a cutting tool. The drilling of small holes, a quarter of an inch in diameter or less, by triangular pieces of hard steel or carboloy, fed with turpentine and operated in a drill press, might be considered something of an exception. Larger holes can be made with a metal cylinder that has a serrated end, operated in a drill press and fed with a mixture of water and abrasive particles.

On the other hand, glass can be ground and lapped very much as can metals, except for slower speeds and extra mounting precautions. It can be polished to much smoother surfaces than can metals.

On the other hand, also, relatively large cylinders can be cut by a variety of simple "crack-off" methods. With the smaller sizes of rod or tubing a scratch with a diamond or a score with a carbide-wheel glass cutter allows the glass to be broken off squarely by putting the scratched region in tension. With thicker walled tubing, electrically heating a Nichrome wire placed in a scratch around the tubing and putting cold water on the scratched region is a more effective way of cracking off the tubing. Heavy pieces of glass can be sawed with abrasive wheels or diamond wheels.

Crews can be quickly trained to do pipe fitting of glass piping in the field with hot-wire crack-off and high-frequency electric sealing or welding.[2,3,4] For uses where service conditions do not require the standard conical glass flanged joint to be employed throughout the lines, a simpler method of field flanging has been developed. Known as pipe beading, the method consists of forming a bead by gas heating on the end of the desired length of piping. Considerably less equipment is needed than with electric sealing. The pipe-beading kits can be bought or rented at reasonable rates.

When glass is yellow hot, it is quite moldable and can be readily blown or pressed into complex shapes. A variety of mass-production methods exists for shaping hot glass as it emerges from the melting tank. Also cold glass can be reheated and then reshaped and welded. In the multiform process powdered glass can be cold-pressed or cold-cast to give articles that, after firing, are perfectly vitreous and vacuum-tight. Multiform articles can be of very intricate shape with holes, grooves, screw threads, and the like, and with some dimensions accurate to high precision.

FACTORS INFLUENCING CHOICE OF GLASS OVER OTHER MATERIALS

For many engineering uses the relatively low tensile strength and the limitations on cold working are disadvantages of glass. However, the toughness of most glass products today is such that the term "brittleness," in the lay sense of the word, is inappropriate, and modern glass is turning out to be strong enough for a great number and variety of new industrial uses. And the ease and speed of hot forming is a great advantage.

Also, the unique and excellent combinations of physical and chemical properties are great advantages. The most outstanding of its characteristics are

1. Transparency—very high and permanent.
2. Transluscency or opaqueness can be obtained as desired.

3. Allows very versatile control over the differential transmission or blocking of infrared, ultraviolet, or undesired parts of visible light.

4. Smoothness of molded surfaces or supersmoothness of ground and polished surfaces.

5. Good hardness and resistance to scratching.

6. Dimensional stability or permanence of shape is better than that of concrete or steel (hence the use of glass for the 20-ton 200-in. mirror disk for the Mount Palomar Observatory reflecting telescope). Glass is free from creep, from elastic hysteresis, and free from liability to permanent deformation under thermal or mechanical stresses.

7. Lightness, or low specific gravity.

8. Low heat conductivity as compared with metals. Yet, as discussed above, glass is suitable for a number of heat-transfer applications.

9. Very low expansion coefficients available in borosilicates or 96 per cent silica glasses.

10. Resistance to heat—good to excellent.

11. Resistance to heat shock, especially with the low-expansion types—good to excellent.

12. Small coefficient of reflection for radiant heat (see under thermal conductivity above).

13. Electrical insulation and dielectric properties—excellent.

14. Wide range of fine permanent colors.

PRINCIPAL TYPES OF INDUSTRIAL PRODUCTS

Chemical. In food- and drug-processing industries and in chemical industries, use is made for the most part of low-expansion, chemical, and heat-resistant glasses for:

1. Containers, tanks, jars
2. Piping and tubing
3. Centrifugal-pump parts
4. Laboratory apparatus
5. Tubing for heat exchangers
6. Cylinders for fractionating columns
7. Sight and gauge glasses
8. Godet wheels or pulleys for drawing rayon thread out of acid baths

Electrical and Electronic. In the electrical and electronic industries, glass is used for the following:

1. Insulators
2. Lightning-arrester bodies for power-line use
3. Ribbon-glass condensers
4. Fuse-plug bodies
5. X-ray-tube envelopes and X-ray absorbing sheet or molded shields
6. Electronic-tube envelopes and tubing
7. Mercury-switch envelopes
8. Metallized glass for containers hermetically sealed by ordinary soldering
9. Metallized-glass electronic components
10. Special glasses for matched expansion glass-metal fusion seals

Lighting. In the lighting industry glass is employed for the products listed below.

1. Incandescent-lamp envelopes, fluorescent-lamp tubing, sign-light tubing
2. Projection-lamp envelopes and cover lenses
3. Diffusing and lens-action prismatic light controlling glassware

Light Filters and Colored Signalware. In the field of light filters and colored signalware, glass is employed in the following materials:

1. Ultraviolet transmitting
    - *a.* Germicidal-lamp tubing
    - *b.* Sun-lamp bulbs
    - *c.* Black-light tubing
    - *d.* Bulbs
    - *e.* Filter sheets

2. Heat or infrared transmitting
    a. Heat-lamp bulbs
    b. Black-light vision filters
    c. Bulbs
    d. Tubing
3. Colored glass
    a. For signalware that often has accurate prismatic surfaces on cylindrical, flat, or lens shapes
    b. For filter sheets or covers

Thermal. In thermal installations, the materials listed below are made of glass.

1. Heat-exchanger tubing
2. Furnace windows
3. Induction-furnace crucibles and calcining jars and trays
4. Thermocouple protection tubes
5. Heat-absorbing sheet glass
6. Furnace cores or linings
7. Thermos- and Dewar-flask glass parts
8. Thermometer tubing
9. Heat-insulation Foamglas
10. Heat-insulation Fiberglas products
11. Heat-insulation double-glazed (sealed-in air space) windows

Optical. In the optical field, glass is used for the following:

1. Instrument lenses, prisms, and mirrors
2. Eyeglasses and goggles

Mechanical. Glass is used in the following materials in the mechanical field:

1. Jewel bearings for precision instruments
2. Optically flat surface plates
3. Machinists' gauges
4. Torsion fibers
5. Paraboloidal reflector shells for sealed-beam headlamps and projection lamps
6. Accurate-bore tubing for gas and liquid flowmeters
7. Blueprint cylinders and sheets
8. Molds for rubber and plastics
9. Diffusion vacuum pumps
10. Lubricator and oil cups
11. Rollers for yarns and thread
12. Fiberglas filters
13. Slide-valve disks for gas-meter valves and rotary-valve seats
14. Fiberglas textiles
15. Humidifier floats
16. Multiform glass pipe-line filters
17. Centrifuge containers
18. Level vials

Acoustical. In the acoustical field, glass is used for Fiberglas products for sound absorption.

Structural. In the structural field glass is used for the following:

1. Laminated safety-glass windshields
2. Tempered sheet for doors, machine windows, fire screens, backstops
3. Building blocks
4. Fiberglas-plastics board of very high strength-to-weight ratio
5. Cases and covers
6. Ultralow-expansion spacers
7. Mirror disks for ultraprecision reflecting telescopes
8. Foamglas floats

CARE AND MAINTENANCE OF GLASS EQUIPMENT

Recommendations for Handling and Installing. It has been found that workers handling and installing glass equipment tend automatically to be sufficiently careful.

Moreover, modern industrial glassware is more rugged and tougher than the glass products of five decades ago, and consequently, if properly installed, it has been found to give long years of trouble-free service under hard, continuous operation.

Nevertheless, since the tensile strength of glass is relatively low compared with that of steel; since assembly stresses cannot be relieved by any cold flow of the glass; and since deep scratches and sharp bruises may cause excessive stress concentrations when mechanical or thermal stresses occur later on, it may be worth while to list the following recommendations, no matter how obvious and natural some of them may be.

To avoid scratches, chipping, and sharp bruises, it is well not to unpack or unwrap glass equipment until the time of installation. Thus bumping against hard objects is avoided, and dirt, which sometimes may include abrasive particles, is kept away from the glass. Gaskets or pads of soft, resilient materials should not be exposed to picking up abrasive particles. Glass piping should not be racked along with steel piping, and rack pegs should be of wood or covered with rubber or friction tape if they are of metal. Hangars for glass piping should be padded with clean, soft, resilient material.

To prevent assembly stresses from exceeding the safe maximum tensile stress of 1,000 psi, it is well to check a number of installation details. In coupling flanged glass piping, gauge glasses, sight glasses, or the like, to metal parts, high point loading of the glass should be guarded against. The metal parts should be rugged enough not to deform under assembly stresses and should be flat and smooth. Resilient gaskets of materials and designs recommended by the glass manufacturer should be used. Nut, and bolts should be drawn up evenly and only tightly enough to prevent leakage. Stud and nut clearances should be sufficient but not greater than necessary. Moreover, since glass is appreciably stronger in compression than in tension, it is well to have coupling stresses on the glass compressive as much as possible. Good alignment is desirable at all times, and tanks, pumps, or other similar equipment to which glass piping is attached should be rigidly stationary so they cannot shift and put the glass under excessive stress. Hangers should be of recommended number, position, and alignment. Vibration in itself does not harm glass at all, but when piping is connected to vibrating equipment, the adjacent hanger should be located far enough away so that the displacements due to the vibration may be absorbed in the resilient gaskets at the flanged joints.

Inspection for strains in glass is simple with a polariscope and may be worth while.

Operating Recommendations. So as not to exceed the maximum safe tensile stress of glass piping in sizes up to 4 in. ID, it is recommended that pressures not exceed 50 psi. Pressures in the line should be checked or relief valves used, particularly with positive pumps, to protect against pressure surges. Closing valves too suddenly may cause excessive "water hammer" and vibration. Sometimes hydrostatic heads are sufficient to require allowance.

The maximum operating temperature depends greatly on whether the article is made of a high-expansion soda-lime glass or of a low-expansion borosilicate glass. With equipment made of borosilicate glasses of expansion coefficients around 35×10^{-7} per deg C, temperatures up to 220°F may be used freely in almost all cases, and higher temperatures—up to about 400°F are not uncommon. With the higher temperatures there might be thermal shock problems, and the glass manufacturer should be consulted. Pyrex-brand piping, which is made of a low-expansion borosilicate glass, will easily stand flushing with alternate cycles of cold water and live steam. Actually, Pyrex-brand industrial glassware does not have its strength measurably affected until the temperature reaches about 900°F, but the glass at a temperature over 400°F can be subjected to excessive thermal shock if it comes into contact with rain, cold-air blasts, and the like. Consultation should be made first with the glass manufacturer if such high-temperature service is contemplated.

Do not bring flames into direct contact with glass since some spots in the glass may be heated hot enough to introduce undesirable permanent strains.

Wooden guards may profitably be placed around glassware in regions of heavy traffic or around glassware outdoors that might have icicles fall on it.

Do not use wire brushes in cleaning glass. Cleaning can be accomplished often by flushing with cold or hot water. Acid solutions or hot 0.6 per cent alkaline detergent solutions containing a wetting agent may be used in various cases that are more obdurate.

Large glass vessels under gas pressure or vacuum should be caged for safety.

Alteration and Repair of Glass Equipment. Generally, alteration of glass equipment or the repair of cracked or broken articles out in the factory is not very feasible. A laboratory glass blower with the requisite knowledge, skill, and facilities can reform yellow-hot glass and properly anneal it if the article is not too large or of too great a wall thickness.

Recently, two methods have been developed for making alterations in glass piping installations made of Pyrex-brand piping. In both methods a stock length of flanged piping is cut with an electrically heated wire method of crack-off. Then in the first method[2,3,4] a glass flange is sealed to the cut end of the piping by high-frequency electric sealing or welding to give a flanged pipe of desired length. After the sealing, the weld is annealed by a small electric furnace that clamps around the pipe.

In the second method of field flanging or pipe fitting in the field, the cut end of the pipe is heated in a ring gas burner; and when the glass is soft enough, a beaded end is formed on the pipe by a rapid spinning of the pipe. The beaded end is annealed by rotating it in the same ring burner. Beaded piping can be joined to standard flanged piping or to other beaded piping. The beaded joint requires the use of rubber gaskets and is not ordinarily used for service pressures over 25 psi, for temperatures over 250°F, or for piping of inside diameter over 3 in.

GLASS SPECIALTIES AND THEIR CHARACTERISTICS

Building Blocks. Glass building blocks are hollow, partially evacuated blocks of variously figured transparent glass. They are used in buildings for non-load-bearing walls and partitions that give controlled "daylighting" combined with insulation against heat and cold, and insulation against noise.

Glass blocks are made by fusing together two halves of pressed clear glass, and thus the hollow space inside is completely and permanently sealed. The air pressure in the hollow is only about one-third an atmosphere, and hence a panel of blocks provides good thermal insulation. For example, the over-all coefficient of heat transfer, U value, is 0.49 Btu/(sq ft)(hr)(°F difference) for a 15-mph wind on the exterior of a panel.

The edges of the blocks are coated with a grit-bearing plastic material to ensure permanent and effective bond with portland-cement mortar in which the blocks are set. The crushing strength of panels so set varies from 400 to 600 psi for edgewise loading. Hence panels are self-supporting but should not bear any of the building load. Panels have a lateral strength or wind resistance of 55 psf for the maximum limit of 144 sq ft of unsupported area.

Owing to the different figurings of the surfaces obtained in the pressing, different patterns of blocks range from a high transluscency to a reasonable transparency and include prism light-directing and -diffusing patterns. In the diffusing units a screen of Fiberglas is sealed inside the block.

The blocks are available in three sizes: 5.75, 7.75, and 11.75 in. square in face dimension by 3⅞ in. in over-all thickness. Special shapes for corner pieces are made in 5.75- and 7.75-in. sizes, and special radial units for curved panels in 7.75-in. size.

Foamglas. Foamglas, made by the Pittsburgh-Corning Corporation, is a rigid material consisting of inert-gas bubbles completely encapsulated by glass. More than 5,000 bubbles per cu in. are formed in the foaming or cellulating of the glass by the evolution of internal gas at a high temperature. Foamglas is used for an easily installed industrial and commercial thermal insulation both in low-temperature and high-temperature applications. Also it is as good as or better than cork as a material for floats.

It retains its insulating property permanently since it is impervious to moisture, acid (except hydrofluoric and glacial phosphoric) atmospheres, vapors, and fumes. It does not pack, swell or shrink, slip, warp, or rot; and it is verminproof. Not only is it noncombustible, but it has resistance to heat and can be used at temperatures up to 800°F if thermal shocks are avoided.

Although it is friable to the extent that it crushes locally with point loading, *e.g.*, yielding to such surface irregularities as rivet heads and welds, it does not crack or break and will support its own weight permanently in any type of wall construction. It should not be used for load-bearing walls. The friability permits cutting and shaping Foamglas with ordinary tools.

The specific values of the physical properties of Foamglas are as shown in Table 33-4.

Table 33-4

Density	10.5 lb/cu ft (0.17 grams/cc)
Thermal conductivity, K,	
at 50°F	0.40 (Btu) (in.)/(hr)(sq ft)(°F)
at 300°F	0.55 (Btu)(in.)/(hr)(sq ft)(°F)
Expansion coefficient	0.000005 (in.)/(in.)(°F)
Specific heat	0.16 to 0.19 Btu/(lb)(°F).
Crushing strength	150 psi (average)
Modulus of rupture	90 psi (average)
Absorption (24-hr immersion in water)	2 percent by weight (average, all at surfaces)
Air infiltration of permeability	0
Capillarity . . .	0
Volume change with moisture	0

Standard sizes are slabs 12 by 18 in. of thicknesses 2, 3, 4, and 5 in and standard pipe insulation forms.

Double-glazed Windows. Double-glazed windows consist of two sheets of glass with an air space between. There are various types made with different kinds of spacers and sealing techniques. Their properties are approximately the same as those of glass blocks in insulating against heat and cold, reducing condensation on interior face, and in insulating against noise.

Heat-absorbing Glass. Heat-absorbing glass is made of both high-expansion and low-expansion compositions. These glasses contain certain metallic oxides which cause high absorption of infrared rays or heat waves. They are used in filters or windows to pass visible light while at the same time stopping appreciable amounts of the heat radiation, which causes unwanted heating, unwanted glare, and unwanted effects on fabrics, dyes, etc.

The low-expansion types, which are therefore heat resistant as well as heat absorbing, are about 85 per cent as transmitting for visible sunlight as ordinary sheet glass, while they exclude about 30 per cent of the solar-heat waves.

Heat-absorbing plate glass transmits less than 45 per cent of the solar heat, and at the same time more than 70 per cent of the total solar light. It is made of high-expansion glass (linear thermal-expansion coefficient 8.3×10^{-6} per deg C) and has a thermal endurance of 125°F. It is available in $\frac{1}{8}$-, $\frac{1}{4}$-, and $\frac{3}{8}$-in. thicknesses with plates ranging up to 10 by 18 ft for the $\frac{1}{8}$-in. thickness.

Tempered Plate Glass. Subjecting plate glass, after cutting, drilling, or fabricating in any way that is desired in the finished piece, to a heat-treatment or tempering puts a permanent compressive stress into its surfaces. This makes the glass tougher without impairing its other properties. It is four to five times stronger and will flex four to five times farther than normal annealed plate glass. It does not shatter into jagged splinters but rather falls apart into small, relatively blunt and harmless chunks. It is harder and more resistant to scratching. And it can be used at temperatures between 120 and 650°F. These improvements greatly broaden the field of use for plate glass.

NOTE: The tempered glass cannot be cut or drilled after tempering.

Tempered plate glass can be used for oven glazing, fire screens, fire windows where wired glass is not permitted by law, kick plates, all-glass doors, backstops at baseball fields and the like, as well as for many other applications that need tough windows.

It is available in $\frac{3}{16}$- to $1\frac{1}{4}$-in.-thick plates up to 48 by 108 in. in clear, colored, or colored-opaque glasses.

Pyrex-brand Piping.* The glass piping is made from borosilicate glass (properties listed in preceding material), which is mechanically tough and rugged, which has high resistance to heat and to heat shock, and which has high chemical-corrosion resistance. The piping conveys liquids and gases safely, visibly, and without contamination. Its low expansion coefficient makes possible the flushing of the line with alternate cycles of cold water and live steam, hot water, or hot detergent solutions for cleaning or sterilizing. It is quite resistant to pitting, and its "glass smoothness" facilitates cleaning and minimizes scale formation.

It is resistant to all acids (except hydrofluoric) and to alkalies. Frequently it is used to handle hydrochloric acid, nitric acid, wet chlorine, bromine, chromic acid, chlorinated hydrocarbons, brine, organic acids, pharmaceuticals, sensitive drugs, and wine.

Piping is made with conical flanged ends, and with standard fittings and gaskets stocked by the glass manufacturer, it is possible to join lengths together and couple to metal flanges. It is available with 1-, 1½-, 2-, 3-, and 4-in. ID sizes in standard 10-ft lengths. Other lengths are made to the purchaser's designs, and while it is unusual to use pieces over 10 ft, it is not impractical to use lengths up to 15 ft.

Some Other Pyrex-brand Industrial Glassware of Borosilicate Glass. Tubing of the same glass used in making piping is available in a wide range of sizes for use in manufacturing laboratory equipment, tapered flowmeters, heat-exchanger tubes, and many miscellaneous pieces of plant equipment. Ranges of diameters extend from ¼ to 4½ in. in intervals of ¼ to ½ in. Wall thicknesses range from 1/16 in. for the small sizes up to ¼ in. for the heavier pieces. During the Second World War the drawing machines were equipped to make tubes up to 6½ in. OD and in lengths up to 6 ft. These larger sizes are made to order and are not ordinarily stocked.

The tubing can be of elliptical cross section, rather than circular, for special applications such as cell cooling coils.

Tube and shell heat exchangers can be equipped with thin-wall glass tubes if the end sheets are designed to hold a packing that permits the glass tubes to move slightly with respect to the shell. A number of successful heat exchangers of this type use rubber O rings.

Borosilicate hand-blown glass cylinders are made in sizes from 5 in. OD up to 23½ in. OD. Table 33-5 lists the sizes and lengths most frequently produced.

Table 33-5. Frequently Used Sizes of Borosilicate-glass Cylinders

Outside diam, in.	Nominal wall thickness, in.	Max length, in
7	1/8–1/4	24
8	1/8–1/4	16
9⅛	3/16–3/8	24
10	3/16–3/8	20
11	3/16–3/8	19
12	3/16 1/2	24
13	3/16–1/2	23
14¼	3/16–1/2	28
16	3/16–1/2	24

Large cylinders are often used to make observation sections in pilot-plant fractionating columns. Entire fractionating columns can be built with glass cylinders when chemical attack is exceptionally severe or when reactions can be accelerated by the admission of visible light and near ultraviolet

Flat-bottomed borosilicate-glass jars are available in sizes as large as 16 in. OD by 36 in. long and 18 in. OD and 18 in. long. They have been much used for the bodies of electrolytic cells because they are much less vulnerable to temperature stresses than other ceramic containers of the same size.

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* Pyrex and Vycor are registered trademarks of Corning Glass Works. Their sole function is to indicate the origin with Corning of the products on which they appear. A large number of different glass compositions, varying widely in physical and chemical properties are used in the fabrication of glassware and sold under both trade-marks. The Pyrex trade-mark is also applied to certain metal articles.

Borosilicate plate glass is available in thicknesses of 3/16, ¼, and ⅜ in., and ½-in. thicknesses are planned for. Sheets are normally smaller than 24 by 30 in., but it is possible by special order to obtain sheets as large as 24 by 60 in. Applications include low-pressure sight glasses, furnace- and oven-door glasses, and occasionally such unusual purposes as crane cab windows in steel mills. Molded sheets and slabs of borosilicate glass are made in thicknesses up to ¼ in. These heavier pieces are ground flat and polished for use as sight glasses in pressure vessels.

Molded sight glasses are made in several standard sizes ranging from 2½- to 8-in. diameter for low-pressure service and from 4- to 8⅜-in. diameter for pressures ranging between 150 psi for the former and 30 psi for the latter. When necessary, these glasses can be strengthened by tempering to double their maximum safe working pressures.

Molded borosilicate godet wheels are used in the viscose-rayon industry to draw wet fibers out of the acid baths. Glass is used for this purpose because of its corrosion resistance and its smooth abrasion-resistant surface which requires no finishing operations subsequent to molding. Intricate glass parts for a centrifugal pump are molded and then ground so that impeller, volute, head plate, and seal ring fit together in such a way that chemicals flowing through the pump do not come into any contact with metal.

Products Made of Glass Fibers. Glass in the form of fibers is a new basic engineering material. Fiberglas fibers, made by the Owens-Corning Fiberglas Corporation, are produced in a wide range of diameters and of various glass compositions fitted for specific types of application. The range of diameters is from coarse (0.010 in.) through fine textile (0.00022 in.) down to superfine (0.00009 to 0.00005 in.). The fibers are solid rods of glass and possess most of the good properties of ordinary forms of glass:

1. Noncombustibility and high resistance to heat
2. Nonporosity; hence nonhydroscopic and do not shrink, swell, and stretch
3. Chemical resistance to weathering, acids, oils, corrosive vapors
4. Nonductility
5. Inorganic in composition and hence rotproof, mildew-proof, as well as waterproof. Also immune to attack of insects, rodents, and other vermin
6. Excellent electric-insulation properties
7. Possess no odor and do not pick up odors

Because of their fineness, the fibers possess in addition such new and very desirable properties as

8. Great flexibility
9. Great tensile strengths, ranging up to 250,000 psi, and possessing greatest tensile-strength–weight ratio of any commercial material

The Fiberglas fibers are used in more than 100 Fiberglas products which are derived from four basic forms. All the forms are incombustible, nonabsorbent of moisture, not subject to rot or decay, resistant to most chemicals, and provide no sustenance for insects and vermin. The four basic forms and their properties and primary uses are as follows:

1. Wool forms
   - *a.* Light weight: 1 to 9 lb per cu ft
   - *b.* Operating-temperature range for thermal insulation use: subzero up to 1000°F
   - *c.* Thermal-insulation value: K, 0.25 to 0.30 (Btu)(in.)/(hr)(sq ft)(°F) at various densities, at 70°F mean temperature
   - *d.* Sound absorption: high noise-reduction coefficient at various densities, thicknesses, and finishes
2. Filter and aeration forms
   - *a.* Air filtration: standard 1- and 2-in. thicknesses of fibers covered with special dust-catching adhesive provide highly effective method for cleaning air of dust, lint, and pollen

  - *b.* Industrial aeration packs: semirigid packs of bonded coarse, medium, or fine fibers used to filter air of gases, liquids, sludges, etc., and in contact and eliminator applications where water is used for cooling, humidity control, or air washing. Tower packing is used in petroleum, chemical, and distillery industries for evaporation, absorption, and fractionation applications where surface area of liquids must be greatly increased or gases are to be absorbed

3. Textile forms
   - *a.* Limited stretch, nonswelling or shrinking, low expansion coefficients, and high strength-weight ratios
   - *b.* Electrical insulation: excellent insulation characteristics in cloths, tapes, cordage, braided sleevings; varnished cloths, tapes, and tubing; combination Fiberglas-and-mica sheets and tapes; laminated-Fiberglas sheets, sticks, or wedges; Fiberglas-covered wire. Such forms impregnated with silicone resins make possible great improvements in electric-motor construction—light weights, smaller motors, and ability to stand severe overloads
   - *c.* Industrial textiles: noncombustible and heat resistant
   - *d.* Reinforcement in laminates, plastics, etc.
   - *e.* Chemical filtration
4. Mat forms
   - *a.* Storage-battery retainer mats
   - *b.* Acoustical surfacing
   - *c.* Filtration
   - *d.* Reinforcement for electrical insulating plastics
   - *e.* Pipe wrapping

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SECTION 34

PLASTICS

BY GEORGE K. SCRIBNER

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PLASTICS

BY GEORGE K. SCRIBNER

President, Boonton Molding Company

INTRODUCTION

Plastics are synthetic materials put together by the ingenuity of the laboratory for the explicit purpose of assembling qualities and economies of manufacture not available in natural materials. When the modern age of plastics began with the discovery by Dr. Leo H. Baekeland in the early 1900's of a practical way of utilizing the long-known reaction between phenol and formaldehyde, plastics were first thought of as substitutes for other materials of engineering.

Plastic chemistry usually starts with so-called intermediates. Acetylene, acetone, phenol, formaldehyde, benzene, and many similar compounds are diverted, split apart, and recombined to form elementary plastics. These are still not the plastics seen and used by the consumer. They must be converted to such a stage by a combination of mechanical and chemical processes.

The plastics industry, in diagram, is somewhat similar to a tree. The trunk is made up of a large number of large chemical companies who process the basic raw materials into usable and, as yet, raw compounds. The roots literally absorb from the ground the starting materials, the carbon, hydrogen, oxygen, and nitrogen. Other constituents come from water, air, coal, oil, and miscellaneous natural compounds. All these materials pass up the trunk and are there transformed by chemical and research groups into the secondary usable plastics which are then ready for pressing, mixing, and fabrication into a multitude of products.

When these raw plastics are ready, they are usually sold by the chemical companies to specialists in paint, lacquer, safety glass, paper, novelties, general moldings, plywood, and many other industries. These specialists fabricate the article which is finally seen and used.

PLASTICS CLASSIFICATIONS

Plastics are classified in several ways. The most important division that covers the entire field is by a behavior pattern.

Thermoplastics are those plastics which soften under heat and again harden when cooled. No internal chemical change takes place. They are very like paraffin in this respect. There are roughly 14 basic materials in this group with a softening point ranging from 140 to 220°F. They will vary greatly in their mechanical and electrical qualities within the range of their optimum operating temperature. Some have excellent qualities all the way down to −60°F. Some have a sharp softening point, while some get progressively softer as the temperature approaches the deadline.

Thermosettings will soften only once under heat and then undergo an internal chemical change which makes them hard and impervious to further applications of heat up to the charring or disintegration point, which ranges from 250 to 300°F. There are only three of these plastics in current use, phenol formaldehyde, urea formaldehyde, and melamine formaldehyde, all familiarly called by their first names, phenolics, ureas, and melamines.

Plastics are again classified by their chemical composition into the following general groups:

1. *Thermosetting resins*
   a. Phenolics
   b. Ureas
   c. Melamines
2. *Thermoplastic resins*
   a. Styrene
   b. Acrylics
   c. Polyamides
   d. Polyethylene
   e. Vinylidene chloride
   f. Vinyls
3. *Cellulosics*
   a. Acetate
   b. Butyrate
   c. Propionate
   d. Ethyl
   e. Nitrate
4. *Naturals*
   a. Casein
   b. Lignin
   c. Shellac
   d. Cold-molded

COMPARISON OF PLASTICS

Each of the various plastic materials has its own outstanding field.

The most universal is molded phenolic. It may be outdone in cheapness at times by shellac and cold molded, but only at a sacrifice of strength and appearance. It is outdone in appearance by cast phenolic, urea, melamine, cellulose acetate, acrylate, and styrene.

The thermosetting resins, of which phenolics are the outstanding example, are made up of roughly half binder, the resin itself, and half filler. This allows a wide latitude of characteristics depending on the filler used. The cheapest of the group has wood flour as a filler. This is not only the simplest to make into intricate shapes but has a pleasing, smooth texture. The filler can be changed to cotton flock to get more strength and still maintain the nice surface. Chopped canvas gives more strength according to the size of the particles used. Asbestos filler gives higher resistance to heat and a better water resistance. Mica adds materially to the electrical qualities. Any choice between these materials probably means a compromise between cost and the qualities desired in the end product.

Ureas are based on a water-white resin and therefore give a depth of surface and translucency which affords them a marked advantage in any article in which eye appeal is a factor. They are more expensive to produce both because of the pastel shades in which they commonly appear and the higher cost resin used. Although their appearance can be duplicated in some of the thermoplastics, the fact that they are no longer affected by heat within the range of their operating temperatures in many cases make their use necessary.

Melamines are comparatively new to this group since their commercial development took place during the Second World War. They were popular with the armed services because of their heat resistance, low water absorption, strength, and the fact that they would not sustain an arc at high altitudes.

Cellulose acetate is employed primarily for appearance and secondarily for strength. Where ordinary room temperatures are involved and no undue mechanical strain comes on the piece, it will perform excellently. Since it is a thermoplastic, there is a certain element of cold flow. The raw material is fairly high in cost, but this can often be made up by the advantages inherent in the basic method of shaping, *viz.*, injection. The larger the piece, the less chance there is of making it at a price competitive with the cheaper materials.

Cast phenolics are among the most beautiful of the plastics. They offer a wide range of color, depth of surface, and even variegation of internal appearance. They are expensive in most shapes for the very reason that they are cast. They must be machined at the gate line, which is usually one whole surface. Cast phenolics are an exception to almost everything that has been said about plastics to this point. They are made in their raw state, a liquid.

Laminated phenolics and ureas have a broad field. Their ability to reproduce on their surface almost anything that can be printed on paper permits them to imitate wood, marble, onyx, or any other material commonly used for facing. Developments have brought out surface decorations that cannot be duplicated by any natural product, and like all plastics they are being accepted in their own right and not for imitative purposes. Their extreme strength and easy tooling in simple forms fit them into increasingly wider fields.

Acrylics have excellent optical qualities in the transparents and are outstandingly beautiful in colors. They are the most expensive of the eye-appeal thermoplastics and have a depth of surface that makes them very attractive in many applications.

Styrene is the cheapest of the thermoplastics on a cubic basis because of its comparatively low price per pound and its low specific gravity. Its surface tends to meet the eye like a sharply varnished article. It tends to be brittle, although not seriously so. Some chemicals will cause it to craze, but on the whole it has a high reputation for its chemical and water resistance. It definitely has outstanding electrical qualities at high frequencies.

Cellulose nitrate is not moldable. It is made up in sheets, rods, and tubes and must be machined to the desired form. Because of its excellent tooling qualities, this is not difficult. It is strong, but tends to become brittle with age and is flammable.

Cold-molded can be highest in heat resistance of any of the plastics if a cement-base material is used. However, the general classification also includes bitumen or asphalt binder, which is definitely a low-temperature material. The cement-base cold molded is used for such applications as electric-stove heating-coil holders; the bitumen base is indicated for inside electrical parts where appearance is of no importance. Strength is not one of its outstanding qualities.

The vinyls comprise one of the newest, best known, and most versatile families in the plastic group. Although some of them are molding compounds producing rigid pieces, it is as rubbery, often transparent, materials that the group is best known to the public. These materials usually have excellent water and chemical resistance, so that they have become valuable as replacements for rubber not only to the public but also for industry. They are important as coatings for other materials, especially textiles and paper.

Polyamides, although a large family capable of many variations in characteristics, are so far represented only by nylon. This same material prepared in granule form makes excellent moldings having great strength in small cross sections. It has the highest heat resistance of the thermoplastics and a very sudden softening curve. It is also the highest priced molding material in its raw form.

Vinylidene chloride is most widely used as monofilaments and tubing. Some molding is done, but because it is sensitive under the heat of molding, having a tendency to throw off hydrochloric acid in the molding machine, it requires special metals for heating chambers and careful handling. This characteristic does not extend to the material after molding.

Casein, a product of cow's milk, can be made only in sheets and rods from which it must be machined. Its principal use is in buttons.

Lignin is present as the binder in nature, holding together the cellulose in plants. As a plastic binder it has intrigued the chemists for years, but so far they have not been able to extract it in sufficiently uniform composition to be practical for general molding. It is widely used in wallboard and similar flat sections.

Shellac is one of the oldest plastics. It is a natural gum produced by insects and is subject to all the variations of quality and end characteristics found in natural products.

PLASTICS MATERIALS

Phenolic-base Compounds. Phenol, commonly called carbolic acid, is obtained from coal tar and is produced in the form of white crystals. It readily liquefies when heated or may be obtained in liquid form produced by the slight addition of water.

Formaldehyde is made by passing heated wood alcohol mixed with air over heated copper or platinum, causing partial oxidation. The resulting product is a gas, and for commercial purposes it is dissolved in water. This solution is commonly called Formalin.

Under carefully controlled conditions these two materials are blended to form a resin which has the valuable characteristic of softening under the application of moderate heat and very shortly hardening into an infusible mass. This characteristic is called "heat irreversibility."

This resin is combined in varying percentages with many different fillers such as wood flour, asbestos, fabric, graphite, paper, and many others so as to obtain the added value given by the distinctive properties of these materials: better molding qualities, greater toughness and strength, and more resistance to heat.

The chemical reaction, known as the "cure," requires from 45 sec to 12 min depending on the particular class of compound used and the shape and size of the piece in question. Practically all modern phenolics are removed from the mold without reduction of temperature, although in some cases cooling is used in an attempt to freeze an exceptionally smooth and lustrous surface.

These materials burn very reluctantly. Age has no effect, but sunlight will discolor the lighter shades. Weak acids, reducing acids, and organic acids have little or no effect. Strong oxidizing acids decompose them.

Weak alkalies have a slight effect, while strong alkalies decompose them. Organic solvents have no effect when bleedproof materials are used. Organic solvents do not affect moldable transparents and cast phenolics.

Phenolic molding compounds are generally classified by their filler as follows:

1. Cellulose filled, general purpose
2. Flock filled, medium impact
3. Chopped-fabric filled, impact
4. Asbestos filled, heat and water resistance
5. Resin filled, moldable transparent
6. Special purpose

Industrial Resins. At the present time various industrial resins are finding wide use in the manufacture of resinoid grinding wheels, brake linings, thermal-insulation bolts, electrical and structural laminates, plywood surfacing papers, lamp-basing cements, plywood adhesives, sealants, and miscellaneous adhesives.

Phenol-furfural Resins. Most of the progress in the use of phenol-furfural resins and molding compounds has been made in transfer, injection, and low-pressure molding of these materials, whose set time and flow characteristics are well suited for these forming processes.

It has recently been shown that by adding of resins made with furfural and amines, phenol, etc., to wet aggregate before it is mixed with bituminous road surfacing reduces the tendency of the aggregate to strip. The same property makes it suitable for roofs, airplane runways, and other load-bearing surfaces. This product is suitable for combination with soil as well as aggregate.

The main source of furfural is waste farm products such as oat and rice hulls, corncobs, and cottonseed hulls, all of which are rich in pentosans. These raw materials are digested with an acid, such as sulfuric acid, to yield furfural.

Furan Resins. The outstanding characteristics of furan resins to date appear to be their penetrating ability as liquid-impregnating compounds, their chemical resistance as surfacing agents, and their bonding strength as adhesives. Furan derivatives form thermosetting plastics which resemble phenolics in physical properties but which bear no chemical resemblance to this group of compounds.

The most widely used raw materials in the manufacture of furan resins are furfuraldehyde and furfuryl alcohol. The furan ring has long been recognized as capable of self-polymerization under the influence of strong acid catalysts.

Furan-resin adhesives have now been established successfully in several important lines. Furan-impregnated plasters offer superior strength and durability, which were unheard of in gypsum plastics before their treatment with resins. Surface coatings of furan-resin derivatives are now being manufactured by several different companies. Liquid furan resins are also available for casting purposes.

Some trade names for furan resins are Plaspreg, Resin X compounds, Duralon, and Furetone.

Aniline-formaldehyde Resins. Resins made with diphenylamine serve a definite industrial need. They are fusible but cure rapidly at 265 to 320°F under pressure to form hard products of high softening point.

Aniline-formaldehyde resins can with proper care be machined; the electrical properties are little changed by plasticization, and the cold flow of the unplasticized grade is very slight. Ultraviolet rays and weather do not affect this material. It has a low power factor, dielectric constant, and loss factor, all of which are remarkably stable over an extremely wide range of frequencies and in the face of the most adverse moisture conditions. Its dielectric strength is high.

It has noteworthy resistance to alkalies and is not soluble in any of the common organic solvents. Its resistance to gasoline, aromatic hydrocarbons, acetone, and alcohol is excellent.

Aniline-formaldehyde resins are available in impregnating varnishes, sheets, and rods.

Resorcinol-formaldehyde resins are outstanding for use in gluing exterior plywood and for assembly work requiring high water resistance. They are particularly suitable for the assembly of boats and sections for prefabricated houses. They are appropriate for bonding wood, leather, plastics, and a variety of materials to other surfaces.

Mass production of fine wood assemblies has been facilitated and rendered less expensive through the use of a special grade of resorcinol resin. As many as six operations can be eliminated from the assembly of a molded-plywood radio cabinet in which the resin has been employed. Moreover, the assembly is resistant to warping and delamination.

The most important characteristic of resorcinol resins is their ability to react at temperatures of 75 to 90°F within controlled time periods of several minutes to several hours. Also characteristic of these resins is their excellent compatability with many other plastics and elastomers. In general, improved water and heat resistance result from such combinations.

Resorcinol resins and copolymer resorcinol-phenol resins are available as solids and in various solvents. The solid resins are generally two stage, requiring an added setting agent before use.

Resorcinol glues are the only low-temperature glues providing maximum resistance to exposure. Because of their high affinity for wood, the most important application of these resin adhesives has been in wood gluing, both in laminating and molding. Some trade names for these resins are Amberlite, Bakelite Resorcinol-formaldehyde, Cascophen, Durez, Durite, Lauxite, Penacolite, Phenac, Plyophen, and Taccalin.

Phenolic Sisal. In phenolic sisal long sisal fibers are worked into a fluffy batting which in turn is mechanically reduced to a firm feltlike blanket. This blanket is impregnated with resin, and powdered resin is sometimes vibrated into its interstices.

Alkyd Resins. Alkyd resins comprise a useful class of synthetic polymeric materials generally defined as reaction products of polybasic acids or anhydrides and polyhydric alcohols.

Alkyds based on maleic anhydride, when used with some of the newer types of varnish oils, produce varnishes which have outstanding characteristics of durability and high resistance to alkalies. They also dry rapidly. According to end use, the resins are generally divided into two categories—the varnish type and the lacquer type. The varnish type is widely used as a vehicle for metal decorating, household utility varnishes, and architectural white-enamel vehicles. The lacquer-type maleic resins are usually dissolved in solvents and used with cellulose nitrate in the production of lacquers.

Alkyds based on sebacic acid and other high-molecular-weight aliphatic polybasic acids have been used with polyvinyl chloride and its copolymers. They are used in fabric coatings of all types. Typical examples are raincoat, upholstery, and bookcloth fabrics. They are also used in rubber and cable lacquers of all kinds.

Trade names for the phthalic alkyd resins are Amberlac, Aroplaz, Beckosol, Dulux, Duraplex, Dyal, Esterol, Falkyd, Glyptal, Mirasol, Rezyl, and Syntex; for the maleic alkyd resins, Amberol, Arochem, Beckasite, Carbic and C-9, Dulux, Dymal, Lewisol, Paranol, Pentalyn, Petrex, and Teglac; for sebacic and others, Paraplex.

Polyester Resins. The polyester resins constitute a large and useful family of resins based on acids, alcohols or glycols, and unsaturated hydrocarbons. The combinations and permutations seem to be endless, and most of them have very valuable qualities. They are available in liquid or semisolid resin form, cast sheets, rods, and tubes. The outstanding qualities offered are chemical resistance, dielectric properties, low water absorption, low-temperature cure, toughness, low-pressure or contact cure.

Trade names of polyester resins are Bakelite Polyester Resins, Kriston, Laminac, MR-Resins, Paraplex, Plaskon, Selectron, Thalid, and Vibrin.

Phenolic-base Compounds—Laminated. The fundamental difference between laminated and molded phenolics is in the nature of the filler used. The resin is basically the same, although in the molded group, it has been developed along the lines of speed of cure, a characteristic which is not so important in the laminated industry. Molded phenolics are powders or comparatively small pieces impregnated with resin. Laminated uses sheets of paper, linen, canvas, and silk, impregnated and dried in a continuous operation, so the original interlacing and length of fiber are not destroyed. The trade has standardized on grades which, by variation of the filler and treatment, have different qualities. Like molded phenolics, if one characteristic is to be emphasized, others may have to be minimized. The strength, flexibility, and imperviousness to oil and moisture are the outstanding qualities that suit this material for its wide use as silent gears. These are hobbed from molded or cut blanks. Its high dielectric strength, high insulation resistance, and great strength in thin sections recommend it for simple punchings. It is widely used in the plating industry in place of wood for barrels and carriers. The rayon industry uses it not only because it resists alkaline reagents but because of its lightness and strength.

Manufacturers of laminated material of this type affiliated with the National Electrical Manufacturers' Association have adopted a common nomenclature in describing their sheets as follows:

Grade X—for general use. It has a paper base furnished in natural, chocolate brown, and black with natural core. Thicknesses are from 0.015 in. and up. It punches up to 1/32 in. cold and to greater thicknesses when heated and machines readily.

Grade XX—for electrical applications requiring low moisture absorption. It has a paper base in natural or black. Thicknesses are from 0.015 in. up. It has low moisture absorption.

Grade XXX—for extremely low moisture absorption and high dielectric strength. It has a paper base, natural or black. Thicknesses are from 1/32 in. up.

Grade XP—for punching operations and electrical applications. It has a paper base and comes in natural, or chocolate brown with natural core, black with natural core, black with black core, and chocolate brown with shiny or satin finish. It punches and shears cold up to 3/32 in. and in thicker sections depending on the design of the die and the temperature of the material.

Grade C—for exceptional structural strength. It has a canvas base and comes in natural or black. It punches readily and makes highly resilient insulation. Its structural qualities recommend it where high tensile and transverse strength are required in connection with good insulating properties.

Grade L—for fine machining. It has a linen base in natural or black. Thicknesses are from 0.015 in. up. It is not usually required over 1/8 in.

Different manufacturers produce minor variations of the above general classifications to suit their individual customers' requirements and in some cases have prefixed or suffixed the above symbols with symbols of their own. The price varies with the resin content although not in direct ratio.

Tubes are produced in two grades, wrapped and molded.

The essential difference between wrapped and molded is principally in their physical characteristics. Molded is naturally more dense and less water-absorbent. The wrapped tube is, however, the more resilient. Also, the wrapped tube, properly made, is a more reliable dielectric for high voltage.

Postforming of Laminated Phenolics. Laminated phenolic was for many years believed to be virtually inert at temperatures below decomposition. Recently there has been developed a line of hot-forming laminated plastics which, when heated rapidly to a temperature of about 275°F, pass through a stage of considerable flexibility and, when cooled to ordinary temperatures, recover their average physical properties.

Phenolic-base Compounds—Cast. The ingredients of cast phenolic-base compounds, of which there are several well-known makes, are essentially the same as the phenol-formaldehyde molding resins. The difference is that they do not contain any filler and are cast in lead molds instead of being molded under pressure in steel dies. Castings can be purchased in stock forms such as sheets, rods, and tubes. Special castings such as cutlery handles, gear-shift knobs, handles of various types and designs, clock cases, and any item that does not require a split mold can be made. Sheets are of $\frac{3}{32}$ in. minimum thickness. The material in its finished form is nonflammable, odorless, tasteless, and possesses high dielectric strength. Sanding is done on equipment like that used for wood, and a standard buffing wheel is used for polishing.

These resins may be bent or formed to shape by immersing them in hot oil at 250°F for a short time, varying with the thickness, then placing them in a form and cooling. They may be stamped, embossed, engraved, or carved. They can be heated and shrunk on metal shafts such as cutlery handles on metal tangs. They can be machined to close tolerances, the shrinkage and expansion being so minute as to be negligible. Age slightly increases the hardness of the material and improves the electrical properties. Sunlight yellows and fades nonfast colors. Weak acids have no effect, but strong acids slowly attack it. The material can be washed with soap or soap powders. It slowly softens when immersed in alkaline solutions, showing slight discoloration. Strong alkalies will attack it slowly. It is insoluble in alcohols but shows slight swelling in ethyl alcohol after long immersion. It is unaffected by ketones, esters, hydrocarbons, mineral, animal, or vegetable oils; insoluble in water, organic solvents, and acids; slightly soluble in alkalies.

This material is probably the most beautiful of all the plastics when proper finish is applied. It has a luster and depth that cannot be approached, and its fine-flowing qualities as it is poured into the mold give effects very closely resembling minerals and precious stones. It is available in translucent, transparent, or opaque forms. The very definite limits on the application of this material come from the fact that it is cast only roughly and not pressed. The comparison is similar to that of a cast-iron part with a pressed-steel piece.

Urea-base Compounds—Molded. Urea molding compounds are well established in the field of commercial plastics molding and offer certain outstanding advantages. They are thermosetting: *i.e.*, while being molded under heat and pressure, a chemical reaction occurs which makes the finished molded piece resistant to further applications of heat.

The chief appeal of these moldings lies in the beautiful color effects obtainable, and in addition, the finished moldings are odorless and tasteless. Unlike the colored phenolic compounds, the molded urea resins can be made with a considerable degree of translucency giving a brightness and depth of color somewhat similar, although superior, to opal glass.

The urea resins are much more expensive than the ordinary black phenolic compositions. There is an additional cost during the molding operation owing to the great care that must be taken in keeping the moldings free from foreign matter. Such small defects do not appear on the surface of black and darker colored moldings.

Colors. Urea is produced in all colors from translucent colorless to natural and white and lustrous black. There is a list of standard colors, however, which carries a price somewhat lower than specials. It is better to use one of the standards wherever possible, although any shades can be matched. This price differential applies only to quantities less than 3,000 lb. Some tints, such as orchid and pink, are very difficult for the molder to control.

Light has no effect on red, natural, gray, green, blue, and black urea compounds. It has a slight effect on orange, yellow, ivory, and coral. Pink and orchid fade badly on exposure to light. These effects are subject to variation between the light and dark shades of each color.

Water Absorption. Molding of the urea-resin type shows a certain amount of water absorption. Nevertheless, these resins will withstand a great deal of service in contact with water without deterioration. The leading manufacturers claim that they will withstand short immersion in boiling water without deleterious effect.

Heat. Urea-formaldehyde moldings are noninflammable but char at about 200°C. They are not, therefore, suitable for such articles as ash trays, etc. Tests show that temperatures from −10 to +175°F have no effect, but higher temperatures over prolonged periods will cause fading and eventual blistering.

The intermediate water-soluble products are starting materials for the production of resin adhesives, surface coatings, paper conditioners, and other special items, all of which are finally cured through heat and catalysts. The resin is currently being applied in the production of toweling, blueprint, tag stocks, map, and chart papers, bag and wrapping papers, glassine and greaseproof papers, tissue papers, and manual and book papers.

The resin may be used as the principal bonding materials for laminated sheets. In laminates which are bonded by other materials, urea resins are sometimes employed only on surface laminae for color effects. Urea resin is also used as a binder in the making of sand foundry cores. As an adhesive it has been found suitable for edge gluing of lumber cores and various wood articles. One resin has been developed for use in conjunction with starch as an adhesive in the manufacture of fiber containers.

Urea-base compounds were approved by the Underwriters' Laboratories, Inc., as "acceptable for use in electrical devices and for the mounting of current-carrying parts," Mar. 28, 1932.

Melamine-base Compounds—Molded. Melamine (introduced in this country in 1939) is one of the most recent additions to the plastics industry. It bears a close relation to the compound calcium cyanamide and is produced from that compound by the initial production of dicyandiamide. Melamine is considered as a trimer of cyanamide CH_2N_2 of which dicyandiamide is the dimer.

Melamine, in combination with formaldehyde as a resin, may be combined with various fillers to form molding compounds. In combination with an alpha-cellulose filler the melamine molding compound has much the appearance of urea-formaldehyde compounds but has superior heat, water, and chemical resistance. It is odorless and tasteless. In combination with a mineral filler such as asbestos, it has the highest arc resistance of any synthetic molding compound and has high dielectric strength, dimensional stability, and low moisture absorption. In combination with chopped cotton rags, the melamine resin has high flexural strength, will absorb considerable shock, has a fairly good arc resistance, but will not support combustion. When combined with urea formaldehyde, melamine provides an adhesive producing a bond which withstands boiling water. These resins can be used in the production of laminated plastics at pressures as low as 30 psi. Numerous special applications of the resin include beater treatment of paper for wet strength, nonshrink treatment of cottons and woolens, tanning of leather, and a variety of coatings.

Styrenes. Although styrenes have been commercially available in the United States since 1937, it is only in the last few years that polystyrene has become widely popular. Its electrical properties remain nearly constant over a wide range of frequencies, temperatures, and humidities. Chemical resistance, low water absorption, optical clarity, and dimensional stability are other outstanding qualities. At the present time it is the most available and lowest priced of the molding materials, owing mainly to the enormous monomer capacity built during the Second World War for synthetic-rubber production. It is commercially available in compression and injection molding compounds, extruded shapes such as rods from fibers to 6-in.-diameter slugs, sheets from 1/1,000 in. thickness to several inches, and tubing from small coaxial sizes to 5-in. diameters. In sunlight styrene tends to yellow slightly. Its machinability is not good. Its water absorption is so infinitesimal that its insulating qualities are not affected. Surface hardness, permanence of dimensions, and excellent mechanical properties are other outstanding characteristics. For short periods acetone has no visible effect, but for longer contacts the styrene will soften and swell. If acetone is to be used around this material, it would be better to check the results. Any percentage of aromatic hydrocarbons will cause crazing, no matter how small that percentage may seem to be. There is no dropping off of impact strength when parts molded of styrene are placed in contact with dry ice.

Chemical Resistance. Styrene molding compounds have good resistance to acids, alkalies, and salts.

They are soluble in aromatic and chlorinated hydrocarbons, esters, some ethers, and some terpenes.

They are softened or attacked by higher alcohols, gasoline, ether alcohols, ketones, glacial acetic acid, strong oxidizing acids, chlorine and bromine water, some unsaturated hydrocarbons, and essential oils.

They are resistant to lower alcohols (methyl, ethyl, propyl), mineral oils, vegetable oils, glycols, and water.

Mixtures of two or more chemicals may affect styrenes differently than would one of them alone. Action is hastened by increase in temperature, retarded by decrease. Specifically, these compounds are soluble in styrene, benzene, toluene, aromatic and chlorinated hydrocarbons, ethyl benzene, ethyl acetate, turpentine, dioxane, gasoline, ethylene chloride, cellosolve acetate, carbon tetrachloride, paradichlorbenzene, and amyl acetate.

They are unaffected by acetic acid, formic acid, butanol, octyl alcohol, salt water, most fats, sodium hydroxide (30 per cent), potassium hydroxide (30 per cent), ethylene diamine, writing inks, bleaching solutions, photographic chemicals, animal and vegetable oils and most lubricating oils, strong or weak acids, and strong or weak alkalies.

Styrene Variations. Styrene seems to lend itself to more internal variations than any of the other plastics. Chemists have brought out many combinations in the last few years that may have quite an impact on the market in the not-far-distant future. The material has such basically good qualities that it appears to be an excellent starting point for hybrids. A few of these are briefly described below.

Modified Polystyrene. Addition of chlorinated diphenyl materials to polystyrene results in a compound with nearly equivalent electrical properties with about a 10° higher operating temperature. It is noninflammable and has excellent machining characteristics.

Polydichlorstyrene. The heat-distortion point is raised to 230°F without raising the high-frequencies electric-power losses above those of regular styrene. It is transparent with a slightly yellow color. Water absorption is lower than polystyrene. High cost restricts its use at the present time. The outstanding qualities of polydichlorstyrene are therefore high temperature resistance, surface hardness, dimensional stability at high temperatures, and low water absorption.

Cross-linked Styrene Copolymers. These materials are of a thermosetting nature, and although they have distortion temperatures ranging from 217 to 235°F, they will withstand temperatures over 300° for short periods of time without ill effects. Available forms are limited to cast rods and sheets, from which parts can be machined to close tolerances. These materials are priced somewhat higher than regular polystyrenes but considerably lower than polydichlorstyrenes.

Styrene Elastomers. Styrene elastomers are a group of flexible, rubberlike styrene elastomer resins (trade name Styralloy) developed during the Second World War for cable sheathing for low-power high-frequency losses yet having good durability and flexibility under wide temperature ranges. They are flexible in thin sections and fairly rigid in thick sections. Power losses are higher than other styrenes yet are low enough to be classified with them. They have excellent abrasion resistance and remain strong and flexible during long exposures to temperatures from −90 to +212°F.

Other properties are excellent resistance to corona attack at the reduced pressures of high altitudes, stableness to oxygen and ozone, and ease of fabrication. They are usually dark-blue–gray colored and are available in production quantities in the low-cost range.

Methyl-methacrylate Resins. Publicity given to the commercial production of acrylic resins such as Plexiglas and Lucite is but the climax to a long history of chemical research. These resins, to the chemist, are polymers of the acrylic acid derivatives; to the layman they are beautifully transparent, water-white, optically perfect rods, sheets, and molded parts.

Methyl methacrylate is a thermoplastic, and the fabricator handles it in quite the same way as cellulose acetate. It is somewhat more troublesome, not only because so much is expected of it in the way of crystal clarity but because it hardens more slowly and therefore takes a longer cycle. In time there will be colors to rival cast phenolic resins. It is somewhat more expensive than cellulose acetate.

Dimensional changes in the nature of 0.002 to 0.005 in. per lin in. for at least 30 days after molding must be expected. The rate of change tapers off after that and probably disappears in 60 days. Cast forms are very stable; they are completely polymerized and have no plasticizer to evaporate.

Chemical Resistance. Active solvents for the methyl methacrylate compounds are acetone, butyl acetate, dioxane, ethyl acetate, and methyl acetate.

Slow solvents for these compounds are methyl cellosolve, benzene, toluene, carbon tetrachloride, diethyl phthalate, ketone, aromatic hydrocarbons, ether, and chloroform.

These compounds are attacked by alcohol—95 per cent—and pure methanol.

Under ordinary service conditions these compounds are not attacked by dilute alcohol, alcoholic beverages, gasoline, heptane, kerosene, mineral oils, water, vegetable oils, fatty oils, fruit acids, dry-cleaner solvent, hexane, pentane, aqueous solutions of neutral salts, dilute aqueous solutions of alkalies, hydrochloric acid, and 50 per cent aqueous sulfuric acid.

In general, the methyl methacrylate compounds are resistant to esters, ketones, alkalies, alcohols, commercial solvents, and weak acids. They are not resistant to phenol, formic acid, or mineral acids.

Polyamides. Polyamides are the well-known nylons. A whole family of these interesting plastics exists, including materials suitable for extruding into filaments, molding powders, and some resins for coatings. The greatest volume of the material now being produced goes into the textile filaments, but the molding powders are showing a steady gain. Extruded tubes and filaments for purposes other than textiles are also important industrially.

The nylon filament was the first synthetic competitor of silk that closely resembled silk chemically. It best earned the designation "artificial silk," so generally bestowed upon the rayons and their successors. Nylon is produced by the reaction of polybasic organic acids (usually dibasic) and polyfunctional amines (usually diamines), so controlled that long-chain polymers predominate. As much as possible, cross linkages are avoided. A variety of resins can be produced by varying the acid or amine used, or the proportions of each, or the conditions of polymerization. The result is a chemical compound in many ways similar to the natural proteins in its composition. It is slow burning, very tough, with good tensile strength, takes colors with difficulty, tends to absorb moisture to a limited degree, and has rather poor electrical properties. In molded form it has fair dimensional stability but somewhat lower tensile strength than in filaments. It is still rather expensive. Aside from its use as a textile material, the filament goes into bristles for brushes, surgical sutures, woven screens, cordage, etc. Molded or extruded nylon serves as gaskets, automobile-wire insulation, nonbreakable tableware, slide fasteners, combs, etc.

Polyethylene. Polyethylene was first developed in England and firstused both in England and the United States as dielectric material for flexible coaxial cables. Its power factor and dielectric constant remain nearly constant over almost the complete electrical range. Its volume and surface resistivity, arc resistance, and dielectric-strength properties are excellent.

In appearance, polyethylene is milky white, translucent, and waxlike. It has good flexibility in thin sheets, while in thick sections it is quite stiff and rigid. Its distortion temperature under 66 psi load is 122°F, its softening range between 220 and 240°F, and it has been successfully used under low loads at temperatures considerably above 122°. Standard water-absorption figures are between 0 and 0.01 per cent for 24-hr immersion. This material resists the action of all acids, alkalies, inorganic chemicals, and has no known solvents at room temperature. Three months' immersion will only produce slight swelling by such active solvents as ethyl acetate, methyl isobutyl ketone, carbon tetrachloride, and trichlorobenzene. At temperatures above 60°C, it will dissolve in a number of common solvents. Polyethylene is commercially available in a wide variety of forms and is low priced. Its outstanding qualities are chemical resistance, low water absorption, low-temperature qualities, low specific gravity.

There are no trade names for polyethylene, but it is also known as polythene. It is produced by Union Carbide and Du Pont.

Vinylidene Chloride. Vinylidene chloride resins are odorless, tasteless, and nontoxic. When molded under proper conditions, they produce articles of great strength, abrasion resistance, strong welds, dimensional stability, toughness, and durability. The material is especially suited for the injection at high speed of parts with heavy cross section. Molded products can be readily machined, drilled, sawed, and buffed.

Fittings and parts molded of vinylidene chloride resins are particularly valuable in industries involving the use of chemicals. Pipes of this material, for instance, are superior to iron pipes for disposal of waste acids in phenolic plants. Extruded tubing fitted with injection-molded flare-type fittings of the same resin makes possible all-plastic systems of like chemical resistance which are relatively leakproof because of similar thermal-expansion characteristics. Another valuable property of the tubing is its ability to withstand pressure at low temperature. Films produced from vinylidene chloride resins exhibit an extremely low water-vapor-transmission rate, flexibility over a wide range of temperatures, and heat sealability. They are particularly suitable for various types of packaging, including food products, medicinals, and metal parts.

One year of exposure at room temperature has shown the specific chemical resistance to be

Excellent to 5 per cent sulfuric acid; 10, 20, and 60 per cent sulfuric acid; 10 per cent nitric acid; 10 and 35 per cent hydrochloric acid; 10 per cent and glacial acetic acid; concentrated oleic acid, ethyl alcohol; ethyl gasoline; turpentine; lubricating oil; linseed oil; bleaching solution; 10 per cent duponal; 10 per cent zinc hydrosulfite; 15 per cent calcium chloride; water; and air

Good to 10 and 50 per cent sodium hydroxide; 65 per cent nitric acid; triethanolamine; carbon tetrachloride; and 15 per cent ferrous sulfate.

Fair to 98 per cent sulfuric acid; ethyl acetate; acetone; methyl isobutyl ketone; and benzene

Poor to ethylene dichloride; diethyl ether; 0-dichlorbenzene; bromine water; and chlorine water

Unsuited to 10 and 20 per cent ammonia and diozane

NOTE: Resistance to corrosive liquids and to organic solvents decreases with increasing temperature.

There are no trade names for the vinylidene chloride plastics. They are produced only by Dow Chemical Company and are known generally as Saran.

Vinyl Polymers and Copolymers. Just beginning to achieve commercial recognition at the outbreak of the Second World War, the vinyl polymers and copolymers have grown to be the most used of all in terms of value of output. Not only is their dollar value the greatest, but their volume is two and a half times their nearest competitor.

Although the vinyls differ in composition and properties, certain general characteristics are common to nearly all these materials. Most materials based on vinyls are inherently thermoplastic and heat-sealable. Exceptions to this are the products which have been purposely compounded with thermosetting materials or cross linking agents. In general, vinyls can be plasticized to give a wide range of hardness—from thin flexible free films to rigid molded pieces. Most of the vinyls are naturally clear materials, and unlimited color range is available for most forms of the materials. They may be made opaque as well as clear.

In their elastomeric form, the vinyls generally exhibit properties which are superior to those of natural rubber in flex life, resistance to acids, alcohols, sunlight, wear, and aging. They are nontoxic, tasteless, and odorless and are suitable for use as packaging materials which come in contact with foods or drugs, as well as for decorative packaging requiring ordinary protection. Vinyl resins can be used in printing inks. They can be effectively used in coating paper, leather, wood, and in some cases plastics. Vinyl products in most forms can be printed on.

Insulating characteristics of the vinyls adapt them for extruded jacketings for wire and cable. Other applications have been phonograph records, photographic films, and various rubberlike objects such as grommets, gaskets, diaphragms, and similar items. Films and sheeting of various degrees of rigidity and coated fabrics have been used for shoes, handbags, luggage, and the like. Coatings for fabrics have been one of the fields in which vinyls have excelled. Protective clothing and foul-weather gear have proved the efficacy of vinyls along these lines. Vinyl coatings and vinyl films and sheeting have been used in hospital equipment because of their durability and their hygienic qualities.

The vinyls have a good finish which may be smooth and satiny or may simulate leather of all kinds. They may be colored to fit into modern schemes of decoration, can be handled easily in making up complicated items; will flex around corners well, and can be stitched or heat-sealed. They are for the most part waterproof, self-extinguishing, stain resistant, stable to light, and dimensionally stable. Vinyls lend themselves to almost all conventional methods of fabrication.

Polyvinyl Chloride. Polyvinyl chloride resins exhibit a high degree of resistance to concentrated acids, alkalies, and alcohols. They are nontoxic, odorless, and tasteless. Major applications are in elastomeric forms, or in coatings. Trade names for these resins are Geon, Vinylite, Chemaco, Tygon, and Marvinol.

Polyvinyl Acetate. Polyvinyl acetate is an odorless, tasteless, nontoxic, slow-burning, lightweight, colorless thermoplastic which has reasonably low water absorption. The material is soluble in organic ketones, esters, chlorinated hydrocarbons, aromatic hydrocarbons, and alcohols. It is insoluble in water, aliphatic hydrocarbons, fats, and waxes. Water emulsions have extended the use of this resin. They are used extensively as adhesives for bonding cloth, paper, porcelain, metal, mica, wood, glass, and plastic sheets and films. Other applications for the resins include coatings for paper, sizing for textiles, and finishes for leathers; bases for inks and lacquers; stiffening agents; heat-sealing films; and photoflash-bulb linings. Trade names are Elvacet, Vinylite, Gelva, and Vinyseal.

Polyvinyl Chloride-acetate. The copolymerization of vinyl chloride and vinyl acetate forms resins of marked versatility, combining the best characteristics of both as to physical and chemical properties and general processability. They show extreme chemical inertness, being unaffected by alkalies, oxidizing agents, and most inorganic acids; excellent resistance to water; and good dielectric properties.

Rigid sheeting may be produced in all colors, transparent, translucent, or opaque. Because these sheets do not warp or shrink, they are suitable as watch crystals, drawing instruments, slide rules and other calculating instruments, instrument dials, and aircraft windows. Problems in the printing industry have been eased by the use of molded sheeting in the production of electrotypes. The high-fidelity nonfrangible transcription record successfully established in radio studios is molded of vinyl acetate-chloride. Electrical insulation, hose and tubing for beverages or chemicals, weather stripping, and channels are produced by extrusion.

Extruded monofilament is suitable for weaving fabrics of great chemical resistance and durability. Textiles woven from these resins resist shrinkages due to water. Upholstery fabrics, fireproof curtain material, cloth for raincoats, shoes and textile-printing screens are produced from these fibers. Trade names are Vinylite, Chemaco, and Tygon.

The copolymers of vinyl chloride and vinyl acetate have excellent resistance to weak acids, strong mineral acids, weak and strong alkalies, alcohols, aliphatic hydrocarbons, and mineral, animal, and vegetable oils. They swell in aromatic hydrocarbons and dissolve in ketones and esters. They are marketed under the trade names Tygon and Vinylite.

Polyvinyl Alcohol. Major applications of polyvinyl alcohol resins are in elastomeric products, adhesives, films, and finishes. Extruded hose and tubing are excellent for use when subjected to contact with oils and other chemicals. It is used as a size in the manufacture of nylon hosiery. Trade names are Elvanol and Resistoflex Compars.

Polyvinyl Formal. These resins are soluble in chlorinated hydrocarbons, dioxan, acetic acid, and mixtures of polar and nonpolar solvents but are insoluble in alcohol, acetone, benzene, toluene, and the like. Although major applications are in the field of lacquers, coatings, and impregnations, they are also employed as mold compounds. Trade names are Formvar and Formex Magnet Wire.

Polyvinyl Acetal. These resins are soluble in lacquer solvents and aromatic hydrocarbons but insoluble in aliphatic hydrocarbons. They are most suitable for application in lacquers, adhesives, coatings and impregnations, and as films. In molded form, however, they are available in sheets, rods, and tubes.

Polyvinyl Butyral. These resins are soluble in esters, ketones, alcohols, chlorinated hydrocarbons; insoluble in aliphatic hydrocarbons; and stable in dilute alkalies; but they slowly decompose in dilute acids. They are widely used as safety-glass interlayers and between sheets of methyl methacrylate to protect enclosures of pressurized cabins of airplanes against shattering. They are also used as coatings for textiles and paper and as adhesives. Trade names are Butacite, Butvar, Monsanto Polyvinyl Butyral, Saflex, Vinylite, and Horco X.

Vinyl-chloride–Vinylidene-chloride Copolymers. These copolymers were developed to furnish resins combining the unusual stability and chemical resistance of polyvinyl chloride with the solubility and plasticity essential to certain processing operations, particularly in the field of coatings. Flexible tubing of all kinds has proved successful when based on these resins. They are practical for use in wire insulation. Molding compounds find outlet in such articles as piston cups and seals for grease guns, buckets and ladles for chemicals, storage-battery jars and cases, grommets, valve disks and bellows. Calendared and cast films are suitable for protective clothing, containers, shoe soling, electrical tape, and similar items. Trade names are Geon and Glioflex.

Cellulose Acetates—Molded. Cellulose acetate is made by treating especially purified cotton linters with acetic acid and acetic anhydride. Cellulose acetate molding composition is cellulose acetate with suitable plasticizer, pigments, and dyes. It is thermoplastic, changing its form under the application of heat and pressure without any chemical change (known as curing). It is generally necessary to cool the mold about 100°F below the molding temperature before removing the casting.

Advantages. The chief advantages of cellulose acetate molding composition over other molding compositions are extreme toughness, resilience, and high polish. Because of its toughness it can be used in thin-walled sections where other molding compounds are inadequate because of their brittleness. It is particularly adapted to the molding of articles which come in contact with the skin since it is a low conductor of heat, takes an exceptionally smooth finish, and is pleasant to the touch. It is also without odor or taste.

Colors. Colors available are colored and crystal transparent, slightly pigmented but unfilled translucent, and variegated configurations of mixed-color mottles. The price range on the raw material is wide depending on the brilliance of the color required. Pearl effects are available but rank among the most expensive. Any color can be matched, but a guaranteed minimum of 200 lb for each color is required to hold the standard price. Five thousand pounds of a group of colors rates the minimum price on raw material.

Water Absorption. Cellulose acetate owing to its average 1.9 per cent water absorption, is not suitable for use as containers for water or anything containing water.

Heat. Because it is thermoplastic, softening at 150°F, and not in any way changing its chemical composition as phenolics and ureas change, this material should be used only at temperatures below this figure. Cellulose acetate differs from cellulose nitrate in its reaction under extreme heat. It merely burns like heavy cardboard, fiber, or wood. It is listed under the Underwriters' Laboratories Re-examination Service and rated as follows: "Hazards of this product in use are judged to be small and in storage somewhat less than would be presented by common newsprint paper in the same form and quantity."

Workability. Cellulose acetate has excellent machining qualities, using standard brass-working methods. Two-flute tools are better than those with three or four since they clear the material away faster. The material has an abrasive action which tends to dull the tools; therefore the tools should be nitrided. The basic action of machining should be confined to cutting rather than filing or sanding. It should be cut, turned, punched, or scraped rather than filed, sanded, or burred. Do not overheat.

Cementing. The two surfaces to be joined should be as smooth as possible and absolutely clean. Rough surfaces or a film of oil will cause a poor bond. Lightly pigmented compounds can be cemented by melting the contacting surfaces with acetone. More heavily pigmented compounds do not take the bond so easily. Properly done, an excellent bond can be obtained. The raw-material manufacturers have special cements, one of which may be found necessary.

Chemical. Cellulose acetate is uninjured by contact with vegetable and most mineral oils. Alcohols and some essential oils of an aromatic nature are partial solvents and will soften or spot the surface of a molded piece. Dilute acids and alkalies have only a slight effect, but the material decomposes under increasing concentrations. The passage of time and the action of sunlight have very little effect. Cellulose acetate is soluble in ketones and esters such as acetone, ethyl acetate, cellosolve; softened by alcohol; insoluble in hydrocarbons like benzine, toluene carbon tetrachloride, oils (see Table 34-1).

NOTE: Articles coated with pyroxylin or cellulose nitrate lacquer should not be placed in direct contact with parts molded of acetate powder. The plasticizers in the acetate soften the nitrate film causing tackiness which may result in a portion of the film being removed when the two articles are separated.

Each manufacturer of cellulose acetate molding compound has several formulas which he supplies the molder to fit certain conditions. For instance, the molders can buy a compound with emphasis on free flow, fast setting, welding, optical, and the like. These formulas are based on different plasticizers. In each formula the physical-flow properties of the material can be again varied by the percentage of the plasticizer used. These flow variations range from H_7—a very low percentage of plasticizer, hence a very reluctant flow—down through H, MH, M (for medium), S for soft, and a range of S numbers, even to S_{10}. Some formulas have as many as 14 different flows available, none fewer than 7.

An all-purpose formulation may range from H_2 to S_6 in 11 steps. The molded parts are very different in physical properties, while electrical properties are not affected appreciably.

Table 34-1. Chemical Effects of Cellulose Acetate Molding Compound

	Reagent	Effect
Acid, standard	30 % sulfuric	Partially decomposes
	3 % sulfuric	No change
Supplementary	10 % nitric	Decomposed
	5 % acetic	No change
	10 % hydrochloric	Decomposed
	Oleic	No change
Alkalies, standard	10 % sodium hydroxide	Decomposed
	1 % sodium hydroxide	Much softened and swollen
Supplementary	10 % ammonium hydroxide	Decomposed
	2 % sodium carbonate	No change
Solvents, standard	95 % ethyl alcohol	Softened, blushed, swollen
	50 % ethyl alcohol	Softened, blushed, swollen
	Acetone	Dissolved
	Ethyl acetate (85–88 %)	Dissolved
	Ethylene dichloride	Dissolved or very soft—flabby
	Carbon tetrachloride C.P.	No change or swollen
	Toluene	No change or swollen
	Gasoline (white)	No change
Miscellaneous, standard	Water	No change
	10 % sodium chloride	No change
Supplementary	3 % hydrogen peroxide	No change

Cellulose acetate molding compounds are made under the following trade names: Ampacet, Fibestos, Koppers, Lumarith, Nixon C.A., Plastocele, and Tenite I.

High-acetyl Acetate. In ordinary cellulose acetate the percentage of the combined acetic acid runs from 5 to 56. From 56 to about 62, the product is called high-acetyl acetate. This material has come into its own mainly during and since the Second World War. It requires special solvents to make it and is more tricky to mold. Moisture and heat resistance are improved over the regular type of acetate, also resistance to organic solvents, to mineral, animal, and vegetable oils, and to gasoline. It loses none of the inherent qualities of toughness, strength, and resiliency for which cellulose derivatives are noted.

Effects of sunlight and outdoor aging are dependent upon the locality in which the articles are exposed and their particular composition, *i.e.*, the type and quantity of plasticizer, etc. Requirements for resistance to outdoor exposure should be considered a special problem. Experience has shown that articles molded from cellulose acetate remain fully serviceable with indoor aging at normal temperatures.

Cellulose Acetate Butyrate. Cellulose acetate butyrate molding compositions are made of cellulose acetate butyrate and plasticizers with or without the addition of pigments or coloring matter. Cellulose acetate butyrate is made by the esterification of alpha-cellulose with acetic and butyric acids and anhydrides in the presence of a catalyst. For chemical effects, see Table 34-2.

Table 34-2. Chemical Effects of Cellulose Acetate Butyrate

	Reagent	Effect
Acid, standard	30 % sulfuric	No change
	3 % sulfuric	No change
Supplementary	10 % nitric	Discolored
	5 % acetic	No change
	10 % hydrochloric	No change
	Oleic	No change
Alkalies, standard	10 % sodium hydroxide	Slightly softened
	1 % sodium hydroxide	No change
Supplementary	10 % ammonium hydroxide	Discolored
	2 % sodium carbonate	No change
Solvents, standard	95 % ethyl alcohol	Swollen
	50 % ethyl alcohol	Swollen
	Acetone	Dissolved
	Ethyl acetete (85–88 %)	Dissolved
	Ethylene dichloride	Dissolved
	Carbon tetrachloride C.P.	Much swollen
	Toluene	Much swollen
	Gasoline (white)	No change
Miscellaneous, standard	Water	No change
	10 % sodium chloride	No change
Supplementary	3 % hydrogen peroxide	No change

Their stability is good under service conditions because they show very little loss in weight over long periods of time because of the fact that the plasticizers used have low volatility, low solubility in water, and are present in relatively small quantities. Because of the low plasticizer content, moldings have an excellent finish and usually do not require buffing. Certain hard formulas will resist distortion and impairment when boiled for a short period of time. Other outstanding properties are their toughness, high impact strength, wide compatibility with organic substances, and their ease of molding. Choice of this material should be governed by whether average properties are sufficient for the intended purpose or whether superior resistance to either heat or impact may be required. It should be remembered that, in general, superior shock resistance is obtained at a sacrifice of heat resistance and conversely.

Cellulose acetate butyrate compounds are used for such items as toothbrushes, telephones, toilet float bulbs, bayonet scabbards, pistol grips, gas-mask valves and guards, and steering wheels.

Effects of sunlight and outdoor aging are dependent upon the locality in which the articles are exposed and their particular composition, *i.e.*, type and quantity of plasticizer, etc. Requirements for resistance to outdoor exposure should be considered a special problem. Experience has shown articles molded from cellulose acetate butyrate remain fully serviceable with indoor aging at normal temperature.

Cellulose Propionate. Cellulose propionate, was the first new plastic to be announced after the close of the Second World War. Too little experience is available at this moment to do more than cite the claims of its makers, the Celanese Corporation of America. They state it has the best balance of basic characteristics of any cellulose plastic in existence today.

Its dimensional stability, impact strength, beauty, and moldability are said to be outstanding. Chemically it is a straight cellulose ester, cellulose propionate combined with suitable plasticizers. The propionate is made in the form of a flake by the reaction of propionic acid and anhydride with wood pulp or cotton linters in the presence of a catalyst. Propionic acid is a 3-carbon acid falling between acetic and butyric in the fatty-acid series. The propionic acid for this new plastic comes from petroleum raw materials.

Specific gravity is 1.2. Impact strength is roughly twice that of cellulose acetate. Moisture absorption is low, expansion in high humidity a minimum. Dielectric strength is the same as cellulose acetate; power factor and dielectric constant are lower. The trade name for cellulose propionate is Forticel.

Ethyl Cellulose. Ethyl cellulose plastics are made from the ethyl ether of cellulose combined with suitable plasticizers, mold lubricants, and coloring agents. They are distinct chemically from the other cellulose plastics, which are esters of cellulose. The well-known chemical stability of ether compounds is reflected in their stability to hydrolysis, or to chemical breakdown, and to their resistance to dilute acids, to concentrated and dilute alkalies, and to heat. Moldings of these thermoplastic materials are outstanding in toughness and strength, low temperature, shock resistance, and retention of original properties on aging or on exposure to a wide range of temperature-humidity conditions. Their moldings also possess excellent dimensional stability and a high degree of resistance to water and distortion by heat. They are free from odor, light in weight, and have good insulating qualities.

Cellulose Nitrate. Cellulose nitrate is the oldest of the group of plastics now in common use. It is not moldable because of its flammability, a disadvantage that has been the subject of much expensive research by its makers without too much in the way of results. Parts made of nitrate are formed or machined. It is the toughest of all the thermoplastics. This quality and the ease of fabrication are its chief advantages. Other desirable qualities are its dimensional stability, resilience, low water absorption, and resistance to mold acids. A wide variety of transparent, opaque, and mottled colors can be produced. It may be cut, sawed, punched, drilled, drawn, embossed, printed, polished, and swaged. It is easily cemented to itself and other materials.

It is widely used in fountain pens, spectacle frames, and drawing instruments, where dimensional stability is necessary.

It is available in beads, color chips, lacquers and emulsions, rods, tubes, and sheets of various thicknesses, usually 20 by 50 ft standard.

Benzyl Cellulose. Among the most noteworthy properties of this new material, benzyl cellulose, are its extremely low moisture absorption, 0.44 per cent in 24 hr, its excellent electrical properties, including high arc resistance, and its good resistance, when properly formulated, to chemical attack. It may have a definite place in the plastics field as a hot-melt coating, extruded on wire for electrical insulation, or in extruded or injection molded shapes. Its trade name is Hercules Benzyl Cellulose.

Methyl Cellulose. Methyl cellulose is forming a long-felt need in industry for a powerful synthetic thickener, free from property variations present in natural gums because of their variable growing conditions.

It is used in the field of emulsions, particularly in water-emulsion paints. In the packaging industry, the property of oil and grease resistance makes it very useful. Its trade name is Methocel.

Polyisobutylene. Polyisobutylene first appeared in Germany under the name of Oppanol and was later introduced in the United States as Vistanex.

It is made in a wide range of molecular weights which produce materials from lubricants to tough rubberlike polymers.

It is not used alone because it undergoes considerable cold flow and has little structural strength. It is compatible with a wide variety of other materials and is used to change and improve their qualities. It can also be compounded with various fillers. It differs from natural rubber in that it cannot be vulcanized. It is completely saturated and therefore has good resistance to strong chemicals and ozone. It is used in cable insulation and sheathing and is available in commercial quantities at low prices.

Polytetrafluorethylene. This material known as "Teflon," is a remarkable resin that possesses mechanical toughness over a wide range of temperatures and excellent electrical properties over a wide band of frequencies. Teflon is chemically inert and has high heat resistance. It has no true melting point. At about 420°F small amounts of gas are given off, and at 750°F it decomposes slowly. While its heat-distortion temperature is 266°F, it can be used over a temperature range of 100 to 480°F. Dielectric losses of Teflon are practically constant from 60 cycles to 3,000 megacycles and are as low as either polystyrene or polyethylene. Its resistance to surface arcover is high, and on failing it melts and vaporizes and does not leave a carbonized conducting path. Boiled in aqua regia, hydrofluoric acid, or fumic nitric acid, it shows no change in weight or properties. The effect of strong alkalies and organic materials is almost zero, and outdoor weathering gives no detectable change in a 1-year test. It can be extruded as rods and tubes or as a wire coating. Compression molding of simple articles can be done, but more complicated shapes must be machined.

Teflon can be machined easily with regular wood- or metal-working tools if they are kept sharp. So far no usable cements have been found. Only limited quantities are available.

Casein. Casein has been known for a long period as milk curds or cheese, which is casein in a very impure form. It occurs in the milk of all mammals, although it differs in quality and quantity from one species to another. An average cow's milk carries in suspended or colloidal condition about 3 per cent of this material, which is best described as phosphoprotein, a line compound in combination with calcium phosphate. A perfectly clean and, so far as possible, neutral separated milk is used, and a quantity of rennet is added. Rennet casein is usually chosen as being most suitable, giving the toughest finished raw material.

If casein is to be made into sheets, the extruded material is placed in hydraulic presses, compressed to the desired thickness, cured in formaldehyde, dried, and straightened. If the material is to be used as rods, the extruded rods are cured, dried, and ground to size.

It is made and sold in the form of sheets, rods, and tubes. Ameroid and Galom are trade names for casein.

Lignin-base Compounds. Lignin is nature's binder in the formation of growing things. It is widely used in forming wallboard and sheets of various sorts but has not proved itself in competition with phenolics for individual shaped pieces. Its principal attraction is its low price, but this is overborne by the need to chill in the mold and its reluctant flowing qualities. Woody materials consist of water solubles, such as turpentine, rosin, tannin, and the like; wood-sugar anhydrides, which can be converted by acid hydrolysis to water solubles; cellulose, the fibrous portion; and lignin, the binder. In papermaking the lignin is removed. In the process of making lignin-base materials, the chips of wood are heated under high pressure which is suddenly released, resulting in an explosion of the fibrous material. The mass is treated to remove the sugars, and a moldable material is left. The lignin pulp remaining is essentially a thermoplastic material but definitely does not lend itself to injection molding. As a compression-molding compound it reacts with amines, furfural, and phenol to form a thermosetting resin. An average formula would be 15 to 28 per cent phenol resin and 77 per cent ground-lignin pulp. It can be pulled hot from the mold.

Shellac. In the plastic-molding field the first extensively used material was porcelain. While this material had excellent heat-, water-, and electrical-resistant properties, its brittleness and lack of accuracy led to the development of rubber binders. These in turn were displaced by the use of shellac compounds because of the constantly increasing price of rubber. Shellac as a binder in a molding compound probably outdates any other binder known. Other natural gums such as copal, sandroc, etc., have been incorporated in the mixtures as the price of shellac rose. The fillers used are innumerable, but the outstanding ones are asbestos, mica, marble flour, and wood flour.

The applications of shellac materials are varied and widely diversified. Finish is excellent, the parts taking a high polish. Accuracy can be maintained within 0.002 in. if necessary. Dies are about the same as used for phenolic materials. For a long period shellac products were marketed in practically any color, but now only natural and black are used.

The material is rather brittle. It is noncombustible and will not sustain flame. It has a good weather resistance and is practically impervious to moisture. Its greatest disadvantage is its low heat resistance which at present has a limit of 175 to 180°F.

Various experiments are under way to raise this softening point. These may, if successful, increase the popularity and application of this material. A trade name for shellac is Complac.

Cold-molded Compounds. Cold-molded compounds can be grouped in three classes: those using bitumen binders, cement binders, and resin binders. The most widely used are the bitumen binders consisting of asphalt or gilsonite, drying oils, drier, coloring matter, and occasionally a small percentage of natural gums combined with asbestos fiber, the latter about 80 per cent of the resulting compound. Impact and tensile strength are about half those of phenolic materials. Cost of material and dies is considerably less than for hot-molded phenolics. When strength is required and weight and volume are not important, cross sections can be increased considerably and costs still kept low.

In brief, the three types can be characterized as follows:

Bitumen binders are the least expensive and are used principally in parts requiring a high degree of plastic flow in molding.

Cement binders have great physical strength and high heat and arc resistance when suitably treated. Their cost relatively high.

Resin binders are physically stronger but not so heat resisting as bitumen binders.

PLASTICS VERSUS METALS

Any attempt to compare plastics and metals on a competitive basis is difficult because of the versatility of both groups of materials. The general idea that plastics will in some way be competitors of metals (especially light metal) may be true to a certain extent, but the area of overlapping will be found to be comparatively small. Plastics are not a tonnage industry as are practically all metals. Even magnesium, the newest metal, is being produced at something like double the entire output of plastics. Even in the competitive area the rivalry is not general. *Molded plastics* compete with metal chiefly when metals are die-cast, sand-cast, or machined. *Laminated plastics* are competitors of metals when the latter are considered in sheets, formed, or machined.

Resin coatings compete with metals used as plating mediums. These are the three principal occupants of the zone where plastics and metals overlap. Outside this area the two types of materials are "mutually exclusive." Plastics cannot be used where continuous heat above 400°F will be applied. Plastics lack the surface hardness of most metals, although many plastics seem to be better under abrasion than many metals. Again, if only a small space is available and the cross section therefore limited, metals are invariably chosen because they provide the requisite strength.

Broad Distinctions. Plastics are beyond the competitive reach of metals where (1) satisfactory electrical insulating quality is required; (2) a material is needed that will waterproof textiles so as to leave them flexible enough to use as raincoats, shower curtains, and the like; (3) transparency is required; (4) a material is sought that is suitable for protection against ordinary heat conduction in such simple applications as coffee-pot, gas-stove and electric-iron handles, and numerous other similar parts; (5) the material must feel warm and comfortable to the touch; and (6) a choice of built-in colors is desired. If those were the sole bases of comparison between plastics and metals, metals would already be at the start of a long decline. However, the problem cannot be settled so simply in favor of either side. In the two most important fields where the choice may be difficult—formed metals vs. molded plastics and sheet metals vs. sheet plastics—it may be necessary to draw finer distinctions. Not all metals can be formed with equal ease. As with plastics, compromise is usually invoked. A certain alloy steel may be ideal as to service quality, but to form it into the shape needed may involve prohibitive cost. Molded plastics need rather expensive steel molds, but so do metal die castings. Sand-cast metals offer a cheaper approach but may leave too much machining to be done to obtain the needed accuracy. Plastics molded parts seem to fit into a middle band of accuracy. If the part is really rough and $\frac{1}{16}$ in. means nothing, a sand casting is indicated; if the tolerances, however, are less than 0.001 in., carefully machined metal is a better choice. Between these two, molded plastics may be considered. Other things being equal, the greatest advantages plastics offer are built-in colors and surface permanence. Plastics are not affected by either corrosion or electrolysis. Die-cast or stamped metal is preferable when thin sections must be rigid or bear any appreciable strain. Strong thin plastic parts can easily be made in limited shapes by using reinforcements of laminates or fiber stocks, but they will not be rigid. Plastics can be threaded as well as metals if there is enough "stock" around the thread to prevent its cracking out.

Table 34-3. Comparison by Weight of Some Plastics and Metals

(Lightest at top of the list)

Materials and characteristics	*Specific gravities*
Formica Pregwood No. 100; impregnated compressed wood; all laminations parallel, classed as a high-strength product, water absorption 6 per cent maximum	1.30
Formica-canvas laminate phenolic; grade C; cotton fabric—4 oz per yd; tough; water absorption 0.3 per cent	1.33
Formica-paper-base laminate-phenolic; grade X; primarily for mechanical applications; water absorption 4.0 per cent	1.33
Bakelite wood-flour phenolic BM-120—best all-round molding composition; water absorption 0.3 per cent	1.36
Bakelite macerated-canvas phenolic BM-3510; high-production, high-impact molding; water absorption 1.0 per cent	1.38
Formica-glass-mat-base phenolic; grade MF; basically for electrical purposes; water absorption 0.35 per cent	1.50
Formica asbestos-fabric-base laminate—Grade AA; strength, toughness, minimum dimensional changes; water absorption 1.5 per cent	1.80
Magnesium alloy (AM-585)	1.81
Aluminum alloy (24 ST)	2.80
Stainless steel.	7.85
Chrome-moly steel	7.85

Engineering Factors in Selection. The first factor determining the choice of plastics or metals is the available space or volume of the part. If it is relatively small, plastics are almost automatically out unless their use is demanded by one of the six "exclusives" mentioned. The second factor is cost. All plastics cost more than metals on a pound basis. The saving lies in their low specific gravities (see Table 34-3). Even magnesium with a specific gravity of 1.80 is higher than the average range of 1.06 to 1.50 for plastics. A correction of the usual strength values for the specific-gravity effect may bring plastics into the range of the engineering metals.

To make any comparison, the very best plastics material must be chosen, and one of the weaker alloys should be selected.

On this basis, tensile-strength comparisons should be made (see Table 34-4). Chrome-moly steel and stainless steel show around 180,000 psi; aluminum alloy, 62,000; and magnesium alloy, 46,000. One of the strongest plastics is Pregwood, (impregnated plywood). Its tensile strength is 30,000 psi. Other "strong" plastics are paper laminate at 12,500 psi; canvas laminate, 9,500 psi; and wood-flour-phenolic molded parts, 8,500 psi. It is evident that plastics offer no competition in direct comparisons of this type.

Table 34-4. Comparison of Strength–Weight Properties of Plastics and Metals

Material	Specific gravity	Tensile strength, psi	Modulus /sp gr	Ratio (magnesium alloy = 100)
Magnesium alloy	1.81	46,000	25,400	100
Stainless steel	7.85	185,000	23,600	93
Pregwood	1.30	30,000	23,000	90
Chrome-moly steel	7.85	180,000	22,900	90
Aluminum alloy	2.80	62,000	22,100	87
Paper laminate	1.33	12,500	9,400	36
Glass-fabric laminate	1.50	14,000	9,300	36
Canvas-fabric laminate	1.33	9,500	7,100	28
Wood-flour phenolic, molded	1.36	8,500	6,200	24
Asbestos-paper laminate	1.80	10,000	5,500	21
Impact phenolic molded	1.38	7,500	5,400	21

When determining specific tensile strength or tensile strength per pound (see Table 34-4) by dividing the strength figures by specific gravity, one of the plastics, Pregwood, moves to third place on the list at 23,000, following magnesium alloy at 25,400 and stainless steel at 23,600. Chrome-moly steel at 22,900 and aluminum alloy at 22,100 are far above paper laminate at 9,400, canvas laminate at 7,100, and wood-flour phenolic, molded, at 6,200.

This comparison shows that, if competition is confined to sheet materials, the plastics stand up fairly well in tensile strength by weight, but the moment it centers about shapable forms, by bringing in the moldable group, the plastics lose position rapidly.

With respect to compressive strength, the steels are strongest, with some of the plastics better than some of the other metals (see Table 34-5). When the specific-gravity correction is applied, all the plastics except Pregwood rate higher than the metals. Therefore, if compressive strength is the determining factor in a specific design, plastics (some form of thermosetting phenolic or urea) will serve the purpose best pound for pound.

Similar data on modulus of elasticity in tension, which is a measure of a material's rigidity, are given in Table 34-6 and show clearly that the metals are outstanding in this respect even on a per-pound basis. Plastics are very inferior to metals with regard to shear, impact, and flexural strengths.

Metal is chosen over plastic where there is a really severe application of an engineering nature, unless the stronger metal has its own disadvantage (such as poor resistance to corrosion or electrical characteristic) that may be even more serious than the mechanical deficiency of the plastic.

When compressive strength is important in design, almost any kind of thermosetting plastic material, except plywood, will give the best results on a pound basis. Under tension plywood compares favorably with leading metals on a pound-for-pound basis.

Plastics have a few outstanding service qualities where the previously mentioned measures of strength are not all-important. One of these is abrasion resistance. Ammunition chutes of laminated phenolic outwear steel. Plastic gears are not only quieter but wear as well as steel. Under impact plastic sheets will not dent and gradually deform before failure but will break suddenly, while metal, although it will not fail completely, will deform. Plastics will dampen machine chatter better than metals, resulting in less noise and vibration.

Sheets of plastic cannot be drawn to new shapes so easily as metals, but usually when odd shapes are desired they can be created in the plastic itself without making the sheet. In other words, sheets of metals are raw materials suitable for a great deal more working than sheets of plastics.

There is one unfortunate characteristic of plastics that occurs also in metals but not to such an extent, and that is creep. Creep (cold flow) is the constant change in dimension under stress. The thermoplastics are strongly subject to this and therefore are avoided for stressed service. In the thermosetting materials, creep is present, but unless the material is used with practically no factor of safety, it may be discounted.

Table 34-5. Compressive Strength Weight Values of Plastics and Metals

Material	Specific gravity	Compressive strength, psi	Modulus /sp gr	Ratio (canvas laminate = 100)
Canvas laminate	1.33	38,000	28,000	100
Paper laminate	1.33	35,000	26,000	93
Glass-fabric laminate	1.5	40,000	26,000	93
Impact phenolic, molded	1.38	35,000	25,000	89
Wood-flour phenolic, molded	1.36	30,000	22,000	78
Asbestos laminate	1.80	38,000	21,000	75
Magnesium alloy	1.81	35,000	19,300	69
Chrome-moly steel	7.85	150,000	19,100	68
Stainless steel	7 85	150,000	19,100	68
Aluminum alloy	2.80	40,000	14,300	51
Pregwood	1.3	15,000	11,000	39

Table 34-6. Moduli of Elasticity in Tension and Specific Gravities of Plastics and Metals

Material	Specific gravity	Modulus of elasticity, psi	Modulus /sp gr	Ratio (stainless = 100)
Stainless steel	7.85	30×10^6	3.8×10^6	100
Chrome-moly steel	7.85	29×10^6	3.7×10^6	98
Aluminum alloy	2.80	10.4×10^6	3.7×10^6	98
Magnesium alloy	1.81	6.5×10^6	3.6×10^6	94
Pregwood	1.30	3.7×10^6	2.84×10^6	75
Paper laminate	1.33	3.0×10^6	2.25×10^6	59
Glass-fabric laminate	1.50	2.0×10^6	1.34×10^6	35
Impact phenolic, molded	1.38	1.8×10^6	1.30×10^6	34
Canvas laminate	1.33	1.5×10^6	1.12×10^6	29
Asbestos laminate	1.80	1.5×10^6	0.84×10^6	22
Wood-flour phenolic, molded	1.36	0.96×10^6	0.70×10^6	18

SECTION 35

PATENTS AND COPYRIGHTS

BY SHELDON H. GRAVES

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PATENTS

BY SHELDON H. GRAVES

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DEFINITION

A *patent* is a grant by the Federal government to an inventor, his assigns, or personal representatives of exclusive right to make, use, and vend the subject matter of his invention for a limited time throughout the United States and the territories. The term of a patent is 17 years except for a design patent, which may be for 3½, 7, or 14 years, as the applicant for design patent may elect.

A patent may also be considered as a contract between the inventor and the public whereby, in consideration of a full disclosure of the invention to the public, which disclosure is contained in the description and drawing, if any, of the patent, the inventor is granted the exclusive right to practice the invention as this invention is defined in the claim or claims of the patent.

The patent grant is statutory under authority granted to Congress by the Constitution of the United States and differs from the common-law right of the inventor which is merely the natural right to practice his invention and to keep the invention secret until others have acquired knowledge of it. The exclusive right granted by a patent is the right to exclude others from making, using, or vending the subject matter of the invention and does not enlarge the inventor's common-law right to practice his invention.

APPLICANTS, APPLICATIONS, AND PROCEEDINGS

Applicant. An application for patent must be made by the inventor if alive and sane. If the inventor is dead, the application may be made by his executor or administrator and, if insane, by his legally appointed guardian, conservator, or representative.

An inventor is a sole inventor when he alone has conceived the invention, or he may be a joint inventor when he, with one or more others, has contributed to the conception of the invention.

An inventor may hire someone to work out the details of the invention, to embody it in practical form, or to build a model or make drawings. This, however, does not make the one so employed a joint inventor with the employer, even though the one so employed in the course of his duties makes an ancillary invention. One who furnishes capital for the development of the invention or for the filing or prosecution of a patent application thereon does not thereby become a joint inventor, and he cannot be a joint applicant with the one who has conceived the invention, although he may be a joint patentee.

Application. An application for a patent must be signed by the inventor and filed with the Commissioner of Patents in Washington, D.C., and all business with the Patent Office connected therewith is in writing and in the name of the applicant, who may and usually does employ an attorney to conduct the proceedings.

Rules for the filing and prosecution of an application, together with forms to be followed are given in the "Rules of Practice of the United States Patent Office in Patent Cases," a copy of which may be obtained for 40 cents by addressing the Commissioner of Patents. Attention will be called in the following pages to certain requisites of papers filed in the Patent Office. The mention of such requisites is not intended to be all-inclusive, and the "Rules of Practice" should be consulted in every case.

A patent application comprises a petition, signed by the applicant, addressed to the Commissioner, and requesting the grant of a patent; an oath signed by the applicant and executed before a notary public or other designated officer who must place his official seal on the instrument, which petition and oath should follow the form presented by the "Rules of Practice"; a specification and claim or claims, which must also be signed by the applicant, and a drawing when the invention is susceptible of disclosure by drawing. The drawing must be signed by the inventor or his attorney. The petition usually includes the appointment of an attorney to prosecute the application. The petition, oath, specifications and claims, and power of attorney, if one is used, may be included in a single document and may be executed by a single signature of the applicant if an approved single signature form supplied by the office or approved by the office is used.

Application papers must be accompanied by the filing fee, which at the present time is $30, plus $1 for each claim in excess of 20, except in design cases, in which the entire fee is $10, $15, and $30, for terms of 3½, 7, and 14 years, respectively.

The specification, which should follow the orderly form prescribed by the "Rules of Practice" begins with a general statement of the object and nature of the invention which may include reference to advantages of the invention over the prior art. There then follows a brief description of the drawings, if there are drawings, and a detailed description of the invention. The description as a whole must be so clear and complete that one skilled in the art to which the invention relates, on reading the same, may reproduce the invention in a concrete operative form without the necessity of his exercising any invention of his own.

Unnecessary details need not be given in the description provided they can be supplied by one skilled in the art. This is also true of the drawings. They need not show parts in exact dimensions or proportion unless such dimensions or proportions are part of the invention itself. In an application for a design patent, inasmuch as the invention resides in the appearance of the article, a description which merely refers to the drawing will be sufficient.

The drawing must conform to the requirements set forth in the "Rules of Practice." The figures of the drawing, which should be consecutively numbered, are described in the specification in the brief description of the drawing above noted, and reference characters should be applied to parts of the structure shown and such reference characters included in the description of such parts in the specification.

The description is followed by the claims. A claim is a formal statement of the invention stripped of extraneous matter and defines the patent right or monopoly sought. Where the invention is an article of manufacture or a machine, the claim recites the part or combination of parts constituting the invention and should state so much of the characteristics of the part or of the relationship of the parts of a combination that the manner in which the parts cooperate between themselves or with others is clear. Similarly, a process claim recites the step or combination of steps constituting the process, and in a claim for composition of matter the ingredient or ingredients are so set forth.

Claims quite often state the art to which the invention relates, in the form of an introductory clause; *e.g.*, "In a double-acting pump, the combination of, etc.," or "A lubricant consisting of, etc." In a design patent the claim is merely for the particular article as shown.

Claims can be classified as *generic* or *specific*. A generic claim is one which describes or "reads on" several embodiments of the invention or is limited to features which are common to a number of embodiments. A specific claim is one which reads on only one embodiment or on a fewer number of embodiments than a generic claim.

It is common practice to include in the application broad or generic claims as well as specific claims. If eventually it turns out that the invention is broadly new, both kinds of claims are valid. If some prior specific embodiment of the invention which would be included under the generic claim, but not under the specific claim, should subsequently come to light, the specific claim would still be valid.

As the drawing of claims adequate to protect the invention disclosed requires a high degree of skill and experience, the applicant, unless he is very familiar with this practice, is advised to employ a competent patent attorney.

Proceedings. When the patent application is filed in all its parts, the Patent Office gives it a serial number and informs the applicant of the number and the date of filing. The date of filing, if the application discloses an operative invention, constitutes a record date of invention and prima-facie proof of its reduction to practice and is the date upon which the Office regards the invention as having been completed in the absence of other evidence, the importance of which will be more fully discussed hereafter.

When filed, the application is sent to an examining division which examines applications in the art to which the invention relates, and in the course of time (usually several months) the first official action on the application is mailed to the applicant's attorney or to the applicant.

In this office action attention will be called to any informalities in the application papers. Note may be made of grammatical errors, and if there is anything not clear in the description, the applicant will be required to clarify the same by amendment. In many applications, for record purposes as filed, structures or modifications are described in the specification which are not shown in the drawing. In this case the examiner will require that the structure be shown in the drawing or the description of the same canceled from the specification.

If the disclosure or any part thereof is considered inoperative, the applicant's attention will be called to this fact, and he will be called upon to correct the same by amendment. In doing so, or in amending for any other purpose, care must be taken not to introduce into the application any new matter, *i.e.*, any substantive matter not predicated on the disclosure of the specification, drawings, or claims as originally filed. However, simple changes may be made where it is obvious to one skilled in the art what is needed to correct the inoperativeness. Where the inoperativeness goes to the essence of the invention and cannot be remedied without the introduction of new matter, the application is fatally defective.

Informalities in the claims will also be called to applicant's attention, and where the invention claimed can be understood, the examiner will give an action on the merits of the claims unless as a preliminary matter he requires division or requires election of species.

Division. Division is required when the examiner finds that applicant has claimed in his application two or more independent inventions. Where such is the case he will be required to elect which invention he desires to prosecute in that particular application and to limit the claims in such application accordingly. He may claim another of the disclosed inventions in a second or divisional application, which should be filed while the first application is still pending in order that such second application shall have the benefit of the filing date of the first application.

In response to a requirement for division, the applicant may file an argument to the effect that the inventions claimed are not independent and are such as may properly be included in one patent. The examiner then usually refers the question to another examiner known as the classification examiner, who will approve or disapprove the requirement for division, and on approval the examiner in charge of the application will repeat the requirement and make it final. From such final requirement, appeal may be taken to the Board of Appeals; or the applicant may elect which invention he wishes to prosecute, continue the prosecution of such invention, and later appeal to the Board of Appeals on the question of division along with such other matters as may then be appealable, should he so desire.

Election of Species. Election of species may be required when the applicant discloses more than one embodiment of the invention or more than one way in which the invention may be carried out and at the same time separately claims different specific embodiments. Where the applicant files an allowable generic claim covering more than one embodiment of the invention, he may claim specifically in separate claims different embodiments of the invention, not exceeding three, provided all three fall under such generic claims. When there is no such allowable generic claim, the applicant will be required to elect which species he desires to prosecute, and thereafter claims to such species only will be considered on their merits.

Whenever a requirement for division or for election is to be made, the examiner usually makes a cursory examination of prior art and in his requirement cites such patents or publications as may assist the applicant in making his election.

When such preliminary questions as above noted are disposed of, the action on the application will be directed to the merits. The claims will be acted on individually, and they will be allowed or rejected. When rejected, the reasons for rejection will be fully given, and where patents or publications are cited in rejection, sufficient information will be furnished applicant to enable him to order from the Patent Office copies of such citations.

The applicant is given 6 months in which to reply to an office action unless for some special reason the examiner specifies a shorter period. The applicant's reply should be completely responsive. Whenever the applicant considers that an objection or rejection of the Examiner is not proper, an argument should be filed giving the reason why it is thought the position taken by the examiner is not proper. Otherwise, the applicant should amend the specification, claims, or drawings, as the case may be, to avoid the objection or rejection.

Amendment to the specification is made by directing specifically that certain matter be canceled or changed or that certain matter be added. The claims may be amended in the same manner, or original claims may be canceled and others substituted or other claims added to those originally filed. Amendments to the drawing usually take the form of a letter requesting the chief draftsman of the Patent Office to make the changes desired, which letter is accompanied by a sketch showing the changes to be made. The Patent Office makes a reasonable charge for such amendments and may demand a deposit to cover such charges.

In applicant's response to an office action he should reply to every point raised by the examiner. Correction of mere informalities will generally not be insisted upon until the application is deemed to contain allowable or patentable subject matter or the application is in condition for appeal.

When the applicant has replied to an office action, the office will again act on the application. Further objections may be raised and new references cited and new reasons for rejection given. The applicant always has a right to reply or amend as long as new references or new reasons are cited for refusing a patent.

When the examiner rejects on the same references and for the same reasons as have been previously given, he can and generally does make the rejection final. At this stage of the case the applicant presumably has presented his reasons for believing the claims patentable, the examiner still believes them unpatentable, and, an issue having been reached, if the applicant wishes to proceed further, he should appeal to the Board of Appeals.

When such an appeal is taken, the examiner forwards to the Board of Appeals a statement giving the reasons for rejection and sends a copy to the applicant. The Board of Appeals then sets the case for hearing and appellant is required within 60 days from the date of appeal or within 6 months from the date of the action appealed from, to file a brief, which complies with provisions set forth in the "Rule of Practice," at the same time indicating if he desires an oral hearing. If so, the applicant may appear at the hearing and orally argue his case.

From a decision of the Board of Appeals of the Patent Office affirming the rejection of the examiner appeal may be taken to the United States Court of Customs and Patent Appeals; or, alternatively, a bill in equity may be brought against the Commissioner of Patents in the United States District Court for the District of Columbia.

Abandonment of Application. Where no proper reply is made to any office action within 6 months or such shorter time as may be fixed, the application is held abandoned. Unless such an abandoned application can be revived, it is to all intents and purposes dead. A new application can be filed for the same subject matter, but it cannot relate back to the filing date of the abandoned application for the purpose of establishing a date of completion of the invention.

In order to revive an abandoned application, it must be shown to the satisfaction of the Commissioner of Patents that the delay in prosecuting the same was unavoidable. A petition to the Commissioner to revive the application is filed accompanied by a verified statement of facts showing that the delay was unavoidable. Any explanation for the cause of delay should cover the entire period of delay up to the time of filing the petition to revive.

Related Application. Divisional applications have previously been mentioned. They are applications wherein the subject matter or disclosure has been carved out of an earlier filed copending application.

Another type of application commonly met with is called a continuation in part. It often happens that an inventor, after having filed an application on a particular invention, conceives of some improvement or modification which includes more or less of the disclosure of the earlier filed copending application. The later application is called a continuation in part of the earlier application, such continuation in part being thus an application part of the disclosure of which is contained in an earlier filed copending application and part of which disclosure is not contained in the earlier filed application. This later application may contain claims to matter disclosed in the earlier application. As to such claims, the applicant is entitled to the filing date of the earlier application, and as to claims directed to matter disclosed only in the later application, the applicant is entitled only to the filing date of such later application.

Interferences. An *interference* is a proceeding instituted in the Patent Office for the purpose of determining the question of priority of invention between two or more parties claiming substantially the same patentable invention. If two applicants claim the same invention in their respective applications, the Patent Office institutes an interference to determine which of the two made the invention first and grants the patent on the common invention only to him whom the Patent Office considers to be the first inventor.

An applicant may obtain an interference and contest priority of invention with a patentee. The applicant must, however, unless he was claiming the patented invention at the time the patent issued, copy in his application the claim or claims of the patent upon which he wishes to contest priority within 1 year of the grant of the patent. He must at the same time identify the patent and show how the disclosure in his application warrants the making of the patent claim. In case the applicant wins the interference, the Patent Office, although it cannot cancel the patent which it has already granted, will grant him a patent on the common subject matter.

An interference proceeding is highly technical and at times very complicated, and anyone who finds himself involved in interference is advised to employ a competent attorney to represent him.

Allowance. When all claims in an application have been allowed and all formal requirements have been complied with, the application is said to be in condition for allowance, and a notice of allowance is sent to applicant in which he is given 6 months in which to pay the final fee of $30 for 20 or less allowed claims and $1 for each claim in excess of 20, except in design applications for which there is no final fee.

A matter of several weeks elapses between the receipt of the final fee and the grant of the patent. The Patent Office acknowledges receipt of the final fee and informs applicant of the number of the patent and the date it will be issued, whereupon the patentee, his assigns, or legal representatives making or selling any such patented article should mark the same with the word "Patent" together with the number of the patent.

Except during interference proceedings, where the rules permit one to inspect his opponent's application, pending original applications are preserved in secrecy in the Patent Office.

CLASSES OF PATENTABLE INVENTIONS

To be patentable the subject matter of an invention must fall within one of the following statutory classes: art, machine, manufacture, composition of matter, plant, or design. Anything falling within these classes to be patentable must also satisfy the test of novelty, utility, and invention.

Art or Process. An art, by which is meant a process or method, consists of an act or series of acts performed on some subject matter to reduce or transform it to a different state or thing. As a process it must be one which can be employed independently of any particular mechanism for carrying it out, as, for example, where one or more steps of the process may be performed by hand. The result produced by the process must be a physical result, and while it often produces a permanent change in the physical state of an article or composition of matter, it need not do so necessarily. Thus, a method of transmitting speech by causing the flow of electrical undulation corresponding to the sound of the human voice was held patentable.

Also, while most processes involve chemical or other elemental action or involve the employment of forces producing physical change such as heat, light, electricity, and the like, certain methods which involve only mechanical operations have been held patentable.

In a greater part of the field of human endeavor, activities will arise which display ingenuity and novelty and are productive of useful results but are not properly the subjects of patent protection. For example, methods of doing business, systems of bookkeeping, plans of military strategy, and the great mass of activity dealing with human relations are not such arts as are properly the subject of patents.

Machine. A machine is a combination of mechanical parts adapted to receive energy and apply it to the production of some energetic result. It includes any device having moving parts by which energy may be utilized or a useful operation can be performed.

Manufacture. A manufacture or article of manufacture is anything made by the hand of man which is not a machine or composition of matter. This has been held to include building structures.

Composition of Matter. A composition of matter covers all compositions or mixtures of two or more substances, whether the result is chemical union or mechanical mixture or whether they be gases, liquids, powders, or solids. Such a mixture or composition should have properties which are different from or in addition to those possessed by the several ingredients in common A new chemical compound is patentable, and a new mixture of old compounds may be patentable if it produces a new result. This class also includes composite articles.

The distinctions between machines, manufactures, and compositions of matter are not always clear, but an inventor need not know to which of these classes his invention belongs so long as it falls within the field covered by them all.

Plant. The statute provides that anyone who has discovered or invented and asexually reproduced any distinct and new variety of plant other than a tuber-propagated plant may patent the same under conditions which obtain for mechanical patents.

Design. A design is the characteristic of an article which by means of lines, images, configuration, or the like taken as a whole makes a visual impression of uniqueness or distinctive character, and such design, in order to be patentable, must be ornamental.

The article to be the subject of a design patent must have a purpose which is primarily utilitarian. If the purpose is primarily ornamental and not utilitarian, it may, however, be entitled to copyright protection.

NOVELTY AND UTILITY

In order to entitle one to a patent, the invention which he has produced must be new and useful (Revised Statutes, sec. 4886).

Novelty. *Knowledge and Use.* To be new an invention must not have been known or used by others in this country before his invention or discovery thereof, and it must not have been in public use or on sale in this country for more than 1 year prior to his application for patent.

Prior knowledge to defeat a patent must be knowledge of something substantially identical with that for which a patent is sought. It must be more than a mere concept of the invention; it must be knowledge of the thing itself in its concrete operative form. Knowledge "by others" does not mean that such knowledge must be shared generally by the public. It is sufficient if such knowledge was accessible to the public.

The knowledge, use, and sale to constitute a bar must be in this country. But knowledge in this country of use or sale in a foreign country is no bar to the grant of a patent in this country.

Patented and Described. Also, to entitle one to a patent the invention must not have been patented or described in any printed publication in this or any foreign country before his invention or discovery thereof or more than 1 year prior to his application. The printed description to constitute an anticipation must disclose a complete and operative invention in such full, clear, and exact terms as to enable one skilled in the art to which the invention relates or to which it is most nearly related to practice the invention.

Utility. An invention to be patentable must be *useful.* The term is used in contradistinction to frivolous. In point of fact, any utility, however slight, is sufficient to satisfy this requirement. To be patentable the invention must not be mischievous or immoral. A new chemical compound as such may be presumed to be useful, but invention must be involved in its production. There is no statutory requirement of utility for a patent for a design invention.

INVENTION

In the great majority of cases the thing that an applicant describes in his patent application and claims as his invention will differ in some or many respects from anything which had previously been described in any printed publication or was before known or in public use or on sale. In other words, it will differ from anything comprised within that body of material which is known as *prior art.* This does not mean that because of these differences what is claimed is patentable. The differences must be such as to require the exercise of the inventive faculty on the part of one skilled in the art to which the invention pertains in order to reproduce, from the mass of information contained in the prior art, the thing that the applicant devised.

Certain negative rules are applied as a test for invention. For example, a mere change in materials does not generally amount to invention. Thus, if one substitutes for the material of which a part was composed a material which was known to be stronger and more durable and thereby produces a part which is merely stronger and more durable, no invention is involved. On the other hand, sometimes a change in material produces a result which is unexpected and in certain cases brings into use an otherwise unrecognized property of the new material in which case the substitution may involve invention.

Also, it has been held that a mere change in form or proportions or degree does not involve invention. This is because one skilled in the art could readily foresee what result would be produced by such a change. But here again, an unforeseen result may show invention.

A common instance of change in proportions conferring patentability occurs in compositions of matter, as in metal alloys, many of which have been patented. The prior art may show a composition having substantially the same ingredients but in proportions different from those of applicant. It may also be found that, as the proportions are changed, when they depart from those of the prior art and approach the proportions disclosed by applicant, certain unexpected and useful properties are developed in the composition. This is an indication of invention, and the proportions in such case are termed *critical.*

Similar rules of noninvention apply where the change is in location of parts, mere duplication or reversal of parts, making parts integral when formerly separate and vice versa, and adding parts without producing a new and unexpected result.

Also, when a part is omitted from a combination of elements and the sole change produced is the omission of function of that part, no invention is involved, although the omission of a part and rearrangement of the remaining parts so as to perform its function may be a patentable invention.

Invention in Designs. The test of invention for a design patent is the same as for a mechanical patent, *viz.*, whether the design was beyond the powers of an ordinary designer.

PATENTED ON FOREIGN APPLICATION

A further statutory requirement for the grant of a patent is that it must not have been patented by the inventor or his legal representatives or assigns in any foreign country on an application filed more than 12 months (or 6 months in the case of designs) prior to the filing in this country. If the foreign application was filed within 12 months of the filing in this country (or 6 months in the case of designs), the granting of the foreign patent does not constitute a bar to the grant of a patent in this country. Even if the foreign application was filed more than 1 year prior to the filing in this country, the granting of a foreign patent to applicant after the granting of the United States patent does not affect the validity of the United States patent.

ABANDONMENT OF INVENTION

A further requirement for the grant of a patent is that the invention shall not have been abandoned. This occurs by an express declaration of abandonment or relinquishment of a completed invention to the public or by acts or omissions from which an intention to abandon may be inferred. An abandonment of the invention should be distinguished from the abandonment of a patent application, which usually results from failure of an applicant to take timely action in the prosecution of the application as has heretofore been explained.

Another matter should be mentioned here. One who makes an invention may, if he desires, practice it in secret, or he may patent it, in which case it is published to the world. He may not, however, practice it in secret until knowledge of the invention has been acquired by others and then patent it. If one practices an invention in secret until he discovers someone else has entered his field, any patent which he applies for subsequently will be invalid.

DATE OF INVENTION—INTERFERENCES

It frequently becomes necessary for an applicant or a patentee to establish by evidence the date when he made his invention or discovery.

For example, among the statutory requirements for the grant of a patent as above given is that the invention shall not have been described in a printed publication before his invention or more than 1 year prior to his application for patent. If during the prosecution of the application the patent examiner discovers a publication describing the invention and such publication is dated more than 1 year prior to the application, the examiner will call the applicant's attention to the publication and reject the claims of the application thereon, and such publication will constitute an anticipation of the invention. If, however, the publication antedates the application by less than 1 year and does not antedate the completion of the invention, it is not a bar to a patent. The examiner rejects in such case because he has before him as evidence of the date of invention only the application, and in the absence of other evidence, the filing date of the application is deemed to be the date of invention. In reply to such rejection the applicant may submit proof that he made the invention before the date of the publication, and if he does this the examiner will withdraw the publication as a reference.

Another instance where it is necessary to establish a date of invention is where two or more applicants file separate applications, all claiming the same invention. In this case the Patent Office grants a patent covering the common subject matter only to the one who first made the invention, this being determined in an interference proceeding.

In the development of an inventive idea into a concrete embodiment there is often considerable work involved. If the invention is a machine or other device, sketches will often be made, and later a model may be constructed. If the invention is of a chemical nature, experiments may be conducted or samples made and tested to see if the desired results are produced. Either during or after this preliminary work the inventor may file an application for patent, and it becomes important to determine at what point in this activity the invention is considered legally to have been made, *i.e.*, to have been completed in the sense of the patent law.

The general rule is, subject to one exception, that the invention is considered made when it is shown to be complete, and, in general, that is not until it is reduced to practice. *Reduction to practice* is of two kinds—*actual* and *constructive.*

Actual Reduction to Practice. Actual reduction to practice consists in the production, constructing, or building of an embodiment of the invention and the successful testing or operating of such embodiment under conditions actually met with in practice. In certain cases of very simple inventions, where successful operation can be determined from mere inspection of the embodiment, it has been held that proof of test is unnecessary.

Constructive Reduction to Practice. Upon filing an application containing an adequate disclosure of an invention, the applicant is thereafter entitled to the benefit of the filing date of such application as a constructive reduction to practice of whatever is disclosed. He is also entitled to the benefit of the filing date of a copending application containing a disclosure of the invention or of any previously filed application not then pending, provided there was continuously, from the date relied upon, some application pending which contained a disclosure of the invention. An inventor is also entitled to the benefit, for constructive reduction to practice, of the filing date of a foreign application filed by him or his legal representatives or assigns and which discloses the invention, provided the same was filed less than 12 months (6 months in the case of designs) prior to the filing in this country. The benefit of a foreign filing date is given the applicant under a treaty or international convention (known as the International Convention for the Protection of Industrial Property), and a United States application filed within 1 year of the filing of a corresponding foreign application is said to be *filed under the convention.*

The exception to the general rule that to show prior invention one must show reduction to practice, either actual or constructive, is the case frequently arising in ex parte and inter partes proceedings in the Patent Office and in the courts where prior conception is shown by the applicant or patentee accompanied by diligence from the date of the anticipation or bar asserted against him continuing until his own reduction to practice or, in the case of an interference with another inventor, conception prior to that of the other's invention accompanied by diligence continuing without interruption from the time the other inventor entered the field to the date of his own reduction to practice.

From the above it will be seen that it is advisable that an inventor keep a current record of his activity in the development of an invention. He should write a description of the invention as soon as possible, accompanied by drawings, if the invention can be illustrated by drawings, or by examples, both dated and witnessed by someone who understands the invention. A record should be made of the date of conception, the time and place of the first disclosure to others, the making and disclosure of the first sketch and description, the completion of a model or full-size device, tests of the device and results of the tests, extent of the invention's use, and the dates of any applications filed.

OWNERSHIP AND ASSIGNMENT

Ownership. A patent is personal property and may be owned by one or any number of persons. Each of several owners has an undivided interest in the patent, and while such ownership is called "joint," it is more nearly an ownership in common. On the death of such joint owner, his undivided interest passes to his legal representatives. When a patent is granted to two or more joint inventors each becomes a joint owner. To be an owner of a patent jointly with someone else has its disadvantages.

One of two joint owners is to some extent at the mercy of the other. The other can practice the invention, can make, use, and sell a device covered by the patent without accounting to the first owner and license others to make, use, and sell without accounting to the first owner for any royalties received. And this is true no matter how small the other's undivided interest may be, in the absence of an agreement between them providing otherwise.

Assignment. By statute any patent or patent application or any interest therein is made assignable by an instrument in writing, and by such an assignment the exclusive right under the patent may be granted to the whole or any specified part of the United States, the latter being termed a "territorial grant." The assignee in each case should have the written assignment recorded in the Patent Office in Washington, as the law provides that an assignment shall be void against a subsequent purchaser for value without notice unless recorded within 3 months from the date of the assignment or before the subsequent purchase.

If an assignment is acknowledged before a notary public or other officer specified by law, who affixes his official seal, such acknowledgement is prima facie evidence of the making and delivery of the assignment.

One may assign all right, title, and interest in and to a patent, or he may assign one-half, one-third, or any other fractional undivided interest. Where a sharing of profits or royalties among coowners is desired, provision for such sharing should be made by contract.

EXTENSION AND REISSUE OF PATENTS

Extension. The term of a patent can be extended only by act of Congress. A few patents were extended after the First World War, but it is not the policy of Congress to extend patents and has not been for many years.

Reissue. A *reissue patent* may be granted for the unexpired term of a patent already granted when such patent is inoperative or invalid by reason of a defective or insufficient specification or by reason of the patentee claiming as his own invention more than he had a right to claim as new if the error arose by inadvertence, accident, or mistake and without any fraudulent or deceptive intention.

The word "inoperative" as here used, means inoperative to protect the patentee. Thus, if the specification of the patent describes two inventions both of which might have been claimed but only one was claimed, the patent is inoperative to protect the inventor in respect to the other invention.

The statute requires that the reissue patent shall be for the "same invention." By this is meant whatever invention was described in the original patent and appears therein to have been intended to be secured thereby.

An application for reissue must be made by the inventor if living and, if assigned, must be accompanied by the written assent of the assignee. It must be accompanied by the original patent with an offer to surrender the same on grant of the reissue, and the oath accompanying the reissue, in addition to the averments of the oath in the original application, should point out the defects or insufficiency of the specification of the original patent or how the patentee claimed more than he had a right to claim and should specify the inadvertence, accident, or mistake from which the error arose.

Where the claims sought in a reissue application are broader than those of the original patent, the reissue must be applied for promptly—except in most unusual circumstances within 2 years at the most of the grant of the original patent. Where, however, the claims of the reissue are more specific than those of the original application, it is sufficient if the reissue is applied for promptly after the discovery of the inoperativeness or invalidity of the original.

INFRINGEMENT

An infringement of a patent is the unauthorized making, using, or selling of an embodiment of the invention. In the case of a process, it is infringed by him who without ownership or license uses substantially the process of the patent. Actual knowledge of the patent by the infringer is not required provided the patentee has given the notice called for by the statute by marking his invention with the patent number, in default of which damage for prior infringement cannot be collected.

The measure of the monopoly of a patent is the claim or claims of the patent, and to be an infringement there must be a claim of the patent which is infringed. If what has been done or what has been made or used or sold is correctly described in the entire claim of a patent, if each element of the claim is present in the process performed or the thing that is made, used, or sold and is present in the relationship set forth in the claim, such claim is infringed. If for example, a claim contains three elements, those three elements or their equivalents must be present in the infringing structure. By "equivalent of an element" in a claim is meant something which performs the same function as the element and in substantially the same way.

COPYRIGHTS

BY SHELDON H. GRAVES

HISTORICAL

At common law before the invention of printing the works or writings of an author belonged to the author. After the adoption of printing, writings were reproduced in quantities and distributed to the public, the works became public property, and the author was without recourse until 1710 when an English statute gave to authors or their assigns the sole right of publication for a limited period.

The granting of copyright in the United States is under federal statute enacted under authority of the Constitution. The Constitution provides that Congress shall have power to secure for limited times to authors the exclusive right to their writings.

WHAT MAY BE COPYRIGHTED

"Writings" as used in the Constitution has been held to include all forms of writing, engraving, printing, and etching by which ideas of the author are given visible expression. The ideas expressed need not display any high degree of artistic merit. It is not necessary that the materials be new but may include compilation of known facts or data. Music is included as well as pictorial illustrations, photographs, and advertisements.

By act of Congress the author must specify to which of certain designated classes the works for which copyright is sought belongs, but an error in assigning the work to some particular class is not generally fatal.

The **statutory classes** are as follows:

1. *Books, including composite and cyclopedic works, directories, gazetteers, and other compilations.* The book need not be a bound book but may be a single sheet, leaflet, or folder. It may consist merely of tables of figures. It, however, does not include slogans or mottos which are not copyrightable. A compilation includes selected extracts from different authors.

2. *Periodicals, including newspapers.* Each number of a periodical should be separately copyrighted.

3. *Lectures, sermons, and addresses* (prepared for broadcasting or oral delivery). This includes unpublished manuscripts; when published these should be registered as a book.

4. *Dramatic or dramatico-musical compositions.* This includes plays, operas, and oratorios.

5. *Musical compositions.* This includes songs with words and music written in any notation but does not include perforated music rolls or phonograph records. However, the owner of a copyright on the composition can control the production of such rolls and records.

6. *Maps.* When accompanied by text maps should be copyrighted as a book.

7. *Works of art, models or designs for works of art.* Includes paintings, mosaics, carvings, statuary, plaques, and bric-a-brac. Where the artistic appearance is applied to articles of manufacture which have primarily a utilitarian purpose such as clocks, furniture, rugs, etc., the article is patentable as a design but cannot be copyrighted.

8. *Reproduction of works of art.* Includes sketches, photographs, or etchings of original works of art such as paintings or sculptures.

9. *Drawings or plastic works of a scientific character.* Includes anatomical figures and also stock-market charts.

10. *Photographs.* Includes photographs which embody artistic conceptions and also the mere mechanical reproduction of the appearance of any object.

11. *Prints and pictorial illustrations.* Includes illustrations of anything and mere decorative features such as an artistic border for a stock certificate. This class also includes commercial prints and labels for articles of merchandise.

12. *Motion pictures, photoplays.* Includes photoplays whether accompanied by sound or not. Also so-called shorts.

13. *Motion pictures, other than photoplays.* Includes educational films and newsreels.

Commercial prints and pictorial illustrations were until recently registered in the Patent Office but are now registered with the Register of Copyrights in the Library of Congress. A commercial print is an artistic work with or without text published in a periodical or separately in connection with the sale or advertisement of an article or articles of merchandise. A label is an artistic or literary work attached to an article of merchandise or to containers therefor to indicate the nature of the goods.

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A copyright is secured by publication with notice of copyright.

A work is published when the first authorized edition is "placed on sale, sold, or publicly distributed by the proprietor of the copyright or under his authority." From this quotation it would appear that the mere showing of a picture or motion picture is not a publication as required by law. A motion picture therefore would not be published until it is distributed. It is important to note that the publishing with notice of copyright precedes the application for copyright registration and that publication without such notice constitutes an abandonment of the claim of copyright.

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The application for registration should be accompanied by the statutory fee of $2 except for a print or label published in connection with the sale or advertisement of an article of merchandise, where the fee is $6, and of unpublished work and a photograph where no certificate is demanded, where the fee is $1.

INTERIM REGISTRATION

Provision is made for interim registration of foreign books pending printing in this country.

TERM

The term of a copyright is 28 years. The law provides for a renewal by certain specified persons for a renewed term of 28 years.

RIGHTS SECURED BY COPYRIGHT—INFRINGEMENT

As provided in the Copyright Act of 1909, as amended, the copyright owner has the exclusive right

> *Sec.* 1 *a*: to print, reprint, publish copy, and vend the copyrighted work. . . . A copy, is that which comes so near the original as to give to any person seeing it the idea created by the original.

It need not be a word-for-word copy. It may be copied in a different medium as showing a photograph on a screen. The copying of a substantial part of the original is an infringement. However, one may make a fair use of a copyrighted work. The copyright owner alone has the right

> *Sec.* 1 *b*: to translate the copyrighted work into other languages or dialects or to make any other version thereof if it be a literary work.

Versions include abridgments or telling the story in other words, but a mere sketchy outline would be a "fair use" and not an infringement.

> *Sec.* 1 *b* (2): to dramatize it if it be a nondramatic work.
>
> *Sec.* 1 *b* (3): to convert into a novel or other nondramatic work if it be a drama.
>
> *Sec.* 1 *b* (4): to arrange or adapt it if it be a musical work as an arrangement for an orchestra or to change the key or pedaling.
>
> *Sec.* 1 *b* (5): to complete, execute, and finish it if it be a model or design for a work of art.

These are works of the fine arts rather than industrial arts which are covered by the patent law.

> *Sec.* 1 *c*: to deliver or authorize the delivery of the copyrighted work in public for profit if it be a lecture, sermon, address, or similar production.

By *similar production* is meant a production intended in the first instance for oral communication.

> *Sec.* 1 *d*: to perform or represent the copyrighted work publicly if it be a drama [This provision also includes dramatico-musical works] or if it be a dramatic work and not reproduced in copies for sale, to vend any manuscript or any record whatever thereof. . . . [The performance or representation thereof need not be for profit]

This provision includes exhibition of motion-picture plays but does not include poems, as these may be recited in public without infringement.

> *Sec.* 1 *e*: to perform the copyrighted work publicly for profit if it be a musical composition. . . .

For profit does not mean that there must be paid admission to hear the music. The unauthorized broadcasting of a song in a radio program under a sponsor is infringement. The provision includes music with motion pictures.

There are certain exceptions from liability for unauthorized performance of music: (1) reproduction of music by coin-operated machines and (2) certain religious and secular works, even though an admission is charged, if the fee is applied to charitable or educational purposes.

> *Sec.* 1 *e* (continued): and for the purpose of public performance for profit and for the purpose set forth in subsection *a* hereof, to make any arrangement or setting of it (the musical composition) or of the melody of it in any system of notation or any form of record in which the thought of the author may be recorded and from which it may be read or reproduced.

In this section the reproduction from a phonograph or other record is not an infringement; the record itself is not a copy of the original composition, but the proprietor has the control of the manufacture of such records. The law, however, provides that, when the proprietor himself has used or sanctions the use in this way, anyone may make records on payment to the proprietor of 2 cents royalty for each record.

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